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Theoretical analysis of the functioning of the system "operator – portable pneumatic punch" and conditions of ensuring occupational safety

Abstract

The efficiency and safety of the "operator – portable pneumatic punch" systems are associated with a whole complex of unresolved problems to date, related both to the issues of rock destruction physics, and to the imperfections of the designs of modern pneumatic punchers and problems in biology as a science based not on the laws of physics, but on the opinions of the authorities, which does not allow to obtain objective quantitative assessments of the impact on the human body of the influencing factors. This work contains an analysis of the functioning of each element of this system on the final result – productivity, as well as safety, based on existing theoretical works in this field, as well as new ones, which resulted in the identification of effective ways to improve this system, resulting in a significant increase in productivity and occupational safety.

Keywords: pneumatic, perforator, drilling, safety, evaluation, vibration, productivity.

1. Problem status

Portable pneumatic perforator used for drilling blast holes are characterized by heavy mass, high vibration of the handles and noise in the workplace, and also require a lot of effort from operators, low drilling speed and other disadvantages.

Table 1 shows the parameters of the Atlas Copco BBD 94W pneumatic perforator as an example.

Table 1 – Some technical parameters of the Atlas Copco BBD 94W pneumatic punch Atlas Copco BBD 94W

mass, kg	28
length, mm	670
Compressed air consumption, l/s	97
Frequency of strokes, 1/min.	3300
Vibration level on 3 axes (ISO 5349-2) / safe level, м/s^2	15 / 2
Sound pressure level / , r=1 м / safe level, dB(A)	114 / 75

As can be seen from Table 1, the vibration of the handles and the noise in the workplace significantly exceed the permissible limits.

An analysis of the scientific works of various authors in this field has shown that numerous theoretical and experimental studies of various designs of pneumatic impact machines and their working tools, including pneumatic perforators, drill rods and drills, have mainly focused on optimal operating conditions, where the machine body is pressed against the working tool and the working tool is pressed against the material being broken.

As a result, these machines delivered a full-strength impact to the material being destroyed, the impact piston delivered a full-strength impact to the working tool, and so on [1, 3, 7, 8, 9, 25].

Theoretical studies of the operation of pneumatic impact machines, including pneumatic perforators did not consider the quantitative assessment of operators' participation in the work process, but only the hygienic assessment of their work [1, 2].

However, it is these issues that ultimately determine the real safety and performance of pneumatic impact machines, including portable pneumatic perforators, which are the most heavy and dangerous machines.

To solve these problems, let's consider the functioning of the "operator-perforator-drilling rod-drilling crown-destructible material" system in real-world conditions. In this case, we assume that the final result of this system's functioning is the drilling speed.

The time spent on drilling the holes includes all the operations required by the technology of this process: carrying the perforator to and from the drilling site, installing the drill rod in the perforator, drilling, and repositioning the drill rods.

Here, the actual drilling speed – the net drilling speed – characterizes the main process, while other operations represent auxiliary processes. As is known, the net drilling speed depends on the following main parameters [3]:

- the energy of the piston-striker's impact on the drilling rod;
- the frequency of the impacts;
- the feeding force;
- the type of the drilling bit;
- the strength of the material being destroyed.

Thus, in general, the process of drilling holes can be represented as a differential equation:

$$\frac{dh}{dt} = f(j_h, n, Q_t, Q_o, \psi, f_s), \tag{1}$$

where h is the depth of the drilled borehole; t is the time; j_h is the energy of the piston-striker's impact on the drilling rod; n is the frequency of impacts; Q_t is the technical feed force; Q_o is the operator's feed force; ψ is the coefficient of energy transfer from the drilling rod and drill bit; and f_s is the strength of the material being destroyed.

The "operator-perforator-drilling rod-drilling crown-destructible material" system consists of links that are sequentially connected, as shown in Figure 1.

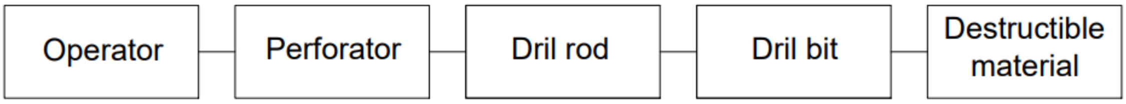


Figure 1 – Structural diagram of the system “operator – perforator – drill rod – drill bit – material to be destroyed”

From here, equation (1) can be represented as:

$$\frac{dh}{dt} = f(j_h) \cdot f(n) \cdot f(Q_t) \cdot f(Q_o) \cdot f(\psi) \cdot (f_s). \quad . \tag{2}$$

As can be seen from equation (2), each element of the system under consideration operates according to its own laws. However, together, they transform the energy of compressed air entering the drill into the energy of rock fracture under the influence of gravity, pneumatic support, and the driller's efforts. To determine the patterns of interaction between the elements of this system, as well as its overall operation, let's examine the functioning of each element.

2. Analysis of the dynamics of material destruction

The drilling conditions and requirements for pneumatic perforators, drill rods and drill bits are determined by the properties of the materials being destroyed.

