Available online at www.sciencedirect.com

SCIENCE DIRECT®

Journal of Sound and Vibration 274 (2004) 351–367

JOURNAL OF SOUND AND VIBRATION

www.elsevier.com/locate/jsvi

Vibration protection for an operator of a hand-held percussion machine

E.V. Golysheva*, V.I. Babitsky, A.M. Veprik

Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, Leicestershire LE11 3TU, UK

Received 11 October 2002; accepted 20 May 2003

Abstract

The present study shows that a passive system of vibration protection, which combines principles of vibration isolation and dynamic absorption, can be effectively used in hand-held percussion machines. The vibration isolator between the handle and the casing of the machine reduces the high-frequency components of acceleration perceived by an operator, whereas the lightly weighted tuned dynamic absorbers attached to the handle allow suppression of dominating harmonics of machine acceleration. Such a vibration attenuation system allows the use of relatively stiff isolators and significantly reduces hand-transmitted vibration without essential increasing the overall mass of a tool. Results of numerical simulations have been verified by experiment and these are presented in this paper.

1. Introduction

Hand-held percussion machines rely on systematic impacts for the demolition or treatment of hard materials. They are widely used in various fields of industry, construction and transport, primarily due to their convenience in service, high efficiency and adaptability for various types of operation processes. However, a common problem, associated with a systematic use of such machines is severe vibration resulting in injury to the operator. Regular exposure of hands and fingers to such a vibration typically results in a complex combination of signs and symptoms of physiological disorders (vascular, bone and joint, peripheral neurological, muscular, etc.) commonly referred to as the Hand–Arm Vibration Syndrome (HAVS) [1].

A survey in 1997/98 gave a national prevalence estimate of 301,000 sufferers from Vibration White Finger (VWF), a vascular disorder caused by hand and arm exposure to vibration [2]. This number does not include those people suffering from carpal tunnel syndrome and other diseases

*Corresponding author. Tel.: +44-1509-227-566; fax: +44-1509-227-502.
E-mail address: v.golycheva@lboro.ac.uk (E.V. Golysheva).

0022-460X/$ - see front matter
doi:10.1016/j.jsv.2003.05.019

caused by hand-transmitted vibration. The estimated annual cost of HAVS illness to the economy is £447 million [3], but more importantly, the quality of life of the operators of hand-held power tools is of particular concern. Although research is continuing, there is no effective treatment available for the vascular symptoms of HAVS and there is no treatment for its neurological component. Therefore, hand-transmitted vibration has gained increasing attention in the literature and is now regarded as one of the most important occupational hazards.

The electro-pneumatic hammer, which is one of the most representative piston-operated vibrating tools and is widely employed as a concrete breaker, hammer-drill and impact ripper in various areas of industry and construction, is analyzed in the present study.

Schematics of the electro-pneumatic hammer is shown in Fig. 1. The electric motor drives the exciting piston via the crank and connecting rod. The reciprocating piston periodically compresses and decompresses the air in the pneumatic chamber, thereby setting the striker in motion. The striker hits the intermediate piston that in turn hits the pick which then hits the material being treated. The pick has a collar that prevents it from going too far into the mechanism after rebounding against the object of treatment. The operator has to press the machine against the object of treatment thereby applying a permanent force (this is called the *feed force*). Theoretically, the operator can control the working process by changing the feed force and, therefore, operator–machine interaction should be taken into consideration when the dynamics of the percussion machine is being investigated [4].

The constant direct interaction between the operator and the hand-held power tool defines the necessity and the specific features of the vibration protection system for this type of machinery. From the operator perspective, the static stiffness of the flexible element between the hand and the tool should be high enough to ensure its small relative deflection along with a wide range of feed force which can be applied by the operator without affecting the stability of the operation. The weight of the vibration attenuation system is another limiting design factor, as additional weight of the tool will increase load upon the operator.

There are a variety of methods for protecting the operator from hand-transmitted vibration. Although the comprehensive classification is difficult, mention will be made of the main groups. The first method involves reduction of the vibratory effects imposed by the machine upon the operator by reducing the intensity of the sources of harmful vibration through the proper design of the machine. All sources of vibration cannot be completely eliminated under real working

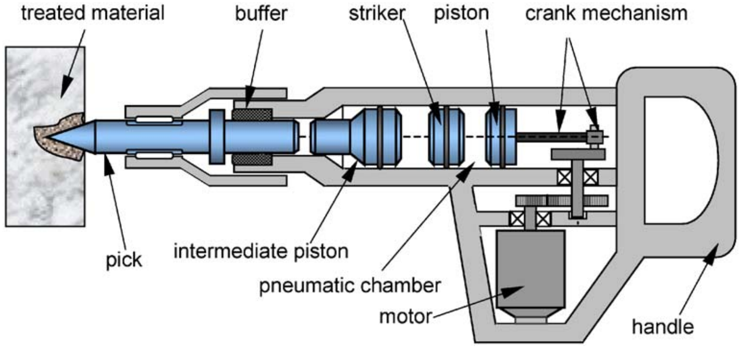

Fig. 1. Schematic cross-section diagram of electro-pneumatic hammer.

conditions. However, in some instances, the intensity of the hand-transmitted vibration may be reduced considerably [4–6].

The second and the most widespread method of vibration protection is vibration isolation. This is primarily attained using two approaches: isolation of the tool handle from the vibrating source and isolation of the hand from the vibrating handle (this is achieved by using gloves, energy flow dividers, operator substitution by a guided machine or a robot, remote control, etc.). However, simple passive systems of vibration isolation are usually effective only in a high-frequency range and cannot provide high static stiffness, as compared with dynamic one. Active vibration isolation systems can be designed in order to meet these criteria, but such systems are more complicated and in most cases the cost increase is unjustifiable.

The third method of vibration attenuation is dynamic absorption of vibration. The operating frequency of a typical hand-held percussion machine, supported by a drive mechanism with stiff characteristics, is constant. Hence, it is possible to effectively reduce the vibration by employing a dynamic absorber tuned to the driving frequency and attached to the machine. However, a conventional passive dynamic absorber is effective only in a narrow frequency range and causes an undesirable increase in weight of the tool.

In Refs. [7,8] it was shown that a tuned dynamic absorber in combination with a stiff and heavily damped vibration isolator could be used successfully for the simultaneous dynamic suppression of the fundamental harmonic of vibration and also over a high-frequency span. This principle can be used in the design of a vibration isolation system for hand-held percussion machines, where the tuned dynamic absorber is attached to a relatively light handle isolated from the casing of the machine. Application of such a vibration attenuation system in a hand-held percussion machine was first mentioned in Ref. [9]. Unfortunately, this idea was not developed further and no models, experiments or practical design employing this principle have been reported.

