SYNTHESIS OF DYNAMICAL STRUSTURES FOR DESIGN OF VIBRO-SAFE HAND-HELD PERCUSSION MACHINES

V.I. Babitsky, I.J.Sokolov, N.A.Halliwell

Wolfson School of Mechanical & Manufacturing Engineering, Loughborough University, UK

ABSTRACT

Synthesis of a dynamical and engineering structure of the hand-held percussion machine with a vibration-free handle/casing is presented. This is based on the consideration of the optimal dynamic conversion of energy in hand-held percussion machines founded as a solution of the equivalent problem of the optimal control [1]. Dynamic structure of the percussion machine considered as a multi-body vibro-impact system and a solution with one body (handle/casing) being free of vibration is found and investigated. Computer simulations and experiments validated the new approach. The key element of the proposed engineering design realising the synthesised structure is a mechanism with zero differential stiffness. A hydro-pneumatic unit with zero stiffness was developed and tested. Theoretical recommendations have been applied to a commercial hydraulic breaker. A proper modification of the design and effective use of existing hydraulic power source provided a significant improvement in performance.

INTRODUCTION

In [1] the formulation of a general approach to analysis and synthesis of hand held percussion machines using methods of nonlinear dynamics and optimal control was presented. The optimal dynamic conversion of energy in hand-held percussion machines was found to reveal the general relations between the character of excitation of vibro-impact process in machine as well as the main factors of machine-operator interaction and machine output. The optimal machine was presented as a *discrete converter* transforming the operator efforts into sequence of impact impulses. The generation of impulses with a minimum influence on the operator was formulated as an equivalent *optimal control*. Such a control has been found as a solution of mathematical *problem of moments* in the proper normed linear space.

An exact solution of the formulated problem was found in an explicit analytical form. This solution provides a way of absolute estimation of the improvement potential for all modifications of hand-held percussion machines. For the simplification of analysis and synthesis of the real machine design, a sequence of exact estimations was developed for the series of possible quasi-optimal excitations. Percussion machines of different design can be classified now according to the sequence of the quasi-optimal excitations. The electro-pneumatic machines exploit an excitation with condensed acceleration impulse while the working cycle of the hydraulic breakers is closer to optimal excitation with extended impulse of striker acceleration.

For the system shown in Fig. 1,a with a prescribed period of vibration *T*, impact velocity v_0 of the striker 1 and a single impact during the period, the optimal periodic excitation force *u(t)* that has minimum amplitude (deflection): $\min_{u} \max_{t\in[0,T]} |u(t)|$ is shown in Fig. 1,b. Here *t* is time, t_1 is the time interval (part of the period *T*) when the excitation force *u(t)* is positive and U_0 is the amplitude of the excitation force. The time interval t_1 and amplitude U_0 are defined by formulae [1]:

$$\frac{t_1}{T}=\frac{1}{1+R}\left(1-\sqrt{1-\frac{1-R^2}{2}}\right),\quad U_0=\frac{M_1(1+R)^2 v_0}{2T\left(\sqrt{1-0.5(1-R^2)}-0.5(1-R)\right)},\qquad (1)$$

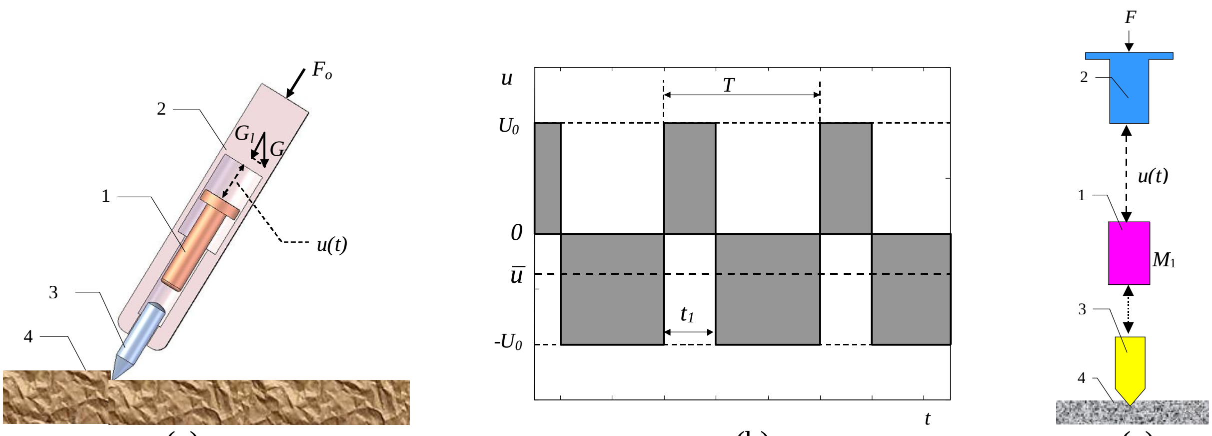
where $R=-v_+/v_0$ indicates the change of the absolute velocity of the striker (Item 1 in Fig. 1,a) during impact, v_+ is its upward velocity after impact and M_1 is the mass of the striker. Hereafter positive values of the force applied to the striker and positive displacements correspond to the upward direction along the machine axis.