One of the main parameters of the materials being destroyed, which play a key role in the destruction process, is their hardness, which is classified according to the widely used Prof. Protodeakonov.

Perforation drilling is mainly used for drilling medium-strength and strong materials.

A feature of medium-strength and strong materials is that their destruction as a result of impacts occurs stepwise, as can be seen from Figure 2, which shows a graph of the dependence of the change in the resistance force of the destroyed material F_p during impact penetration into granite chiseled crown, obtained by R. Simon [4].

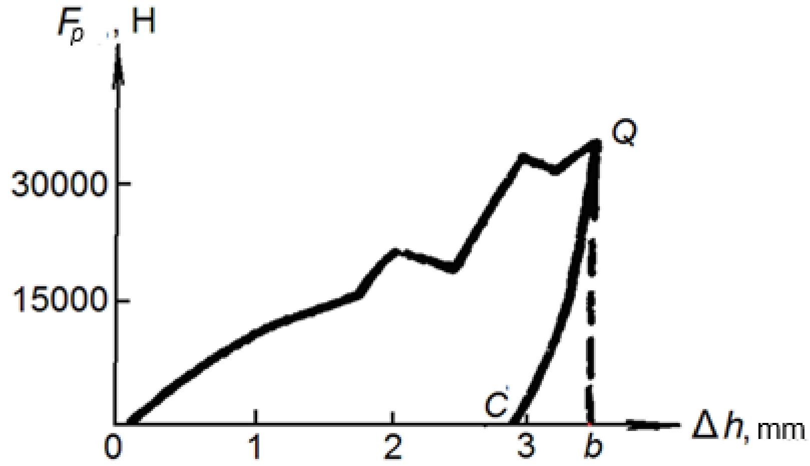


Figure 2 – Graph of impact penetration of a chiseled crown into granite

This can also be seen from Figure 3, which shows a graph of the dependence of the change in the resistance force of the destroyed material upon impact penetration of a four-point crown into granite, obtained by V.A. Khustrulid [4] and Figure 4, which shows graphs of the dependence of the magnitude of the drill bit penetration on pulses of various amplitudes F, which was achieved by measuring the impact velocity of a hammer on a tool [4].

Graph 1 shown in Figure 4 corresponds to a smaller pulse - its characteristic is close to triangular.

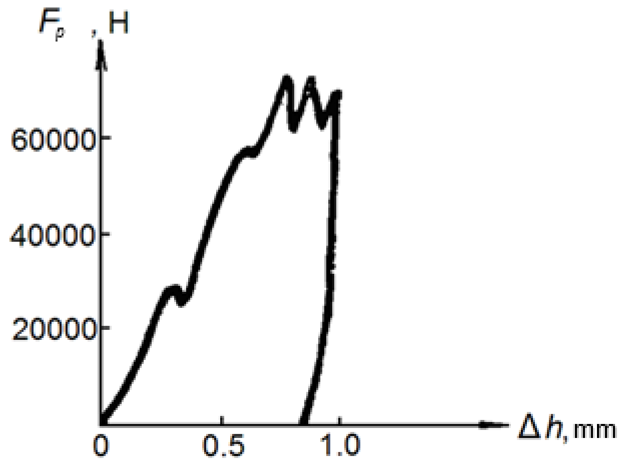


Figure 3 – Graph of impact penetration of a four-point crown into granite

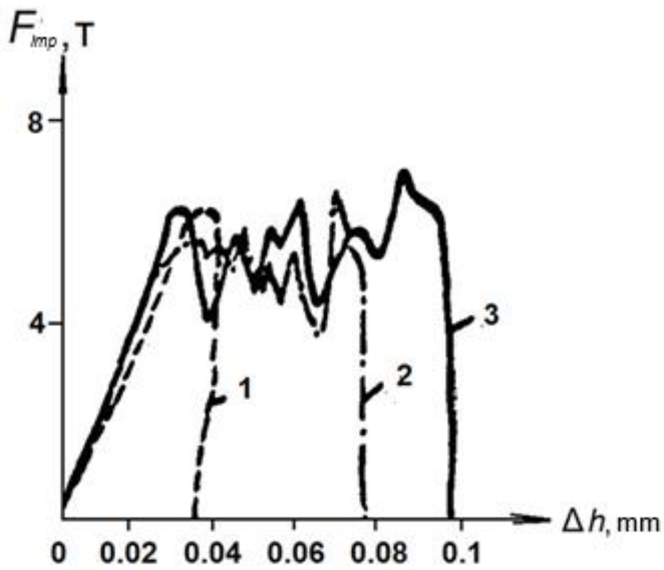


Figure 4 – Graphs of the dependence of the magnitude of the drill bit insertion on pulses of various amplitudes

At high pulses (graphs 2 and 3), the characteristics of the graphs are steeper, which means that there is no increase in the resistance forces of the material.

Figure 5 shows the graphs of the chisel crown embedding in quartzite. As can be seen from the graphs, when the force F is equal to some minimum values, significant rock destruction occurs.