In a vibration attenuation system that combines the principles of dynamic absorption and vibration isolation, the handle is isolated from the casing of the percussion machine in order to achieve dynamic de-coupling between the handle and the casing. The mass of the handle is small in comparison with the mass of the whole machine. This means that successful suppression of the fundamental frequency of handle vibration can be achieved by employing a dynamic absorber with a small effective mass. At the same time, the whole system provides the static stiffness required by the operator. Since a dynamic absorber suppresses the fundamental frequency of vibration, a vibration isolator reduces vibration in the higher frequency range and its stiffness can be chosen from safety and controllability reasons.

Several dynamic absorbers may be used optionally depending on the parameters of the isolator placed between the handle and the casing; in the case where the magnitudes of the second or the third harmonics are still large, these may be successfully reduced by employing additional dynamic absorbers.

2. Dynamic model of electro-pneumatic hammer

In order to investigate the dynamics of a hand-held percussion machine and to develop a vibration isolation system, a lumped parameter dynamic model of an electro-pneumatic hammer was developed. The pneumatic impact in the pressure chamber, impact forces between the pick and the casing, the crank-slider mechanism and the impact force of the pick against the object being treated were included in the model as factors responsible for vibration. The coupling between the operator and the machine was simulated as a single-degree-of-freedom (s.d.o.f.) mass–spring–damper system between the handle of the machine and the rigid attachment. A simplified model of the operator's hand was chosen as a s.d.o.f. system with the following parameters: $k_1 = 2.43 \times 10^5$ N/m, $c_1 = 265$ N s/m, $m_1 = 1.4$ kg (values were obtained from experiment, the details of which are not discussed in the present paper). The model of visco-elastic impact, which relies on the Kelvin–Voigt model, is used as the basis for a description of collisions between parts of the machine [10].

Fig. 2 shows a dynamic model of the electro-pneumatic hammer. Here, masses $m_1, \ldots, m_5$, m_h represent the mass of the operator hand, the machine casing, exciting piston, striker, pick and handle respectively; k_2, c_2 are the parameters of the vibration isolation system (usually the rubber cushion on the handle); k_h, c_h are the parameters of the vibration isolation between the casing and the handle of the percussion machine; k_3, c_3 are the mechanical characteristics of contact between the striker and the pick; k_4, c_4 are the stiffness and the damping ratio of the buffer between the casing and the pick; k_5, c_5 are the parameters of the linear spring–damper combination reflecting the mechanical characteristics of the material being treated; F_{pn} represents the pneumatic force between the driving piston and striker; r is the radius of the crank; l is the length of connecting rod; S is the length of the compression chamber between the exciting piston and the striker when the pressure in the chamber is atmospheric; and d is the stroke length between the striker and the pick (when all elastic elements are not deformed, the exciting piston is in an extreme position from the pick, and the position of the striker is determined by the atmospheric pressure in the pneumatic chamber).

The feed force, produced by the operator, is accounted for by changing parameter a (the distance between the origin of absolute co-ordinates (0,0) and the attachment of the system on the right-hand side) under the assumption that the attachment on the left-hand side is fixed (see Fig. 2). It is suggested that the treated material is restored before each successive impact, therefore, the process of penetration into the material is not taken into consideration. Consequently, the object of treatment is represented as a linear spring–damper combination and does not change its position during the operation.

Non-linearity and the complexity of the model make analytical solutions difficult if not impossible; hence, the model was investigated numerically using Simulink (part of the Matlab software suite).

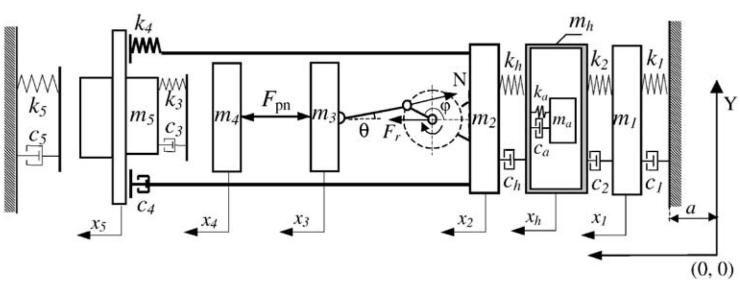

Fig. 2. Dynamic model of the electro-pneumatic hammer.

3. Numerical simulations

Some of the parameters of dynamic model of electro-hammer were taken from typical parameters of a heavy electro-pneumatic hammer drill (Hilti TE 74). The mass of the pick, $m_5 = 0.35$ kg; the mass of the machine casing, $m_2 = 8$ kg; the frequency of impact of the pick against the treated surface, $f = 45$ Hz. The approximate stiffness of the treated material, $k_5 = 3 \times 10^7$ N/m and the loss factor of the treated material, $\zeta_5 = c_5/2\sqrt{k_5 m_5} = 0.7$ were obtained using typical characteristics for concrete loading and formula (7) in Ref. [4]. The energy of impact was chosen as 1.5 J with an acceptable deviation value for the pick velocity just before the impact of 20%. In order to obtain a reference model, the case of an electro-pneumatic hammer without vibration isolation was initially considered (referring to Fig. 2, the stiffness of k_h and k_2 are very high, and the absorber is not attached to the handle).

The size of the pneumatic chamber, stroke length, parameters of the buffer between the casing and the pick, the feed force developed by the operator, the masses and the rest of the parameters used in the dynamic model of the machine were chosen in such a manner that the machine was working with a prescribed frequency and the energy of impacts and the vibration perceived by the operator was minimal. The procedure was carried out using an adaptive random search technique (Monte-Carlo optimization).

Thus, a model of the electro-pneumatic hammer without vibration isolation was developed. A detailed description of the dynamic model together with a Simulink block diagram were presented in our previous study [6]. Fig. 3 portrays results of numerical simulations in time domain. Fig. 3(a) shows the displacements of the pick, the striker and the piston; the pneumatic force (F_{pn}) developed in the air cushion as a result of their interaction is shown below in Fig. 3(b). The hand acceleration curve, as shown in Fig. 3(c) results from a combination of compression of the pneumatic spring (peak "A") and the impact of the pick against the buffer element of the casing (peaks "B" and "C"), where peaks "C" are caused by secondary rebound of the pick. The force of impact between the pick and the casing (F_4^{imp}) is shown in Fig. 3(d), where peaks marked "D" are the result of the impact of the casing that is pressed against the pick by the spring–dashpot combination, k_1, c_1 after the pneumatic impact. Fig. 3(e) represents the displacements of the pick and the casing. Parameters m_1, k_1 and c_1 that represent characteristics of an operator of an electro-pneumatic hammer may vary from one operator to another. Therefore, simulations were carried out for different values of these parameters and the qualitative characteristics of the hand acceleration and the force of pick-casing impact were found to remain unchanged.