The force of excitation *u(t)* is applied to both the striker and the casing (Item 2 in Fig.1 ,a). If the friction between these two bodies and the friction between the tool (Item 3 in Fig.1 ,a) and the casing are negligibly small, the excitation force *u(t)* is the only source of the casing vibration. The optimal excitation was found for a system without friction. In what follows we refer to the media being treated (Item 4 in Fig. 1,a) as *ground*; in all figures constant forces are shown with solid line arrows, alternating forces are shown with dashed line arrows and the sequence of impact impulses is shown with dotted line arrows.

The force of excitation *u(t)* has a constant component $\bar{u}$ and an alternating component $\tilde{u}$ given by equations 1:

$$\bar{u}=-M_1 f(1+R)v_0,\quad \tilde{u}(t)=u(t)-\bar{u}\qquad (2)$$

Here $f=1/T$ is the frequency of impacts. The constant component $\bar{u}$ depends only on the parameters M_1, R, of the system and the parameters T, v_0 of the regime. This component remains the same for any other excitation that provides the same parameters T, v_0 of the vibro-impact regime with a single impact per period [1]. As a result, for the system with parameters M_1 and R the desired vibro-impact regime of operation with parameters T and v_0 is available only if the operator is able to apply a constant feed force F_o that together with the longitudinal component G_l of the casing weight G balances the constant component $\bar{u}$ of the excitation force, i.e.



(a) (b) (c)

Fig. 1. Optimal excitation with minimum amplitude.

$$|\bar{u}|=G_l+F_0=F\,.\qquad (3)$$

Here F is the full feed force. For simplicity we will not distinguish the operator's feed force from the full feed force. However, it is important to note that the latter force includes the casing weight component that depends on the breaker orientation. In what follows when the short term *feed* *force* is used it refers to the full feed force. According to (2), for any excitation that provides a vibro-impact regime with impact velocity v_0 and single impact during a period T the full feed force can be found as

$$F = \frac{M_1(1+R)v_0}{T} = M_1(1+R)v_0 f \ . \tag{4}$$

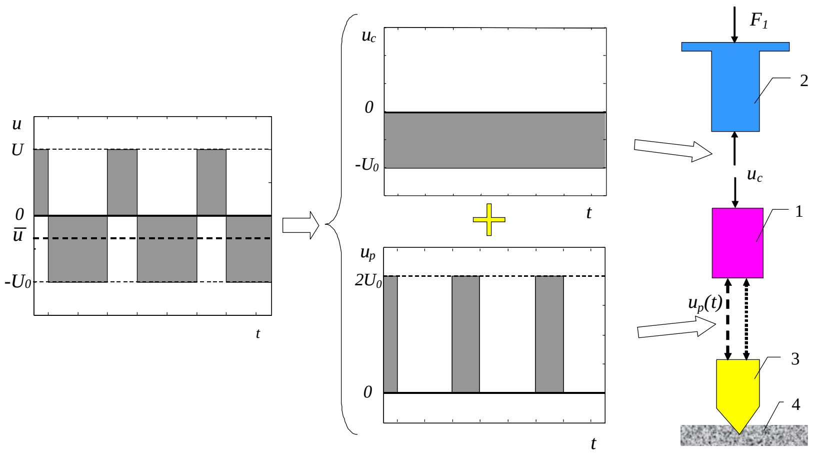
For the optimal excitation shown in Fig. 1,b:

$$F = \left(1 - \frac{2t_1}{T}\right)U_0 . \tag{5}$$

The equivalent structure of such a system with negligibly small friction is shown in Fig. 1,c. For this system a reduction in harmful vibration transmitted to an operator can only be achieved by increasing the casing mass or by adding extra inertial bodies. Increasing the mass of the casing is a very straightforward solution with obvious disadvantages. In this paper other means of improving the design of hand-held percussion machines are investigated. The aim is to reduce significantly the operator's exposure to the hazardous vibration for prescribed intensity of the vibro-impact working process and without increasing the machine's mass. In this aspect the first approach is to change the design so that the reaction force of the alternating component of the excitation is applied not to the casing, but to the ground. This can dramatically reduce vibration of the casing and is analysed thoroughly in the next section.

