Work modeling processes of the hand-held impact machines

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Abstract. The research is devoted to the study of the processes occurring when the hand-held machine is machined by the impactor and their effect on the operator's work. To calculate the level of force impact for performing operations and to evaluate the transmitted vibration from the machine body to the hands of the operator, a model for interaction of the operator-machine-workable system, which is universal for any processed material, was created. The working cycle of the machine is divided into six main phases, for which the main energy indicators are determined. The calculation of the vibration acceleration of the machine body at different stages of the working cycle is made, the most vibro-dangerous phases of the cycle are identified, the elements of the machine are identified that need new design solutions, a layout scheme is proposed, at which the area of the vibration damper is significantly increased and the impact of the striker on the body of the machine is significantly reduced in the return stroke.

1. Introduction
Today, one of the most promising ways of connecting the building fittings is the connection in couplings of various types (crimping, threaded, and bolted). In civil engineering, it seems advisable to use fittings for diameters of up to 18 mm in diameter, which are pieces of steel cylindrical pipes. The crimping operation can be performed by various equipment, but the most compact and universal in terms of climatic and other operating conditions is a hand-operated electric tool, as evidenced by its wide prevalence on construction sites and at home. In this research, we propose the use of hand press machines with a linear electromagnetic drive as press equipment.

When the operation of a manual pulse machine, intensive shock processes occur, which are necessary to perform work operations with the workpiece, which excite the body vibrations transmitted to the operator's hands that have limitations specified in the sanitary norms [1], [2]. Therefore, it is necessary to determine the natural frequencies of machine oscillations, as a complex oscillatory system in the frequency range 5-2800 Hz, in which the vibration of the hand-operated machine must be ensured.

2. Experimental
For further calculations, we use the parameters of the developed V.A. Kargin and A.D. Abramov hand-operated impact machine UIM-3, which meets the requirements of the technological operation at the required energy level of a single impact.

The operating cycle of hand-operated impact machines with a linear electromagnetic drive is repeated in 1 second. This cycle can be divided in time into 6 stages with substantially different dynamic processes:
The 1st phase is a direct acceleration of the striker under the action of an electromagnetic force that occurs as a result of the current interaction in the coil of the machine with the ferromagnetic anchor bolt. Its motion is described by a sinusoidal wave, shown in Figure 1.

![Image of sinusoidal wave](image)

Figure 1. Characteristics of the striker acceleration.

Where \(a\) is the acceleration of the striker, the \(t\)-cycle time.

The 2d phase is the inelastic impact of the striker on the punch. The deformation of the workpiece is carried out with the joint action of the striker and punch. Since the punch and the punch move together after impact, their collision is assumed to be inelastic.

Formulas of the classical impact theory can be used if [3]:

\[
\beta = \frac{t_y}{T} > 3 \div 5
\]  

(1)

Where \(t_y\) - is the duration of the impact, \(t_y = 11,5 \times 10^{-3} \text{s}\), \(T\) - is the period of natural oscillations of the bodies, as systems with distributed parameters, in which elastic tensile-compression waves are transmitted when impact.

\[
\beta = \frac{11,5 \times 10^{-3}}{2,35 \times 10^{-4}} = 48,9 > 5.
\]

The impacted body (punch) before the impact is motionless, that is \(v_{02} = 0\). The initial velocity of the striking body is \(v_{01} = 12 \text{ m/s}\). When the striker strikes the punch, they start joint motion, so we take the recovery factor \(K = 0\) (absolutely inelastic impact). In this case, the joint speed of the "striker-punch" system will be:

\[
v_g = \frac{m_1v_{01}}{m_1 + m_2}
\]

(2)

Where the mass of the striker is \(m_1 = 0.9 \text{ kg}\), \(v_{01} = 12 \text{ m/s}\), punch weight \(m_2 = 0.27 \text{ kg}\).

\[
T_g = \frac{m_1 + m_2}{2} v_g = \frac{1,17}{2} \times 9,2^2 = 49,1 \text{ J}
\]

The 3d phase is the elastoplastic deformation of the workpiece.

To assess the energy intensity of operations and determine the boundary values of the impact energy, the lowest of which will be the minimum energy of a single impact, at which a qualitative realization of the technological operation is possible, the upper value will be the level of vibration acceleration transferred to the handle of the device, depending on the energy of a single impact and the frequency of impacts. In this paper, the process of interaction between the operator-machine system and the workpiece is simulated.
Figure 2. The model of the operator-machine system – the workpiece: 1 – shock assembly; 2 – vibration isolator; 3 – body-handle; 4 – model of the operator’s hand; 5 – the limiter; 6 – model of the body being treated; \( c_1 \) – the rigidity of the medium in the elastic zone; \( F_t \) – resistance of environment to the beginning of introduction; \( c_2 \) – conditional stiffness of the medium in the plastic zone; \( V_0 \) is the pre-shock speed of the striker, \( h_k \) is the sample sediment.