The handle of a percussion machine is usually covered by a rubber-like material in order to isolate higher order harmonics perceived by the operator. This isolator was represented in the model as a spring–damper combination, k_2, c_2. As a first stage of vibration attenuation, the handle was additionally isolated from the casing of the machine. In the dynamic model it was reflected by introducing another spring–damper combination, k_h, c_h between the casing and the handle, as shown in Fig. 2. A minimum possible stiffness of an isolator is limited by restriction imposed on maximum relative displacement of the machine and the necessity to ensure the controllability of the machine by the operator. In order to obtain optimal parameters of vibration isolators providing maximum vibration attenuation and satisfying these criteria, simulations were run for different values of natural frequency and loss factor of vibration isolators.

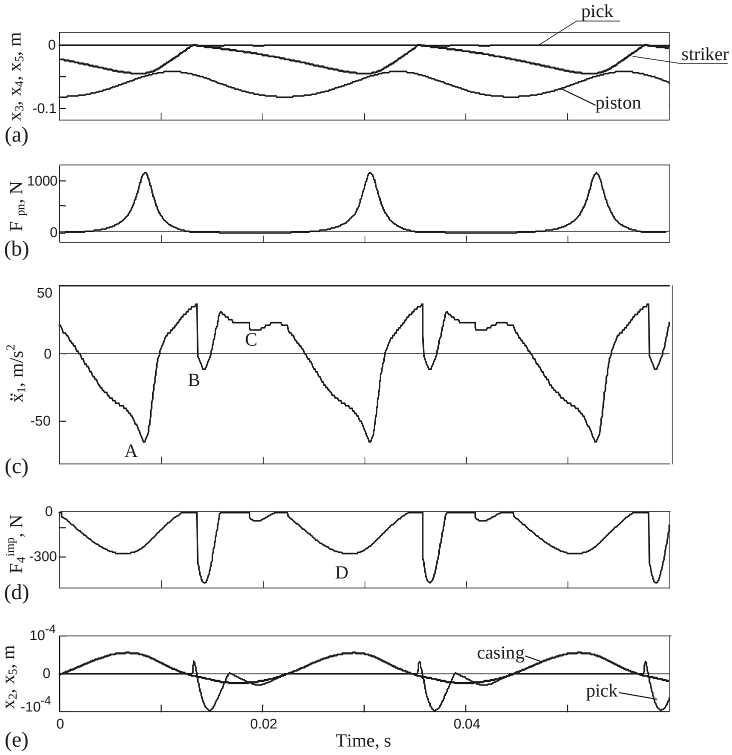

Fig. 3. Results of numerical simulations. (a) Displacements of the piston, the striker and the pick; (b) force of pneumatic impact in the air cushion, F_{pn}; (c) acceleration of an operator hand; (d) the force of impact of the pick against the buffer, F_4^{imp}; and (e) displacements of the casing and the pick.

Fig. 4 shows the r.m.s. of hand acceleration as a function of the parameters of the vibration isolator between the handle and the casing, where $\Omega_2 = \sqrt{k_2/m_1} = 802.3$ rad/s and $\zeta_2 = c_2/2m_1\Omega_2 = 0.22$. Similar functions were obtained for every other pair Ω_2 and ζ_2, so this set of parameters was chosen as one that provides maximum vibration attenuation. As can be seen, the r.m.s. of acceleration perceived by the operator of electro-pneumatic hammer is a function that decreases with decreasing natural frequency and loss factor of vibration isolator. For the sake of convenience, contour lines are also shown on a horizontal plane and labelled with correspondent r.m.s values. The fill area on a horizontal plane corresponds to regimes that satisfy the prescribed conditions (frequency of impacts 45 Hz, energy of impacts 1.5 J) and secure stability of the tool performance during possible variations of feed force or hand–arm parameters. There are a number of possible combinations of vibration isolators parameters that satisfy all conditions, but the greatest achievable effect of reducing the hand acceleration by simple vibration isolation does not exceed a factor of 2. Greater vibration attenuation by using isolators with smaller stiffness is not possible because it will cause disruption of the process and compromise its stability when the feed force is different from the prescribed one, thereby making it difficult for the operator to control the machine and worsen the safety of the working process. The following parameters were chosen for further investigation: $\Omega_h = \sqrt{k_h/m_h} = 389.8$ rad/s and $\zeta_h = c_h/2m_h\Omega_h = 0.24$.

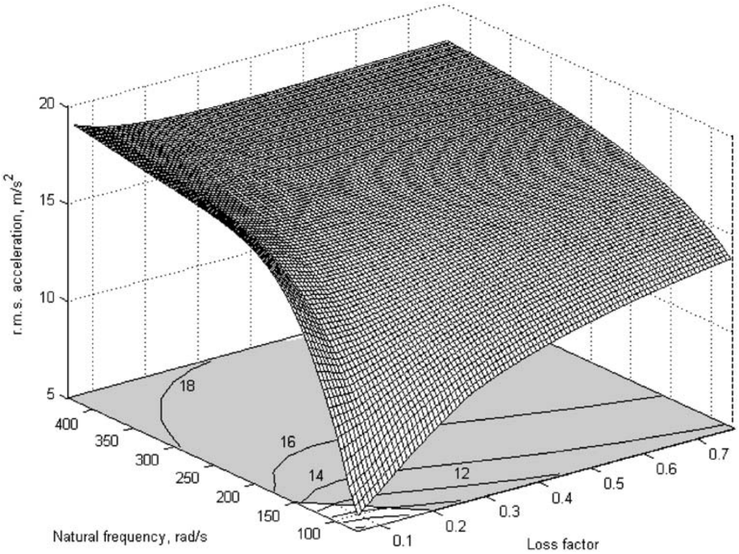

Fig. 4. R.m.s. of hand acceleration as a function of natural frequency and loss factor of the isolator between the handle and the casing (Ω_h, ζ_h), where $\Omega_2 = 802.3$ rad/s and $\zeta_2 = 0.22$.

Vibration isolators are ineffective at attenuating the vibration in the low-frequency range. At the same time, a tuned dynamic absorber attached to the handle can be successfully used for the suppression of the fundamental harmonic of acceleration perceived by the operator. In order to choose the optimum dynamic absorber, numerical simulations were run for different values of mass and loss factor of the absorber attached to the handle and tuned to the fundamental vibration frequency. The performance of the dynamic absorber was characterized using the following ratios between the characteristics of the hand acceleration for the electro-pneumatic hammer without vibration isolation and the same machine with a vibration attenuation system:

$K_{r.m.s.}$—suppression ratio of r.m.s. of the hand acceleration;
K_{fh}—suppression ratio of amplitude of the fundamental harmonic of the hand acceleration;
K_{PktoPk}—suppression ratio of peak-to-peak amplitude of the hand acceleration; and
K_{sh}—suppression ratio of amplitude of the second harmonic of the hand acceleration.