2. OPTIMAL SOLUTION FOR THREE-BODY SYSTEM.

Let us split the excitation force into two components $u(t) = u_c + u_p(t)$, as Fig. 2,a ,b shows. The advantage of this splitting is that the alternating (pulsating) component $u_p(t)$ of the excitation force applied to the striker is now unidirectional and upward.



(a) (b) (c)

Fig. 2. Splitting excitation force into two components

As a result, this component can be applied between the striker 1 and the tool 3. In this case the unidirectional downward counterforce of this component of the excitation force is applied directly to the ground 4 (via the tool 3) as Fig. 2,c shows. The constant component u_c can be applied between the striker 2 and the casing 3. The only force applied to the casing now is the constant component of the excitation force. This component can be balanced by the constant feed force F_1. As a result this mechanical system can theoretically have a strong vibro-impact process on the one (lower) side and total absence of vibration on the other (upper) side. This is the optimal three-body system with minimal emission of harmful vibration. Stable vibro-impact regimes can be obtained in such a system and this has been confirmed by numerical simulations of the structure shown in Fig. 3. To the authors' knowledge this structure have not found wide industrial application. This can possibly be explained by the fact that such a design still has one of the disadvantages intrinsic to the scheme as it needs a higher operator's feed force. The feed force needed $F_1=U_0$ is higher than the feed force F for the traditional system (Fig. 1):

$$\frac{F_1}{F} = \frac{1}{1-2t_1/T} > 1 \qquad (6)$$

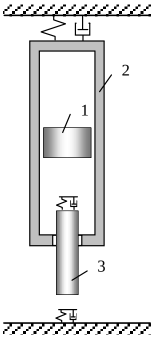


Fig. 3. Mechanical structure of three-body system

As was mentioned above, the optimal excitation with minimal amplitude (1) is optimal again in the new system as in this case it minimises the feed force. However, if, for example, $t_1/T=1/3$ the feed force needed for a vibration-free tool is three times as much as for the initial design (Fig.1). In what follows we show that the introduction of an extra inertial body allows for a further theoretical solution of the problem of low vibration emission without the necessity for increasing the operator's feed force.

3. OPTIMAL SOLUTION FOR FOUR-BODY SYSTEM.

In the system shown in Fig. 4,a the striker 1 is subjected again to a sequence of upward impact impulses.