As can be seen from the graphs in Figures 2-5, a feature of the process of drilling a drill bit into brittle rock is the presence of spikes in the penetration force, which are explained by the brittle puncturing of material particles.

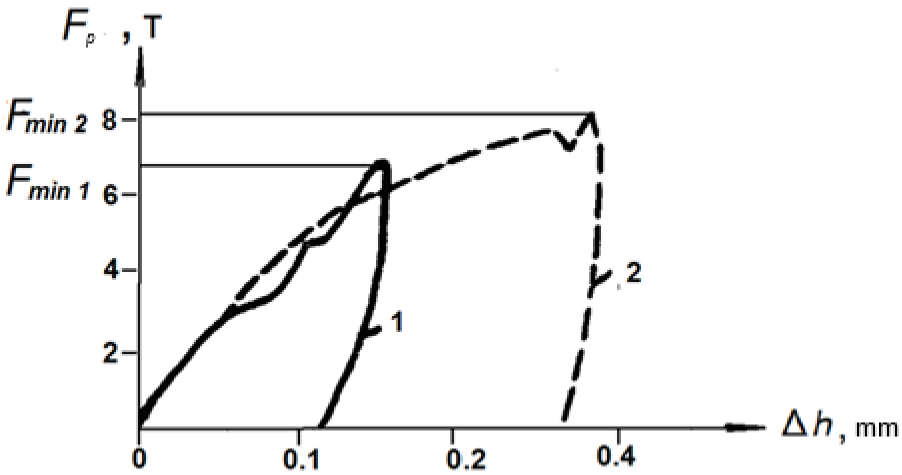


Figure 5 – Graphs of chiseled crown embedding in quartzite

They are characterized by an almost linear increase in the resistance of materials to penetration as the drill bit deepens, accompanied by crushing of rock grains under its blade, and the occurrence at a certain stage of brittle piercing of materials and the corresponding abrupt changes in the "pressure-penetration" characteristic.

When the maximum pressure in contact of the drill bit with the material is reached, the destruction of the material stops and partial elastic recovery of the rock occurs when the load is removed. As a result, the residual penetration depth is less than the maximum achieved penetration by the amount of rebound of the drill bit by the material.

It should be noted here that the above examples of the introduction of a drill bit into a material were carried out in laboratory conditions on samples of materials of certain sizes [9].

Taking into account the fact that the depth of the tool's penetration into the material being destroyed is influenced by such basic parameters as the impact force of the tool, the stiffness coefficient of the tool-material system, the duration of the impact pulse, the pulse duration coefficient and the area of the tool's impact on the material, in real conditions, the amount of the tool's penetration into the material Δh at a single impact can be approximately it is determined by the well-known formula:

$$\Delta h = \frac{f_s \cdot F_{in} \cdot \Delta t}{S}, \tag{3}$$

where F_{in} is the impact force of the tool, Δt is the duration of the shock pulse, and S is the area of impact of the drill bit on the material.

Hence the speed of implementation of the tool:

$$\frac{dh}{dt} = \frac{f_s \cdot F_{in}}{S}. \tag{4}$$

Since the impact energy is

$$j_e = \Delta h \cdot F_{in}, \tag{5}$$

then formula (4) can be rewritten as:

$$\frac{dh}{dt} = j_e \frac{f_s}{\Delta h \cdot S} = j_h \psi \frac{f_s}{\Delta h \cdot S}, \tag{6}$$

where ψ is the energy transfer coefficient of the drilling rod and crown.

For effective destruction of the material, the following conditions must be met:

$$.\frac{F_{in}}{S} > (\frac{F_{in}}{S})_{min}. \tag{7}$$

However, it should be noted here that the rate of destruction of the material is determined precisely by the force F_{in} and at a certain impact area S of this force. It follows that when calculating the rate of tool penetration into the material being destroyed, formula (4) should be used. In this case, it is advisable to determine the effectiveness of the impact, which creates significant contact stresses that destroy the material, by the value of the deformed mass. [4, 5, 6, 7, 8, 9, 10], and with regard to drilling holes – by its volume.

Hence, the dependence of the drilling speed on the combined action of the parameters of the drill rod, the drill bit and the destroyed material can be expressed as:

$$\frac{dh}{dt} = \frac{4v_v}{\pi d_d^2}, \tag{8}$$

where v_v is the volume of the destroyed material, d_d is the diameter of the drilled hole.

3. Analysis of the dynamics of pulse transmission by drilling rods and drill bits

Effective destruction of the material as a result of impacts on it by the drill rod and the drill bit fixed on it as solids is possible only when they come into contact immediately before impact, which leads to high contact tension

in both the drill rod and the crown, as well as in the material.

To describe the processes of collision of solids, there are a number of theories based on various mathematical models of longitudinal collision of bodies: the theory of collision of absolutely solids, based on Newton's mechanics; the Hertz model, based on the fact that the main deformations are local, as well as the dependence of the contact force on the contact deformation upon impact. the same as in the case of static compression of bodies; an energy impact model based on the kinetic energy change theorem and assumptions about the nature of their deformation; a discrete impact model based on the idea of the interaction of elementary discrete masses connected by elastic elements; the Saint-Venant wave theory, which most adequately reflects the real dynamic processes in impacted bodies, and others [11, 12, 13].