The results of the numerical simulations are presented in Fig. 5. The contour lines on the horizontal plane are drawn at regular intervals of magnitudes of the functions and give a good idea of the slope of the characteristics. According to these results, for a dynamic absorber with reasonably low damping (loss factor is about 0.01) tuned to the driving frequency, the essential reduction in the hand acceleration can be achieved even for a small effective mass of dynamic absorber. The amplitude of the second harmonic of hand acceleration is increased slightly compared with the system without the dynamic absorber. At the same time, the impact frequency, the energy of impacts, casing acceleration and the stability of operation are not affected.

A dynamic absorber with mass $m_a = 0.18$ kg, natural frequency $\Omega_a/2\pi = 45$ Hz, and loss factor $\zeta_a = c_a/2m_a\Omega_a = 0.01$ was chosen for further investigations.

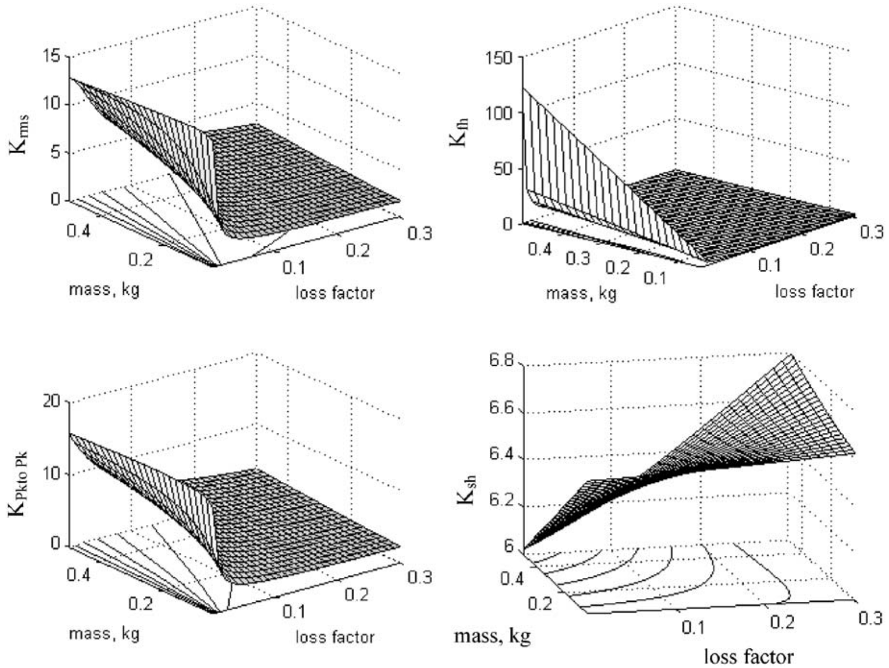

Fig. 5. Vibration attenuation system with one dynamic absorber attached to the isolated handle ($\Omega_h = 389.8\,\text{rad/s}$, $\zeta_h = 0.24$, $\Omega_2 = 802.3\,\text{rad/s}$, $\zeta_2 = 0.22$).

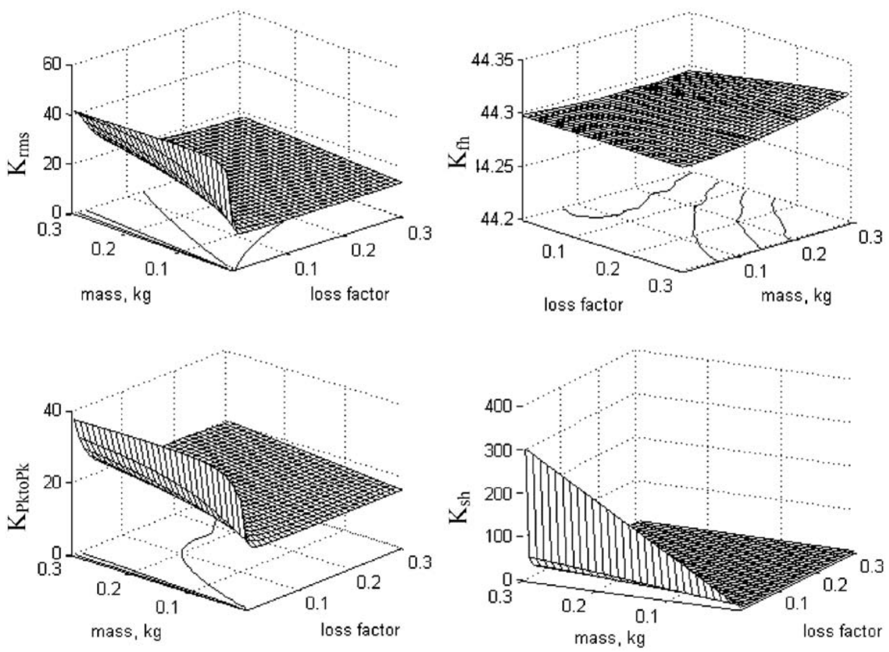

Fig. 6. Vibration attenuation system with two dynamic absorbers attached to the isolated handle as a function of the mass and loss factor of the secondary absorber (parameters of the prime absorber $m_a = 0.18\,\text{kg}$, $\zeta_a = 0.01$).

The second absorber tuned to the second harmonic of acceleration can further improve the effect of the vibration attenuation system under consideration. Fig. 6 shows results of numerical simulations for the system with two absorbers attached to the handle and tuned to the fundamental and second harmonics of the driving frequency, where $m_a = 0.18$ kg, $\zeta_a = 0.01$, and parameters of the second absorber are varied. As can be seen from Fig. 6, the second dynamic absorber attached to the handle increases the coefficient of r.m.s. and the peak-to-peak amplitude suppression ($K_{r.m.s.}$ and K_{PktpPk}) by a factor of approximately 2 and does not affect the fundamental harmonic of acceleration. Using the secondary dynamic absorber does not have any visible effect on the amplitude of the fundamental harmonic of hand acceleration.

Figs. 7 and 8 show spectra and time histories of acceleration perceived by the operator of electro-pneumatic hammer for different systems of vibration attenuation with the following parameters: $\Omega_2 = 802.3$ rad/s, $\zeta_2 = 0.22$, $\Omega_h = 389.8$ rad/s, $\zeta_h = 0.24$, $m_a = 0.18$ kg, $\Omega_a/2\pi = 45$ Hz, $\zeta_a = 0.01$, $m_{a1} = 0.035$ kg, $\Omega_{a1}/2\pi = 90.15$ Hz, $\zeta_{a1} = 0.01$. As can be seen, passive vibration isolators in the electro-pneumatic hammer allowed attenuation of the second and higher harmonics of the hand acceleration, and reduction in the r.m.s. of hand acceleration by a factor of 1.6 and peak-to-peak amplitude by a factor of 2.3. The fundamental and the second harmonic of acceleration were reduced by a factor of 1.2 and 8.5, respectively.