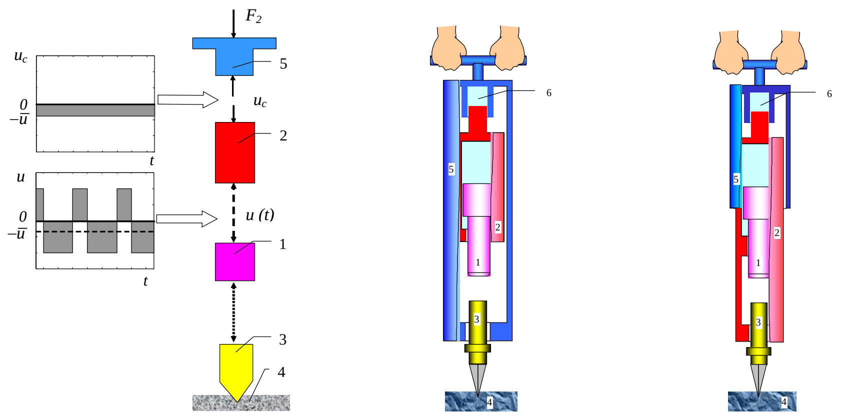


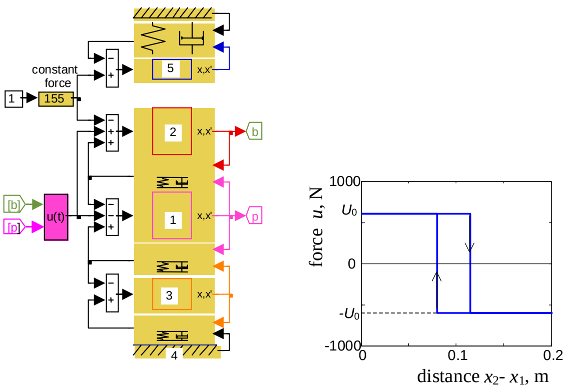
Fig. 4. Four-body system

These impulses are balanced by the excitation force *u(t)* with the constant downward component $\bar{u}$. Inevitably, the excitation force *u(t)* has an alternating component to provide the periodic vibro-impact motion of the striker. The excitation force *u(t)* is applied between the striker 1 and the inertial body 2. Unlike the striker, the latter body can be balanced on average by a downward constant force $u_c = \bar{u}$, but it still vibrates.

The inertia force resulting from this vibration balances the alternating component of the excitation force. For this reason hereafter we conventionally call this body a 'balance body'. The constant force is applied in this system between the balance body 2 and the body 5. The latter can be balanced by a constant feed force F_2 and can carry a vibration-free handle. This is the optimal four-body system with minimum emission of harmful vibration and minimum operator's feed force $F_2 = F = |\bar{u}|$. This scheme can be developed into two designs shown in Fig. 4,b and Fig. 4,c . Each of these modifications has both advantages and disadvantages. The main advantage of the former design is that the tool 3 is the only external part that vibrates in a steady-state regime, while the whole casing 5 does not vibrate. However, this design needs more complex arrangements to provide the alternating excitation force between bodies 1 and 2, both moving inside the motionless casing. The latter design is simpler; although only the handle 5 is free of vibration, while two other external parts (the casing 2 and the tool 3) vibrate.

Fig. 5,a shows an aggregated Mathlab-Simulink model corresponding to the system under review. This is a multi-body mechanical system with nonlinear (impact) interaction of moving parts. Linear differential equations were used to describe the continuous motion of free bodies between impacts or without impacts. Computational difficulties related to discontinuity of some variables during impacts can be avoided by using a Kelvin-Voigt model of a viscous-elastic contact interaction of colliding bodies as introduced in [2,3].

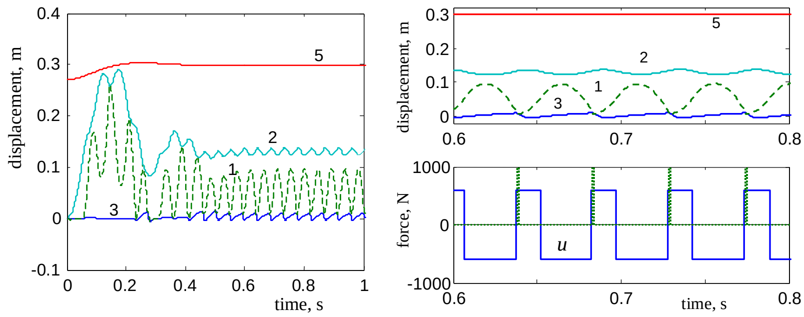
In this self-sustained system the excitation force *u(t)* is controlled by the relative position x_2-x_1 of the striker 1 and the balance body 2. The simplest relay law of control with U_0=600N was used as shown in Fig. 5,b. Such excitation is very close to the one used in real hydraulic and pneumatic percussion machines.



(a) (b)

Fig. 5. Model of four-body system

Fig. 6,a presents an example of the resulting transient process and the stable steady-state regime of vibration of the four-body system; curve numbers correspond to the numbers of the bodies in Fig. 4,a. Fig. 6,b shows the steady-state regime in more detail (upper diagram) along with the excitation force *u* and impact force between the striker and the tool (lower diagram). Impact instants in the latter diagram are shown with the dotted line. These results were obtained for the following parameters: M_1=1.2kg, M_2=5kg, M_3=3.5kg, M_5=10kg, R_{12}=0.25, R_{24}=0.15, F_2=155N. The simulation has shown that the desired stable periodic regimes of vibration with single impact during a period of vibration can be obtained in a four-body system with a vibration-free handle. It is important that these regimes have low sensitivity to parameters of the system.