The ultimate goal of these theories is to mathematically describe the transmission of deformations of solids as a result of impact.

The use of impact energy as a parameter of impact machines was associated with certain possibilities for measuring it, provided that known parameters of materials and manufacturing technologies were used for both the piston-striker and working tools (drilling rods). Standards usually set only indirect requirements for them: a resource of 1600 hours with an average failure time of 20 hours.

The process of transmitting shock pulses from the piston-striker to the drill rod with a crown to the destroyed material occurs as follows.

When the drill is in operation, its hammer piston-striker the end of the shank of the drill rod, which transmits a shock pulse to the drill bit, which transmits it to the material being destroyed.

To analyze the impact processes, we will consider the main parameters of drilling rods.

The drilling rods used in rotary drilling are hexagonal steel rods manufactured according to the relevant standards, equipped with drill bits.

Figure 6 shows a threaded drill rod.

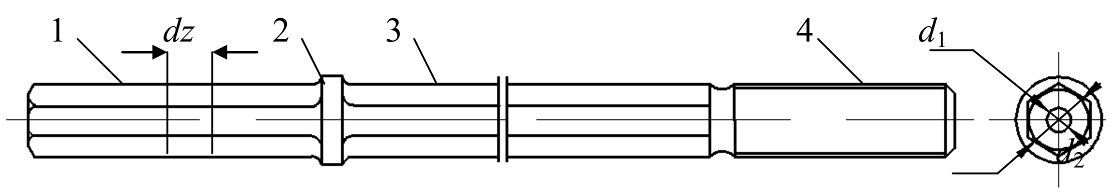
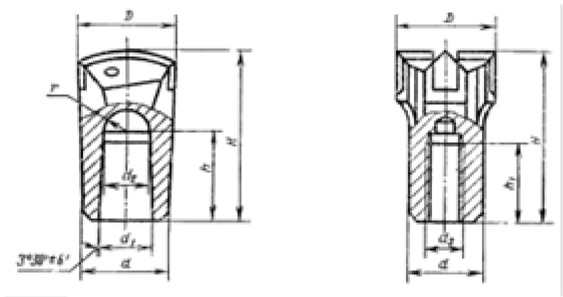


Figure 6 – Threaded drill rod

As can be seen from Figure 6, the drilling rod contains: a shank 1, a collar 2, a hexagonal rod 3 and a threaded part 4 at its end (there are also drilling rods with a conical end). Moreover, the plane of the end of the shank of the drilling rod, which is struck by a piston-strike, is a circle with a diameter of d_2 and a hole in the center with a diameter of d_1.

Figure 7 shows examples of drill bit designs: a) chiseled with a conical connection hole, b) cross–shaped with a threaded connection hole.



a) b)

Figure 7 – Examples of drill bit designs

For drilling holes, the shank 1 of the drill rod is installed in the drill bit and fixed in it by the collar 2. A drill bit is screwed onto the threaded part 4 of the drill rod. As can be seen from Figure 7, b), a threaded connection hole is provided in the drill bit for this purpose.

According to the results of research by various authors, it has been established that the duration of the shock pulse upon impact of the piston-striker on the drill rod is about 50 microseconds, and the velocity of passage of the shock pulse along the drill rod is about 5,300 m/s [14].

Hence, the duration of the passage of the shock pulse along the drilling rod with its length of 1 m is ≈ 200 microseconds.

It follows from this that at the moment of impact of the piston-striker on the drilling rod, the latter is stationary, since it rests against a rigid support by means of the drill bit - the destructible material, which will begin to experience the effects of a shock pulse with a length of the drilling rod of 1 meter only after 200 microseconds, and when using longer drilling rods even more.

When the piston-striker hits the end of the shank of the drilling rod, a contact tension σ_r occurs in the latter, determined by the expression:

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$$\sigma_r = \frac{F_p}{s_r}, \tag{9}$$

where s_r is the cross–sectional area of the drilling rod, F_p is the impact force of the piston impactor on the drilling rod.

The cross-sectional area of the end face of the drill rod shank is:

$$s_r = \frac{\pi(d_2^2 - d_1^2}{4}, \tag{10}$$

where d_2 is the diameter of the end face of the drill rod shank, d_1 is the diameter of the hole in the drill rod shank.

Since the shank of the drill rod in the end zone has a complex shape for the analytical determination of the amount of elastic deformation during impact: a hexagonal rod with a chamfer and a hole in the center, we introduce an additional parameter – the shape coefficient of the end of the shank of the drill rod, which is determined experimentally – k_{fr}.

From here, we determine the amount of elastic deformation of the drilling rod ε_r in the direction of the force vector F_p by the formula:

$$\varepsilon_r = k_{fr} \frac{\sigma_r}{E_r}, \tag{11}$$

For the piston-striker, the value of the contact tension σ_p will be

$$\sigma_p = \frac{F_p}{s_p}, \tag{12}$$

where s_p is the contact area of the piston- strike, equal to the contact area of the drill rod s_r.