The special feature of this model is its versatility for all types of hand-operated impact machines and any bodies exposed to impact. The model shown in Figure 4 includes the hand of the operator holding and activating the tool, the impact machine itself, with a set of movable \( m_1 \) and fixed masses \( m_2 \), and directly the model of the workpiece, which are two springs with conditional stiffness, in the elastic \( c_1 \) and the plastic \( c_2 \) deformation zone, as well as the dry friction damper \( F_t \), describing the static load characteristic. The use of such a model makes it possible to calculate the value of the workpiece draft \( h_k \) according to the previously known formula for several successive cycles of dynamic loading \( k \):

\[
h_k = \sqrt{\left(\frac{F_t}{c_2}\right)^2 \left(\frac{c_1}{c_1+c_2}\right)^k + \frac{mV_0^2 c_1}{c_2^2} \left[1 - \left(\frac{c_1}{c_1+c_2}\right)^k\right] - \frac{F_t}{c_2}}
\]

where \( c_2 \) is the nominal resistance of the plastic deformation part, N / m; 
\( c_1 \) – conditional stiffness of the part in the zone of elastic deformation, N / m; 
m – weight of the striker, kg; 
\( V_0 \) – speed of the striker immediately before the contact, m / s.

The impact energy of the projected machine, which is necessary to deposit the workpiece by an amount \( h_k \) for \( k \) impacts, can be obtained from equation (4):

\[
T \geq \frac{h_k(c_2h_k+2F_t)}{2k}
\]

Under the influence of the forces occurring during the working stroke of the striker and the impact of the punch and the treated body, the machine body moves. The movement spreads in three coordinate axes, but since such machines are produced and designed in the overwhelming majority of cases with pistol-type handles, long-term studies and regulatory documents state that the most critical for the machine operator is the direction of the vibrational movement of the machine body with the handle along the axis of the striker's movement. This circumstance makes it possible to consider the model of interaction between the operator and the handle of the device as two-dimensional, which simplifies further calculations.

Calculations show that with the energy of a single impact of 50 J, it is possible to obtain a qualitative connection of the reinforcement bars from 10 to 18 mm by means of crimping in a steel cylindrical bushing. Depending on the size of the rods, the tooling is replaced and the number of strokes required to complete the operation is increased.

The 4th phase is the transfer of direct shock to the body through the workpiece and the contact connection of the matrix to the body. The diagram of this interaction is shown in the figure. The
machine in question is low speed impact. The impact frequency is 1-2 Hz, the total number of strokes for one operation does not exceed 15.

**Figure 3.** Contact interaction of system elements.

1 – firing pin; 2 – punch; 3 – workpiece; 4 – matrix; 5 – contact stiffness; 6 – housing.

Impact pulses are approximated in the form of a half-sinusoid. Consequently, expanding in a Fourier series we obtain:

\[ x(t) = \frac{2x_0}{\pi} \left( \frac{1}{2} + \frac{1}{4}\cos \omega t + \frac{1}{16}\cos 2\omega t - \frac{1}{32}\cos 4\omega t + \frac{1}{64}\cos 6\omega t - \cdots \right), \quad (5) \]

where \( \omega = \frac{2\pi}{T} = 2\pi f \).

The acceleration calculation of the hull and striker joint motion at the moment of the article finishing treatment appears to be an irresolvable analytical method even for bodies of regular geometric shape, since the elastic and plastic properties of the material change with each impact. In addition, the task of selecting the impact energy for the material to be treated should virtually eliminate or minimize the acceleration of the hull at the moment of rebound by absorbing the vibrations of the treated body. Therefore, such a problem can be solved only for the case of an almost absolutely rigid body – the moment when the blows on the product continue after the crimping termination.

The case of a hard collision can be considered using the kinetic energy equation:

\[ \frac{m_bv_b^2}{2} - \frac{m_kv_k^2}{2} = 0. \quad (6) \]

The calculated vibration velocity in the case of a rigid collision is 5, 1 m / s, which does not exceed the standard value of 5.270 m / s. It should be noted that the actual vibration speed will be significantly lower, since the mass of the body will be added to the mass of the workpiece, which, as a rule, exceeds the mass of the machine body.

This phase of work is controlled by the operator and can be ruled out with timely termination of work.

The 5th phase is a reverse acceleration of the striker by the return mechanism under the action of the spring.
Figure 5. Calculation scheme for the return movement of the striker.

The coordinates $x_n$ are counted from the equilibrium position.