(a) (b)

Fig. 6. Transient process and steady-state regime of four-body syst

4. ZERO-STIFFNESS SUSPENSION

Both designs shown in Fig. 4,b and Fig. 4,c feature some ideal element 6 that provides a constant force of interaction between bodies 2 and 5. The structure in Fig. 4,c is in fact a very particular case of the common one of a percussion machine with the vibro-isolated handle. The main advantage of the new system - vibration-free handle/casing - is a result of the fact that the new ideal suspension ensures constant force and hence it has zero differential stiffness in contrast to a common design with a spring-suspended handle [6].

A question can arise that inaccuracy in the zero-stiffness suspension constant force may destroy the necessary vibro-impact regime. Computer simulations have shown, however, that the vibro-impact regime has a very low sensitivity to the feed force. Fig. 7 demonstrates how the vibro-impact regime of the system shown in Fig. 4,a changes with variation of the operator's feed force.

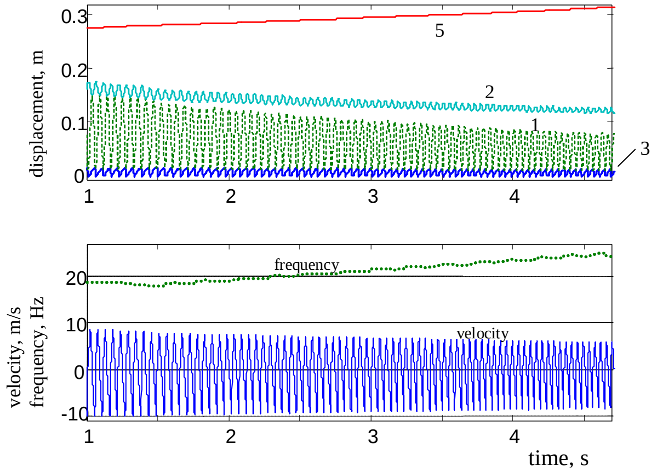


Fig. 7. Feed force variation: effect on vibro-impact regime.

These results were obtained for the same parameters M_1, M_2, M_3, M_5, R_{12}, R_{24}, but for the force F_2 gradually rising from 110N to 185N. The upper graph shows the displacements of four bodies; the curve numbers correspond to the numbers of the bodies in Fig. 4,a. The lower one shows the velocity of the striker and the frequency of self-excited vibration. Frequency in the latter diagram is plotted with the dotted line. It can be seen that the frequency of vibration rises and the impact velocity drops slightly with an increase in the feed force. However, the system demonstrates low sensitivity to the variation of feed force and the vibration remains stable when this force changes within a wide range. It is also important for this structure that in the general case the amplitude of the balance body 2 vibration rises with a decrease in its mass, but this does not affect the handle 5. A zero-stiffness suspension has a great potential to reduce both the harmful vibration and the total mass of heavy-duty hand-held percussion tools.

The zero-stiffness suspension can be naturally introduced into pneumatic and hydraulic machines that are very common in manufacturing and construction. Solutions can be found for using such elements in suspension of electric machines as well.

If a source of constant pressure is available some noticeable leakage through the gap between a piston and a cylinder is admissible. Hence there is no need to make this gap very small and as a result very low friction can be achieved. In this case a pneumatic cylinder can be very close to an ideal zero-stiffness element. However, an ideal source of constant pressure is not always available. In practice a pressure supply can have noticeable pulsations that can reduce all advantages of zero differential stiffness.

For example, hydraulic breakers are often used with a simple constant flow power pack. Reciprocating motion of the striking piston driven by the constant flow of hydraulic fluid results in high pulsations of pressure. Such breakers have a build-in gas accumulator designed to smooth the pressure pulsations. However, the remaining pulsations are still too high for this source of pressure to be used in the zero stiffness suspension. A possible solution for this case is shown in Fig. 8 which allows for filtering of the high-frequency pressure pulsations. These pulsations can be suppressed significantly by increasing the volume V_0 of compressed air under the piston. pressure is available (Fig. 9).