The amount of elastic deformation of the piston- strike ε_p in the direction of the force vector F_p, taking into account the shape coefficient of the piston strikes k_{fp}, will be:

$$\varepsilon_p = k_{fp}\frac{\sigma_p}{E_p}, \tag{13}$$

where E_p is the modulus of elasticity of the steel of the piston impactor

The total value of the elastic deformation of the piston-striker and the drilling rod ε, which is the path of the piston-strike during impact with the stationary drilling rod, will be:

$$\varepsilon_\Sigma = \varepsilon_p + \varepsilon_r. \tag{14}$$

The final velocity of the during impact v_p is zero.

Hence, the change in the velocity of the piston-strike during impact will be: $\Delta v = v_{p\ max}$.

Assuming that during elastic deformation, in accordance with Hooke's law, the velocity of the piston impactor decreases uniformly, we determine the impact time.

To do this, in accordance with Hooke's law and Newton's 2nd law, we will create a system of equations:

$$\begin{cases} F_p = E_p + \varepsilon_\Sigma, \\ F_p = m_p a_p = m_p \dfrac{d^2_{\varepsilon_\Sigma}}{d\tau^2}. \end{cases} \tag{15}$$

Here: m_p is the mass of the piston- strike, a_p is the effective acceleration of the piston-striker, and τ is the duration of the impact.

As a result of solving the system of equations (15), the impact time τ_p will be

$$\tau_p = k_p \frac{m_p ln\varepsilon_\Sigma}{E_p}, \tag{16}$$

where k_p is a coefficient determined by the speed of the striking piston- strike before the impact.

The negative acceleration of the striking piston- strike ap during the impact will be:

$$a_p = \frac{v_{p\,max}}{\tau_p}. \tag{17}$$

Then the impact force F_p will be:

$$\begin{aligned} F_p &= m_p a_p = m_p \frac{v_{p\,max}}{\tau_p} = m_p \frac{v^2_{p\,max}}{2\varepsilon_\Sigma} = \frac{m_p v^2_{p\,max}}{2(\varepsilon_p + \varepsilon_r)} \\ &= \frac{m_p v^2_{p\,max}}{2(k_{fp} \frac{\sigma_p}{E_p} + k_{fr} \frac{\sigma_r}{E_r})}, \end{aligned} \tag{18}$$

where K_{fp} is the shape coefficient of the firing piston-striker, or

$$F_p = \frac{W_p}{k_{fp}\frac{\sigma_p}{E_p} + k_{fr}\frac{\sigma_r}{E_r}}, \tag{19}$$

where W_p is the energy of the striking piston-strike before the impact.

Therefore, the elastic tension of the drill rod σ_r as a result of the impact will be:

$$\sigma_r = \frac{E_r(W_p - F_p k_{fp}\frac{\sigma_p}{E_p}}{F_p k_{fp}}. \tag{20}$$

As shown by the research of many authors [14, 15], the shape of the tension pulse, and, accordingly, the forces formed in a fixed drill rod as a result of a single impact of a piston-striker transmitted along it, is as shown in Figure 8.

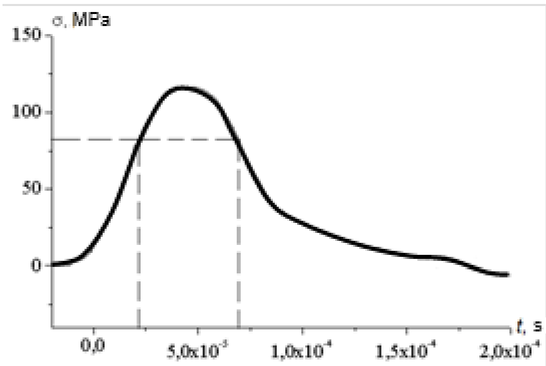

Figure 8 – Shock pulse oscillogram in the drill rod

As can be seen from Figure 8, the impact force pulse has a complex shape with a duration of 50 μs at a level of 0.7 of F_{max}.

This force pulse is transmitted to the material with negligible losses (≈ 3%) through the drilling bit mounted at the end of the drill rod, causing the material to experience appropriate contact stresses that lead to its destruction, as can be seen in Figure 1.

As follows from the above reasoning and formulas (16) and (17), the impact energy does not uniquely determine the effectiveness of material destruction, but depends on the contact stresses experienced by the piston-striker and the drill rod during impact, as well as on their elastic moduli.

Therefore, to solve this problem, it is advisable to include the impact coefficient Kud in the list of basic parameters of pneumatic perforators and other pneumatic impact machines. k_{if} is determined from the following expression:

$$k_{if} = k_{fp}\frac{\sigma_p}{E_p} + k_{fr}\frac{\sigma_r}{e_r}. \tag{21}$$

In this case, the formula for determining the impact force becomes:

$$F_p = \frac{W_p}{k_{if}}. \tag{22}$$

4. Analysis of the perforator's operation in the "operator – perforator – drill rod – drill bit – material to be destroyed" system

When conducting any research, including research on the operation of pneumatic impact machines, certain conditions are always specified under which the research is conducted, often involving certain ideal or boundary assumptions, such as the specific spatial position of the perforator body, its immobility during operation, etc. [3, 4, 15, 16, 17, 18, 19, 20, 21, 22].