A dynamic absorber attached to the handle significantly improved the performance of the vibration attenuation system. The r.m.s. of the acceleration was reduced by a factor of 9 as compared to the original hammer. The magnitude of the fundamental harmonic of acceleration was reduced by a factor of 35 and the peak-to-peak amplitude of acceleration by a factor of 13.

A second dynamic absorber attached to the handle further improved the vibration attenuation effect. The r.m.s. of the acceleration of the operator hand reduced by a factor of 13 compared to the electro-pneumatic hammer without vibration isolation. The magnitude of the fundamental

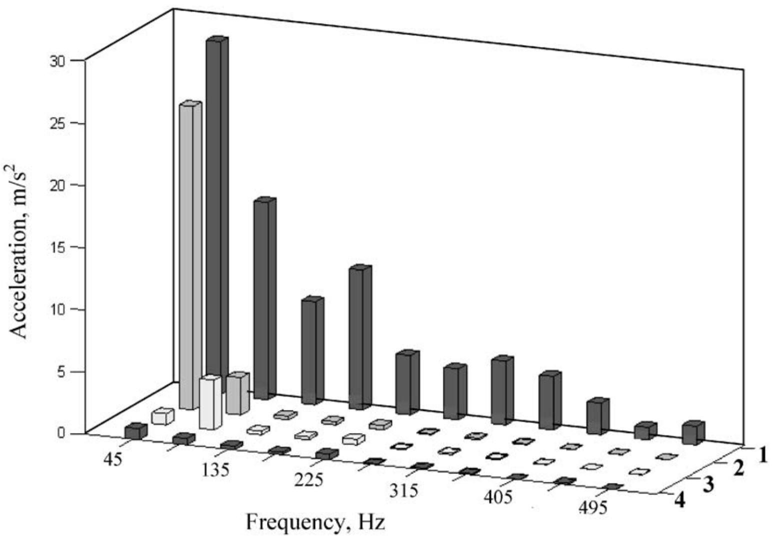

Fig. 7. Spectra of acceleration perceived by the operator for different systems of vibration attenuation (results of numerical simulations). (1) Original hammer without vibration isolation; (2) both-side isolated handle; (3) one dynamic absorber attached to the handle; and (4) two dynamic absorbers attached to the handle.

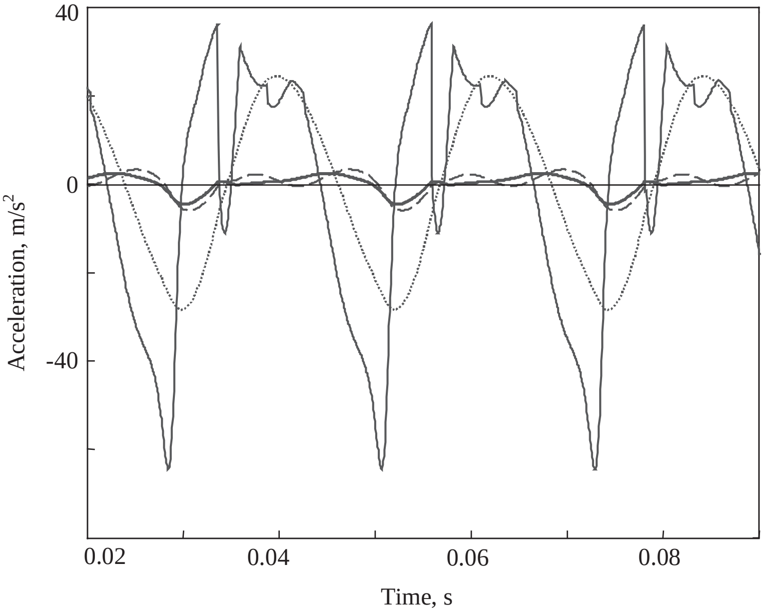

Fig. 8. Time histories of acceleration perceived by the operator of an electro-pneumatic hammer for different systems of vibration attenuation (results of numerical simulations); ——, original hammer without vibration isolation; ·········, both-side isolated handle; – – –, one dynamic absorber; ——, two dynamic absorbers.

harmonic of acceleration was reduced by a factor of 35 (the second absorber did not affect the fundamental harmonic of the oscillation), the second harmonic was reduced by a factor 50 and the peak-to-peak amplitude of acceleration by a factor of 20.

The stiffness of the vibration isolator placed between the casing and the handle plays an important role in the performance of the developed vibration attenuation system. Usually, the lowest possible stiffness of vibration isolator in a hand-held machine is limited by operation stability and controllability factors. On the other hand, in the vibration attenuation system considered, dynamic de-coupling between the handle and the casing can be achieved by employing an isolator with higher stiffness. However, when the stiffness of the isolator placed between the handle and the casing is high, the performance of the dynamic absorber, which has very small mass as compared to the casing mass, worsens. In the system being considered here, such an effect occurs when the natural frequency of the isolator (placed between the handle and the casing) is higher than 990 rad/s and depends weakly on the loss factor of the isolator.

As previously mentioned, the advantage of the dynamic absorber being attached to the isolated handle instead of the casing is the small effective mass of the dynamic absorber. In the system under consideration $\mu = (m_a + m_{a1})/m_m = 0.03$ (where m_m is the mass of the whole machine) while the absolute displacements of the primary and secondary absorbers were 2.85 and 0.6 mm correspondingly. In the case of a similar system with a dynamic absorber attached to the casing, the same effect in vibration reduction could be achieved for a mass ratio μ of 0.1. Such an increase in the weight of the machine is undesirable, because in addition to vibration disorders, workers often suffer from symptoms caused by the handling of heavy tools, such as strained muscles and joints.

4. Experimental verification

In order to verify the results obtained by numerical simulations, a mechanical model of handle–hand–arm was built, while the electro-dynamic shaker attached to the handle reproduced the casing acceleration.

4.1. Mechanical set-up

Fig. 9 shows a schematic diagram of the experimental rig. The experimental rig layout is shown in Fig. 10. The electro-dynamic shaker (Ling Dynamic System, model V409) was attached to the handle by a Shock Tech[1] wire isolator that represented the vibration isolator between the casing and the handle of the electro-pneumatic hammer. This wire isolator consisted of a stainless steel cable wound between light alloy bars and possesses weak non-linearity due to the friction between wires. However, such an isolator was used because the damping provided was ideal for the present experimental study.

The wire isolator had freedom of motion in vertical and torsion directions. Any possible misalignment between the moving platform of the shaker and the handle, might cause handle vibration in directions other than horizontal, which are not considered in the present study. In order to eliminate such vibration, four flat springs (width—30 mm, thickness—0.041 mm, length—200 mm) were placed between the handle and the mounting table (see Fig. 10). Such springs prevented the handle from vibrating in vertical or torsion directions without affecting the behaviour of the experimental mechanical system in horizontal direction.