Differential equation of motion of the striker under the action of the spring force of the return spring:

$$m_1\ddot{x}_1 + c\dot{x}_1 = 0 \Rightarrow \ddot{x}_1 + \frac{c}{m_1}x_1 = 0$$  \hspace{1cm} (7)

Knowing the free move of the striker $A_2 = 0.06$ m, it is possible to calculate the maximum speed of the striker's movement with a return motion $v_{2\text{max}}$

$$v_{1\text{max}} = A_1\omega_1 = 0.06 * 111.8 = 6.7 \text{ m/c}$$  \hspace{1cm} (8)

$v_{1\text{max}}$ is the speed just before the impact of the striker with the rubber gasket - the damper of the case of the return mechanism.

The kinetic energy of the striker before impact:

$$A_{\text{return}} = \frac{m_1v_{\text{2max}}^2}{2} = \frac{0.9 \cdot 6^2}{2} = 20.2 \text{ N} \cdot \text{m.}$$

The 6th phase is the kick of the striker against the body through a rubber gasket.

To get started with this step, you need to get the numerical value of the stiffness of the gasket:

$$\frac{\Delta h}{h} = \frac{F_y}{ES} \Rightarrow F_y = \frac{ES}{h} \Delta h$$  \hspace{1cm} (9)

it is possible to obtain the stiffness of the gasket, knowing the elastic modulus of the rubber $E = 0.5 \cdot 10^7 \text{ N}/\text{m}^2$:

$$C_p = \frac{ES}{h} = \frac{0.5 \cdot 10^7 \cdot 593.5 \cdot 10^{-6}}{3 \cdot 10^{-3}} = 989 \cdot 10^3 \text{ N}/\text{m}.$$  \hspace{1cm} (10)

force $N_m$:

$$N_m = V_1 \sqrt{\frac{c_p m_0 m_1}{m_0 + m_1}} = 6.7 \sqrt{\frac{989 \cdot 10^3 \cdot 5.09}{4.7 + 0.9}} = 5972.9 \text{ N}$$  \hspace{1cm} (11)

time of impact $t_y$:

$$t_y = \pi \sqrt{\frac{m_0 m_1}{c_p (m_0 + m_1)}} = 3.14 \sqrt{\frac{5.09}{989 \cdot 10^3 \cdot 5.6}} = 8.9 \cdot 10^{-3} \text{ c.}$$  \hspace{1cm} (12)

The use of the classical theory of absolutely rigid bodies impact is possible under the following conditions:

$$\frac{t_y}{T} > 3 \div 5$$  \hspace{1cm} (13)
\[
T = 2 \cdot 90 \cdot 10^{-3} \sqrt{\frac{7.9 \cdot 10^3}{2.1 \cdot 10^{11}}} = 3.5 \cdot 10^{-5} \text{ c.}
\]

\[
\tau_T = \frac{8.9 \cdot 10^{-3}}{3.5 \cdot 10^{-5}} = 254.3 \geq 3.
\]

Since the ratio of the time of impact to the frequency of natural oscillations satisfies the given condition, it is possible to use the classical theory of hard collision.

Next, it is necessary to calculate the vibration safety of hand-operated impact machines during the processing of products.

**Figure 6.** Schematic diagram of the system.

1— the body of the machine; 2 – kick with a return mechanism; 3 – rigidity of the workpiece, \(C_{\text{wb}}\); 4 – a returnable spring; 5 – rubber gasket; \(C_6\) – stiffness of the return spring; \(C_p\) – the rigidity of the rubber gasket; \(C_n\) – stiffness of the bias spring; \(m_k\) is the mass of the shell; \(m_p\) is the mass of the punch; \(m_6\) is the mass of the striker.

It is necessary to determine the natural frequencies of the oscillations that occur during the shock impulse action on the workpiece. To compile the differential equations of free vibrations of a vibro-impact machine (VIM), we use the Lagrange equations of the II kind:

\[
\frac{d}{dt} \left( \frac{dT}{dq_i} \right) - \frac{dP}{dq_i} + \frac{dl}{dq_i} = 0
\]

(14)

The computational model is a mechanical system with three degrees of freedom.

The kinetic energy of the system, the inertial elements of which perform rectilinear free oscillations, is determined by the formula:

\[
T = \frac{m_1x_1^2}{2} + \frac{m_2x_2^2}{2} + \frac{m_3x_3^2}{2}
\]

(15)

Potential energy of the system:

\[
P = \frac{(C_p+\bar{C}_6)(x_1-x_2)^2}{2} + \frac{(C_n+C_{\text{wb}})(x_1-x_3)^2}{2}
\]

(16)

The coordinates \(x_1, x_2, x_3\) are counted from the equilibrium position. Further we substitute \(T\) and \(P\) into equation (2. *). As a result, differential equations of free oscillations of the system are obtained.