Nevertheless, the results of such works are often widely used in research and design work in similar or analogous situations.

In particular, for the study of internal processes in impact machines, B. V. Sudnishnikov and his colleagues [20, 21, 22, 24] developed the method of indicator diagrams, which became the most objective way to determine the energy of an impact, providing a dependence of the coordinate and velocity of the piston-striker on time.

Using these data, the geometry of the perforator barrel and the dynamics of changes can be analyzed.

Figure 9 shows an idealized example of indicator diagrams [22], which depicts the cycles of the piston-striker movement, where: a – graph of the force acting on the striker, b – graph of the speed of the pistonstriker, c – graph of the movement of the striker; X3 – graph of the movement of the tool.

pressure of compressed air in its chambers.

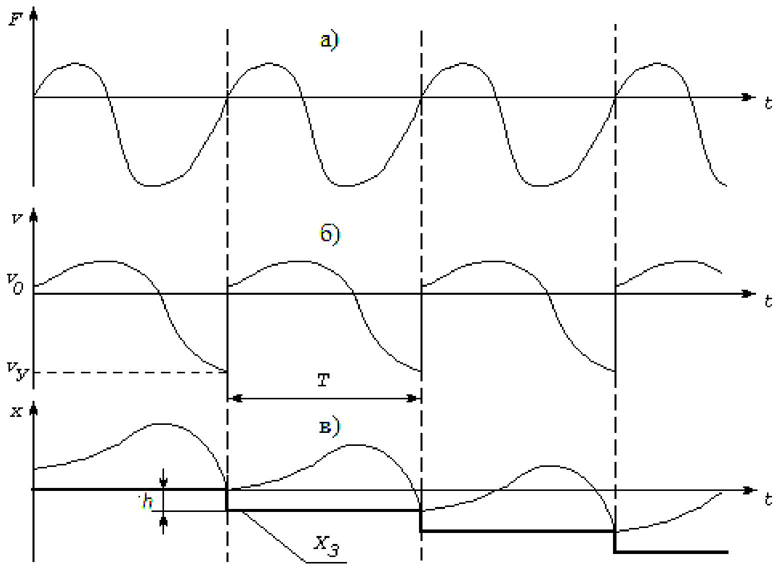

Figure 9 – Cycles of the piston-striker movement of the MOP-3 jackhammer

Figure 10 shows a real indicator chart for the MOP-3 jackhammer.

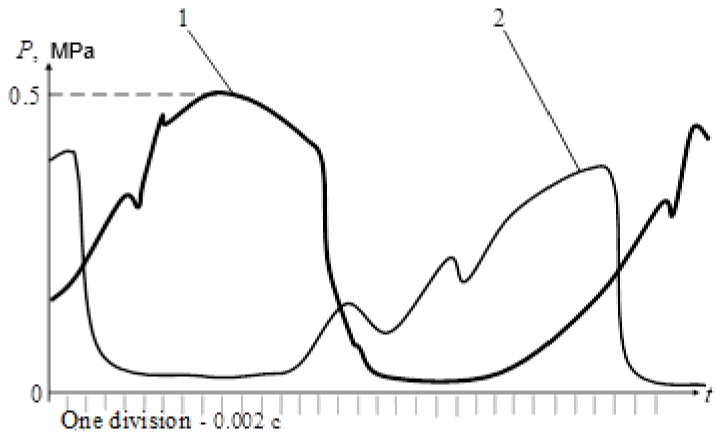

Figure 10 – Indicator chart of the MOP-3 jackhammer

1- forward stroke of the piston-striker (from left to right), 2 – reverse stroke of the piston-striker (from right to left)

As can be seen from Figure 10, during the working stroke of the firing piston, it is first subjected to a compressed air pressure of up to 0.5 MPa, which is released at the end of the working stroke. In turn, the return stroke of the firing piston occurs at a lower pressure due to the speed of its rebound from the tool after impact and the design features.

The use of the indicator chart method for analyzing work processes involves the following ideal research conditions [21, 22]:

1. The impact machine operates in a stable mode at constant pressure and temperature of the compressed air in the network;
2. Constant feed force;
3. Constant strength characteristics of the destroyed material;
4. Friction forces between the body and the striker are negligible in comparison with the pneumatic forces;
5. At each impact, the machine tool is embedded in the destroyed material by the same distance;
6. The feed force is sufficiently large and its sum with the projection of the machine weight on the ox axis provides the transfer of maximum impact energy.
7. At the moment of impact, the tool is stationary, and the machine body rests on its flange.

As can be seen from the list of conditions, these include such significant parameters as a large feed force, the immobility of the tool, and the support of the machine body on the tool flange.