The second smaller Shock Tech wire isolator was used as a vibration isolator between the handle and the hand–arm system. The mass connected to the handle and attached to the mounting table through the two wire isolators represented the operator hand and had a stiffness and damping ratio in accordance with parameters used in numerical simulations.

The electro-dynamic shaker was driven by a standard PC sound card through a power amplifier (B.K. Electronics MXF 400) and reproduced acceleration that was similar to the casing acceleration obtained by numerical simulations. The accelerations of the different parts of the system were measured using an accelerometer (Brüel & Kjær Type 4393) and amplified by a charge amplifier (Brüel & Kjær Type 2635) before it was passed to a dynamic signal analyser (Data Physics Corporation DP 104) as shown in Fig. 9.

The analogue signal applied to the shaker was converted from a digital signal generated in Matlab by means of a Data Acquisition Toolbox. In order to obtain acceleration on the output of the shaker similar to the casing acceleration obtained by numerical simulations, it was necessary to compensate for the influence of the entire electro-mechanical system that included the power amplifier, shaker and tested mechanical system.

4.2. Synthesis of the excitation signal

Distortion of the digital signal passed from the sound card can be overcome by measuring the complex frequency response function (FRF) between the output of the sound card and the

[1] See http://www.shocktech.com.

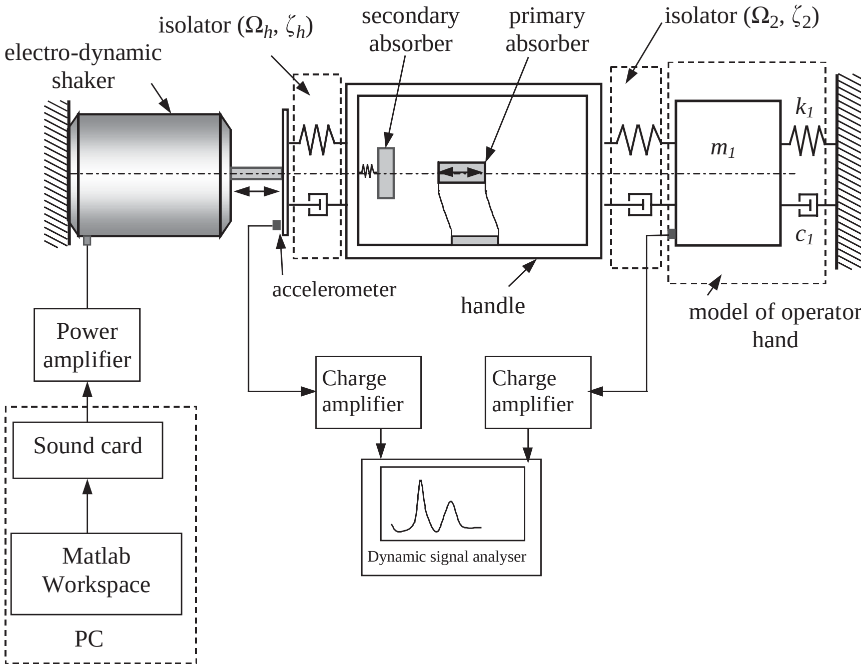

Fig. 9. Schematic diagram of the experimental rig.

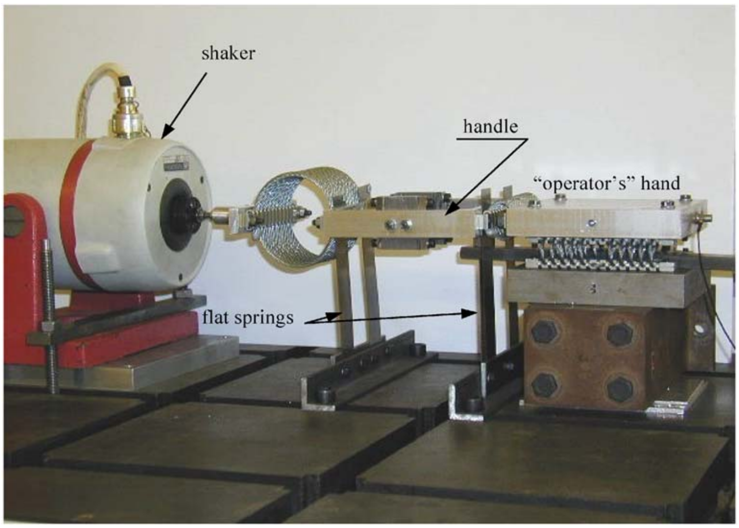

Fig. 10. Experimental rig.

acceleration of the mounting table of the shaker and by passing the digital signal through an inverse function before sending it to the sound card. Fig. 11 shows a typical module of the FRF measured between the output of the sound card and the acceleration reproduced by the shaker as a function of frequency. For such a function, retrieving an analytical expression of an inverse

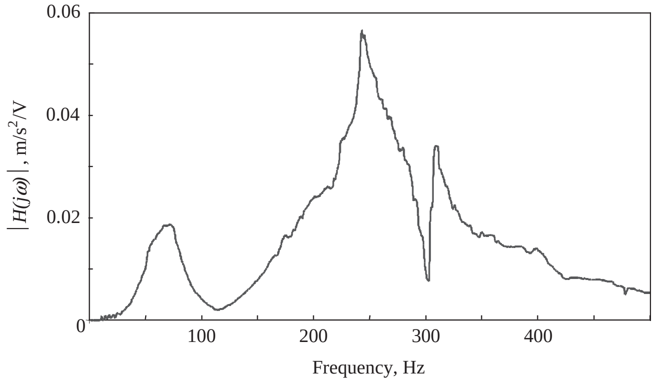

Fig. 11. The module of the FRF between the output of the sound card and the acceleration reproduced by the shaker.