\[
\begin{align*}
&\begin{cases}
m_1\ddot{x}_1 + C_2(x_1 - x_2) + C_3(x_1 - x_3) = 0 \\
m_2\ddot{x}_2 + C_2(x_1 - x_2) = 0 \\
m_3\ddot{x}_3 + C_2(x_1 - x_3) = 0
\end{cases}
\end{align*}
\]

(17)

We assume, in the first approximation, that all the elastic elements in the mechanical system are linear. Then the free oscillations of inertial elements \(m_1, m_2, m_3\) will be harmonic:

\[
x_i = A_i \sin \omega t, \quad \dot{x}_i = A_i \omega \cos \omega t, \quad \ddot{x}_i = -A_i \omega^2 \cos \omega t
\]

where \(i = 1, 2, 3\); substituting in the differential equation (2. *):

\[
\begin{align*}
&\begin{cases}
-m_1A_1\omega^2 + (C_2 + C_3)A_1 - C_2A_2 - C_3A_3 = 0 \\
-m_2A_2\omega^2 + C_2A_2 - C_2A_1 = 0 \\
-m_3A_3\omega^2 + C_3A_3 - C_3A_1 = 0
\end{cases}
\end{align*}
\]

(18)
After substituting numerical values:
\[
\begin{vmatrix}
(7 \cdot 10^8 - 5\omega^2) & -322 \cdot 10^4 & -7 \cdot 10^8 \\
322 \cdot 10^4 & (322 \cdot 10^4 - \omega^2) & 0 \\
7 \cdot 10^8 & 0 & (7 \cdot 10^8 - 0,3\omega^2)
\end{vmatrix} = 0
\]

Further, expanding according to the rule of the triangle:
\[
(7 \cdot 10^8 - 5\omega^2) (322 \cdot 10^4 - \omega^2) (7 \cdot 10^8 - 0,3\omega^2) = 0. \tag{19}
\]

We assume \(\omega^2 = \gamma\). After the expansion of equation (19), it takes the form of the third order algebraic equation, which is solved with the help of a computer:
\[
ay^3 + by^2 + cy + d = 0. \tag{20}
\]
\[
\omega^6 - 23,3 \cdot 10^9 \omega^4 + 32,7 \cdot 10^16 \omega^2 - 99 \cdot 10^{22} = 0.
\]
\[
\gamma^3 - 2,3 \cdot 10^9 \gamma^2 + 32,7 \cdot 10^16 \gamma - 99 \cdot 10^{22} = 0.
\]

\(f_1 = 278\ \text{Hz}, f_2 = 1928\ \text{Hz}, f_3 = 7431\ \text{Hz}\). The predicted vibration levels in these frequencies are not less than 2 times greater than the levels in neighboring frequency bands. At these frequencies, it is possible to exceed the sanitary standards of vibration acceleration in the phase of the striker impact against the body when returning to its original position, so it is necessary to improve the damping element.

3. Results and Discussion

It was found that the most successful solution is the universal low-frequency manual impact machine with a linear electromagnetic motor (LEM) for the reciprocal action of the UIM-3, the principal scheme of which is shown in Fig. 7.

Figure 7. Diagram of the created hand-operated impact machine prototype with a spring return mechanism: 1 – body; 2 – electromagnetic coil; 3 – striker; 4 – a returnable spring; 5 – damper; 6 – matrix holder; 7 – punch; 8 – matrix; 9 – a spring; 10 – the guide; 11 – damper.

This design diagram of the machine has several drawbacks. The main ones are the low reliability of the return spring working on the principle of tension, due to the concentration of stresses in the places of its attachment to the striker and the body, as well as high vibration loading in the phase of impact of the striker against the body when returning.

To solve these problems, a new design solution [4] is proposed, based on the improvement of the layout scheme of the previously mentioned known machine, which consists in the use of a compression spring, which allows to increase by 3.2 times the area of the rubber gasket - the vibration damping damper hitting the striker against the body when returning to its original position, which in turn reduces the impact force, perceived by the body 57 times. Figure 13 shows an example of the device design being developed with a tool for crimping steel cylindrical bushings at the ends of reinforcing bars. The peculiarity of this scheme is, in addition to an improved return mechanism, a detachable matrix holder, which is a necessity for such a technological operation.
Figure 8. Schematic diagram of the device for crimping the valve.

1 – body, 2 – electromagnetic coil of the direct travel, 3 – massive body of the striker, 4 – impactor, 5 – thrust element, 6 – return spring, 7 – damper, 8 – protective casing, 9 – detachable matrix holder, 10 – removable part of the matrix holder, 11 – a punch, 12 – a matrix, 13 – a clamping spring, 14 – a fixing ring, 15 – the holding mechanism of a ring, 16 – a processed product.

4. Conclusion
Thus, the construction diagram of the striker return system drive, with a compression spring located in a separate housing, allows by increasing in the area of the damper by 3.2 times increase the reliability of the machine and reduce the vibration load on the operator to sanitary standards.

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