These conditions correspond to the optimal operating modes of impact pneumatic machines, but they are not realized in the actual operation of the perforators, as the feed force in the optimal mode should be 1250 ÷ 1400 N, while the actual forces provided by the total weight of the perforator and the operator's feed force are 450 ÷ 550 N when drilling downward and the total average force of the pneumatic support and the operator's force is 600 ÷ 700 N when drilling horizontal and inclined drills.

Thus, this method can only be used to study the movement of the firing piston relative to the perforator barrel, but it cannot be used to study the transfer of impact energy in real-world conditions.

For this reason, in order to determine the actual impact energy of pneumatic perforators, the regulatory documents provide for testing the perforators on a drilling stand during drilling of horizontal holes when it is mounted on a pneumatic support, as shown in Figure 11, without the operator's effort.

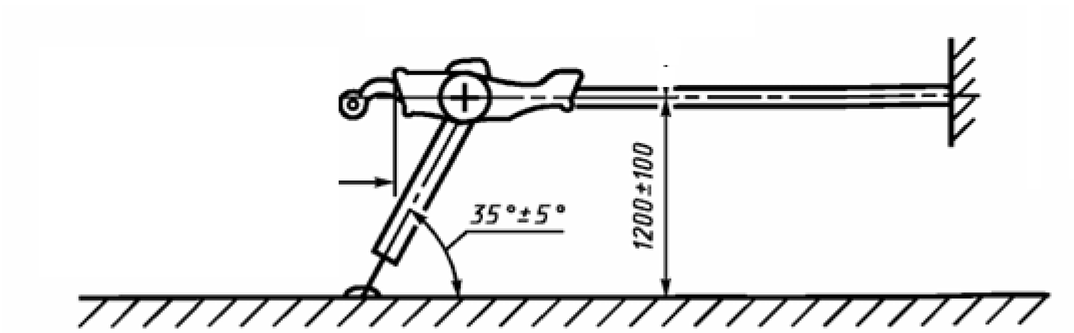

Figure 11 – Installation diagram of the perforator on the drilling stand

As can be seen from Figure 11, the angle of inclination of the pneumatic support that balances the punch is $\alpha = 35° \pm 5°$.

With a perforator weight of Q_{pr} = 350 N, the feed force Qτ provided by the pneumatic support will be:

$$Q_\tau = Q_{pr} \cdot \text{ctg}\alpha = 350 \cdot \text{ctg}35° = 350 \cdot 1.43 = 500 \text{ N}. \quad (23)$$

As can be seen from the above, the standard provides for testing the perforators under real-world conditions, with a certain average position of the pneumatic support used for drilling the holes, which is very far from the optimal position corresponding to the feed force of ≈ 1400 N, as mentioned above.

Thus, the existing theory of the operation of impact pneumatic machines, based on the ideal conditions listed above, does not reflect their operation in real conditions, and the impact pneumatic machines developed based on this theory cannot operate in an optimal mode.

The main reason for this is that the forces of compressed air acting on the piston-striker in the forward direction also act on the housing, causing it to move in the opposite direction, resulting in the working tool attached to the housing moving away from the material being destroyed when the piston-striker strikes it.

Figures 12 and 13 show graphs of the jackhammer body movements at feed forces of 160 N and 500 N [15].

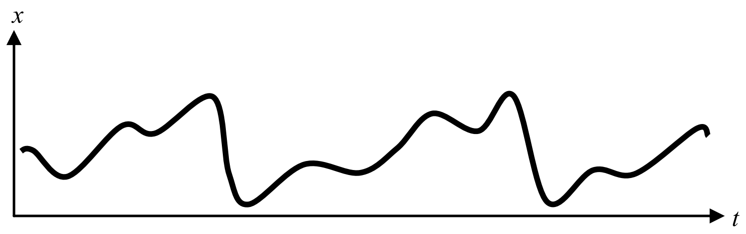

Figure 12 – Graph of the jackhammer body vibrations at a low feed force (160 N).

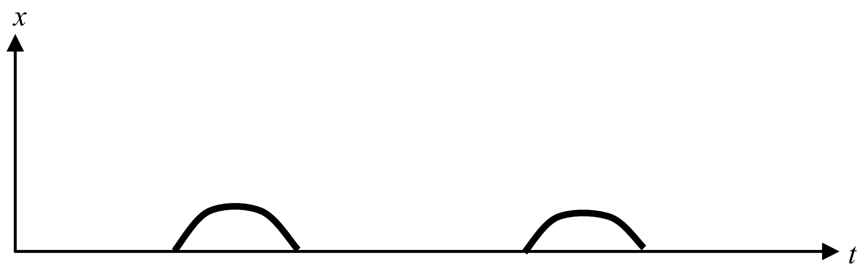

Figure 13 – Graph of the jackhammer body vibrations at a significant feed force (500 N)

As can be seen from the graphs in Figures 12 and 13, when the feed force is low (160 N), the vibration graph of the housing has a complex shape, reflecting the movement of the piston-percussion. However, when the feed force is high (500 N), the vibration level of the housing decreases significantly.

In the author's research on pneumatic drills during downward drilling, the amplitude of vertical vibrations of the housings was up to 15 mm when the feed force was 350 N.