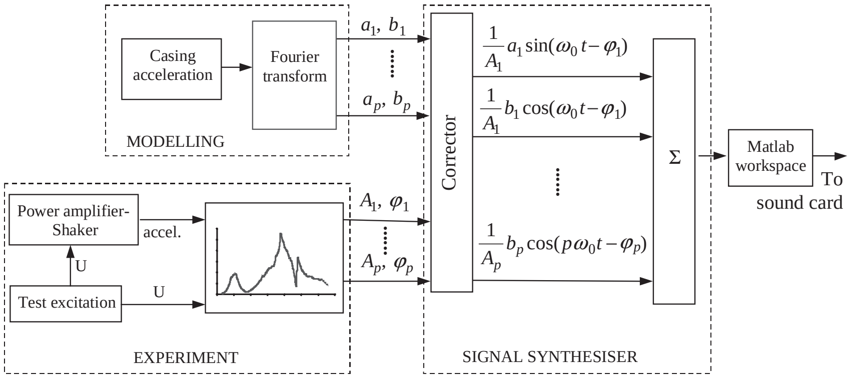

Fig. 12. Diagram showing synthesis of the excitation signal.

function that would be satisfactory for both the module and phase of the FRF is difficult; at the same time available electronics cannot provide the required amplification for a high-frequency range. In the case of periodic excitation, compensation for the dynamics of electro-mechanical system can be achieved by using the system response only at specific discrete frequencies that are a multiple of the driving frequency instead of the complete transfer function. Fig. 12 shows schematically the synthesis of the digital signal with compensated influence of the power amplifier and the shaker. The system response may be expressed in terms of amplitudes A_p and phases φ_p in the following manner:

$$A_p = |H(\mathrm{j}p\omega_0)|,$$
$$\varphi_p = \arg(H(\mathrm{j}p\omega_0)), \tag{1}$$

where $H(\mathrm{j}\omega)$ is the FRF obtained in the above experiment as a ratio between acceleration measured on the mounting table of the shaker and the analogue output of the sound card (see Fig. 12); ω_0 is the fundamental frequency; $p = 1, 2, \ldots, 11$. The FRF was measured in a 500 Hz frequency range because the amplitudes of higher harmonics of acceleration perceived by the operator were found to be insignificant.

Casing acceleration obtained by numerical simulations was represented by a Fourier series, namely, a series of sine and cosine waves of frequencies $p\omega_0$ and amplitudes a_p and b_p correspondingly. The final excitation compensated signal was synthesized as a sum of sine and cosine waves with phases φ_p and amplitudes calculated as a ratio a_p/A_p and b_p/A_p, as shown in Fig. 12. Due to system non-linearities, the values of A_p and φ_p were slightly adjusted manually in order to match the time history of the signal to the theoretical casing acceleration. As a result of the entire procedure the electrical signal applied to the shaker was synthesized in such a way that the acceleration reproduced by the shaker was similar to the casing acceleration obtained in the numerical simulations.

Fig. 13 shows the final electrical signal measured on the output of the sound card and the acceleration measured on the output of the shaker superimposed with the time history of the casing acceleration obtained by numerical simulations. It can be seen that the time histories and spectra of numerical and experimental accelerations are very similar. The r.m.s. of numerical casing acceleration is $29.5\,\mathrm{m/s^2}$, peak-to-peak amplitude is $106\,\mathrm{m/s^2}$ and for acceleration reproduced by the shaker these values are $27.4\,\mathrm{m/s^2}$ and $110\,\mathrm{m/s^2}$, respectively.

4.3. Experimental results

Once the casing acceleration was reproduced by the shaker, the effect of a vibration attenuation system can be compared with results of numerical simulations by measuring acceleration of the mass that represents the operator hand.

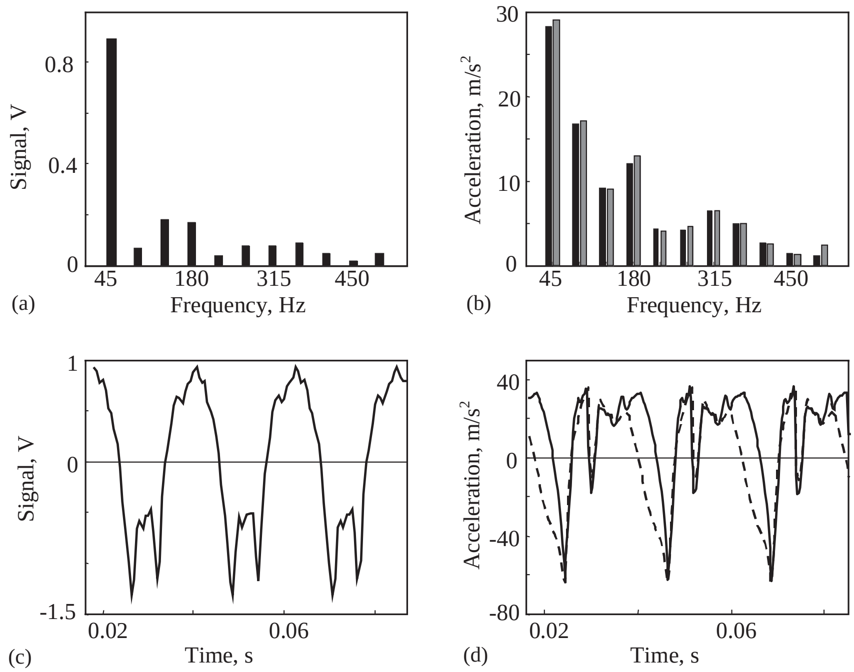

Fig. 13. (a), (c) spectrum and time history of electrical signal, (b) spectra of casing acceleration: ■, experimental, ▩, numerical; and (d) time history acceleration reproduced by the shaker (solid line) and casing acceleration obtained by numerical simulations (dashed line).

The first dynamic absorber ("primary absorber") attached to the handle, as shown in Fig. 9, was tuned to the fundamental frequency. The flexural of the above dynamic absorber was designed as four flat cantilever springs separated by steel liners in order to eliminate friction between the springs and to maintain the loss factor as low as possible. The absorber had a mass of 0.18 kg, while its natural frequency was tuned by displacing the mass along the flat springs to 45 Hz. In the case of a possible difference in the position between the mass of the absorber and the centre of gravity of the handle, the torque of the handle about its vertical axis was compensated for by flat springs between the handle and the mounting table. The cantilever design of the dynamic absorber worked well for experimental studies; nevertheless, a symmetrical design would be preferable for its practical implementation.

The secondary dynamic absorber tuned to the second harmonic of excitation had a mass of 0.035 kg. The mass of the absorber was attached to the handle by flexural elements that were designed as all-metal "Oxford" type flat springs providing a low loss factor.

Figs. 14 and 15 show experimental spectra and time histories of acceleration perceived by the operator of electro-pneumatic hammer for different vibration attenuation systems. As can be seen, experimental results are in good agreement with results obtained by means of numerical simulations (see Figs. 7 and 8). Employing vibration isolators between the vibrating casing and the operator hardly reduced the magnitude of the fundamental harmonic of acceleration, but the second and higher harmonics were attenuated very successfully. The r.m.s. of hand acceleration was reduced by a factor of 1.6, peak-to-peak amplitude of acceleration by a factor of 2.2, amplitude of the fundamental harmonic by a factor of 1.25 and amplitude of the second harmonic by a factor of 11.