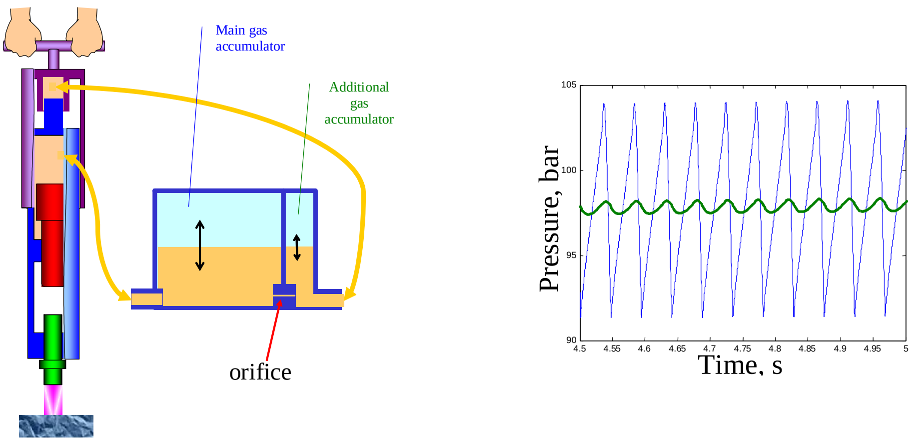


Fig. 8. Combination of gas accumulators in hydraulic breaker with hydro-pneumatic suspension

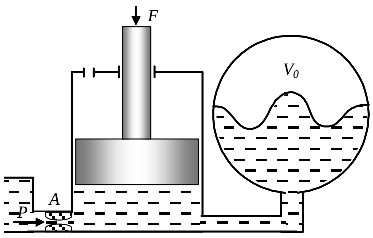


Fig. 9. Hydro-pneumatic suspension

For high-frequency vibration the air volume below the piston can be considered as isolated from external pressure supply and adiabatic law can be applied: $PV^{\gamma} = P_0 V_0^{\gamma}$, where P is pressure, V is volume, γ=1.44 for air. The differential (dynamic) stiffness is calculated as

$$c_d = \frac{dF}{dx} = S\frac{dP}{dV}\frac{dV}{dx}\bigg|_{V=V_0} = \frac{S^2 \gamma P_0}{V_0}. \qquad (7)$$

Here S is the piston area; P_0 is the average pressure; V_0 is total volume of air below the piston in the working position. Dynamic stiffness is inversely proportional to volume V_0 and can be made as low as needed increasing this volume. Also in this case the value of the gap between the cylinder and the piston is more important. If the effective area of the gap is comparable with the effective area A of the inlet orifice it affects the average pressure P_0 in the cylinder.

In order to make a simple comparison between the hydraulic and pneumatic systems the expression for dynamic stiffness (7) can be rewritten in the following form:

$$c_d = \gamma F^2 \frac{1}{P_0 V_0}, \qquad (8)$$

where F=P_0S is the feed force. This shows that for the prescribed feed force F the same low value of dynamic stiffness c_d can be achieved in a hydraulic system with a smaller volume V_0 of compressed air than in a pneumatic system. This is because the working pressure P_0 in hydraulic systems is usually much higher than in pneumatic systems.

As dynamic stiffness can easily be made very low when using hydraulic or pneumatic devices, more attention should be paid to friction (damping) that becomes the governing factor in transmission of vibration. Traditional bellows and modern rubber-metal analogues can be considered as a substitute to cylinders in some cases with the aim of reducing friction in the zero-stiffness suspension.

The solution shown in Fig. 8 has been patented by JCB [7]. It was successfully tested on a modified commercially available heavy-duty hand-held hydraulic breaker HM25. Experiments have shown that more than a twofold increase in suppression of vertical vibration in the experimental model can be achieved compared to the original breaker with spring-suspended handles [8].

These experiments have also shown that an operator does not have any serious difficulties while operating a breaker with zero-stiffness suspension, although the feel is slightly different as compared to operating a breaker with spring-suspended handles. The new commercial model HM25LV (Fig. 10) incorporating this solution was released recently by JCB.

Fig. 10 New HM25LV hand-held commercial breaker

Despite being more powerful than predecessor HM25, this breaker has much lower hand and arm vibration (HAV) levels than competing models. Operators can safely use this machine for a full eight-hour day without exceeding new EU strict recommended levels for HAV exposure.

5. CONCLUSIONS

It is possible to design a hand-held percussion machine with an intensive vibro-impact process on one side and a vibration-free handle on the other side. The simplest three-body system with one vibration-free body needs higher operator's feed force than conventional machines. In the four-body system, however, this can be achieved without increase in the feed force. Feasibility of this solution was proved by computer simulations and tested experimentally. This approach has a great potential of reduction in vibration transmitted to the operator. In practice this reduction is limited only by friction in the zero-stiffness suspension of the handle.

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