The introduction of the dynamic absorber, tuned to the fundamental harmonic of the signal applied to the handle enabled a reduction in the r.m.s. of the hand acceleration by a factor of 8, and reductions in the magnitude of the fundamental harmonic by a factor of 62 and the

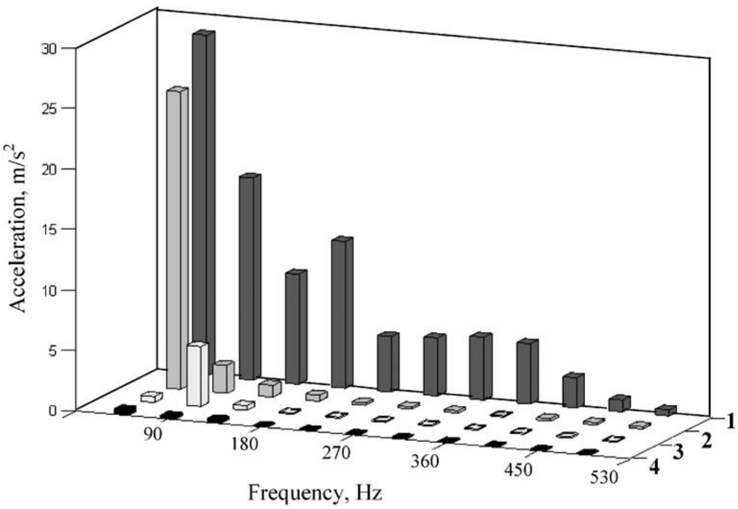

Fig. 14. Spectra of acceleration perceived by the operator for different systems of vibration attenuation (experiment). (1) Original hammer without vibration isolation; (2) both-side isolated handle; (3) one dynamic absorber attached to the handle; and (4) two dynamic absorbers attached to the handle.

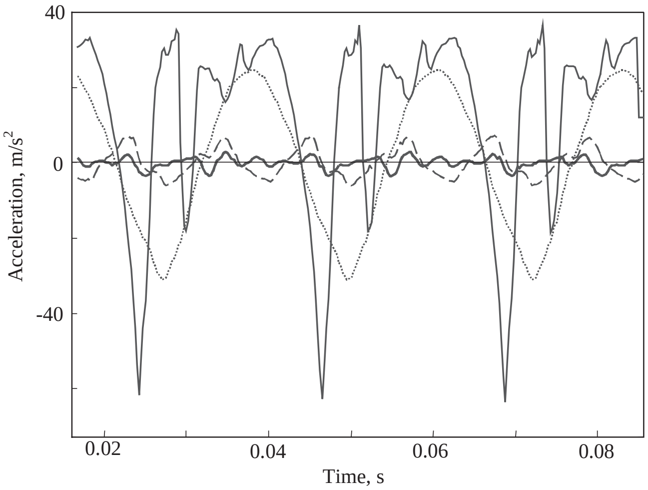

Fig. 15. Time histories of acceleration perceived by the operator of an electro-pneumatic hammer for different systems of vibration attenuation (results of numerical simulations); ——, original hammer without vibration isolation; ········, both-side isolated handle; – – –, one dynamic absorber; ——, two dynamic absorbers.

peak-to-peak amplitude of acceleration by a factor of 10 in comparison to the system without vibration isolation. Displacement of the mass that represented the operator hand was reduced by a factor of 20.

The vibration attenuation system with two dynamic absorbers attached to the isolated handle yields reduction in the r.m.s. of hand acceleration by a factor of 19, the amplitude of the fundamental harmonic by a factor of 62, the second harmonic by a factor of 70 and peak-to-peak amplitude of acceleration by a factor of 16. Displacement of the operator hand was reduced by a factor of 30.

5. Conclusions

A study showed that passive vibration attenuation systems that combined vibration isolation and dynamic absorption principles can significantly reduce vibration perceived by the operator of hand-held percussion machine. Such a system consists of the vibration isolators placed between the vibrating casing and the handle and the dynamic absorber attached to the handle for suppression of the dominant harmonic of handle acceleration that is not affected by vibration isolation. The developed system has several advantages that are particularly important for vibration protection of the operator of hand-held percussive power tools.

Firstly, the dynamic absorber reduces low-frequency harmonic of acceleration perceived by the operator. Therefore, there is no need to use vibration isolators with the lowest possible stiffness compromising controllability and safety of operation. Secondly, the dynamic absorber, attached to the handle and isolated from the casing, suppresses acceleration of the light handle only, so that the effective mass of the absorber is very small. This fact allows several dynamic absorbers tuned to different frequencies to be used, achieving an even greater effect of vibration attenuation without significant increase in the overall weight of the machine. Finally, such vibration attenuation systems can be easily implemented into the existing design of hand-held percussion machines without significant additional cost.

The performance of vibration attenuation system can be further improved. The spectrum of acceleration perceived by the operator contains linked frequencies. This fact suggests that the system could be improved by employing a dynamic absorber that would suppress several linked frequencies simultaneously.

Acknowledgements

The first author would like to gratefully acknowledge the financial support of the Engineering Faculty of Loughborough University.

References

[1] M.J. Griffin, *Handbook of Human Vibration*, Academic Press, London, 1990.
[2] Health and Safety Executive, *Health and Safety Statistics 2000/2001*, HSE Books, Suffolk, 2001.
[3] Health and Safety Executive, August 20-last update, Regulatory impact assessment of the physical agents (vibration) directive 2002/44/EC. [Online], 2002. Available: http://www.hse.gov.uk/ria/vibrate/vibrate.pdf
[4] V.I. Babitsky, Hand-held percussion machine as discrete non-linear converter, *Journal of Sound and Vibration* 214 (1998) 165–182 (Erratum (1999) 222).
[5] S. Ohtsu, Y. Kawakami, K. Kaneda, Study on vibration reduction of concrete breakers (2nd report, Vibration reduction due to the dynamic balancer driven by positive cam, and vibration isolation due to the elastic coupling between crank shaft and motor), *Transactions of Japan Society of Mechanical Engineers* 476 (1985) 1218–1223.
[6] E.V. Golycheva, V.I. Babitsky, A.M. Veprik, Dynamic correction of excitation in hand-held electro-pneumatic percussion machines, *Journal of Sound and Vibration* 259 (2002) 829–843.
[7] A.M. Veprik, V.I. Babitsky, N. Pundak, S.V. Riabzev, Vibration control of linear split stirling cryogenic cooler for airborne infrared application, *Shock and Vibration* 7 (2000) 1–17.
[8] A.M. Veprik, V.I. Babitsky, N. Pundak, S.V. Riabzev, Vibration and thermal control of a linear cryogenic cooler, *Proceedings of 19th International Cryogenic Engineering Conference (INEC 19)*, Grenoble, France, Narosa Publ. Home, New Delhi, 2003, pp. 515–518.
[9] I.I. Byhovsky, B.G. Goldshtein, *Principles of Design of Vibration-Safe Portable Tools*, Moskow, Mashinostroenie, 1977 (in Russian).
[10] V.I. Babitsky, A.M. Veprik, Universal bumpered isolator for severe environment, *Journal of Sound and Vibration* 218 (1998) 269–292.