At the same time, following the body moving upward under the influence of reactive forces, the drill rod also moved, losing contact with the material being destroyed.

At the same time, the drill rod moved up along with the body, losing contact with the material being destroyed.

Therefore, some of the piston-striker blows on the drill rod were wasted, resulting in a significantly lower drilling speed than the maximum. With a hammer drill's gravity force of 350 N, the driller must provide a feed force of 850-1050 N for efficient drilling, which significantly increases the likelihood of full-fledged impacts between the piston-striker and the drill rod, as well as between the drill rod and the material being drilled. However, this condition does not align with the operator's physiological capabilities.

To solve this problem, in contrast to B.V. Sudnishnikov's theory, which considered the body of a percussion machine as a stationary ideal case, the author proposed a theory in which the operation of a percussion machine, the working tool, and the material being destroyed is represented as a non-equilibrium thermodynamic system that functions relative to an inertial coordinate system.

Figure 14 shows a diagram of the functioning of such a system. In the impact mechanism under consideration, under the action of compressed air, the piston-striker and the body with the working tool fixed on it, in accordance with the 1st law of Newton, continuously perform reciprocating movements in opposite directions relative to the x coordinate with the fixed center of mass, designated as 0 on the x-axis. This occurs as a result of the continuous conversion of compressed air energy into kinetic energy.

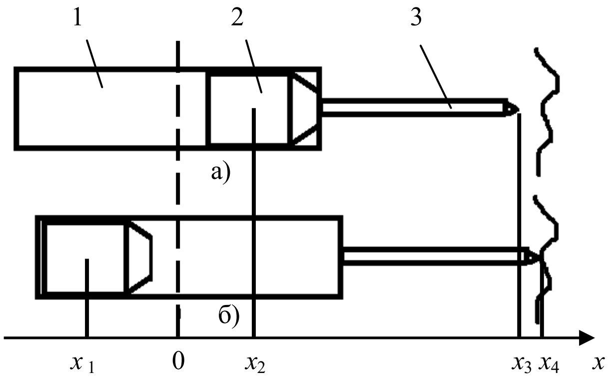

Figure 14 - Operation of the impact machine relative to an inertial coordinate system linked to the destroyed material

1 – housing, 2 – piston-striker, 3 – working tool.

As can be seen from Figure 14, a), when the piston-striker takes the extreme right position corresponding to the forward stroke – x2, the body with the working tool takes the extreme left position – x3, and when the piston-striker takes the extreme left position – x1 (Figure 14, b)) during the reverse stroke, the body with the working tool takes the extreme right position – x4. The amplitude of the piston-striker's reciprocating motion is determined by the design of the impact machine, while the amplitude of the body with the working tool's oscillation is determined by the ratio of their masses and the mass of the piston-striker.

The compressed air energy entering the machine undergoes two main conversion cycles:

1. Conversion into kinetic energy, which causes the body and the piston-striker to reciprocate.

2. Conversion of the kinetic energy into the energy of the impact impulse, which is transferred to the working tool.

In addition, some of the energy is lost to the atmosphere.

Since only a portion of the energy is transferred to the working tool, and the rest is spent on various inevitable losses, such as body motion, friction, compressed air release into the atmosphere, and more, the machine will always be in a non-equilibrium thermodynamic state, which can be described by the following equation:

$$\Delta j_{en} = j_{ea} - \mathrm{o}_{ei} - j_{el} \neq 0, \tag{24}$$

where ΔJ_{en} is the energy of the non-equilibrium thermodynamic state, J_{ea} is the energy of the compressed air, J_{ei} is the energy transferred to the working tool as a result of the impact, and J_{el} is the loss energy.

Under constant operating conditions of the machine: the pressure of compressed air in the network, temperature and other conditions, the frequency of movement of the piston-striker will also be constant. At the same time, its non-equilibrium thermodynamic state can be considered stable.

Considering the functioning of the working tool, on which periodic blows of the piston-striker are made, we note that after each blow in the working tool, as indicated earlier, longitudinal rapidly damped shock pulses of stress pass, after which the working tool returns to its initial state. This process means that the working tool, after each impact in which it gains energy and enters a non-equilibrium thermodynamic state, tends to reach thermodynamic equilibrium.

Thus, two thermodynamic systems operate during the operation of a pneumatic impact machine.:

- active, which is a shock mechanism in which the energy of compressed air is converted into kinetic energy, tending to a maximum stable nonequilibrium state;

- passive, which is a working tool striving for an equilibrium state [23].

Considering the perforator operation as an active type thermodynamic system, we note that the reciprocating movements of the piston-striker and the operation of the drilling rod rotation mechanism occur under the influence of compressed air energy. However, the compressed air energy also affects the perforator body, causing it to reciprocate in an opposite phase to the piston-striker movements. Therefore, during the perforator operation, both the piston-striker and the body are in an unbalanced thermodynamic state.

Since, according to B.V. Sudnishnikov's theory, the operation of pneumatic machines was considered without taking into account their thermodynamic state and always in the presence of a large feed force that provided ideal conditions for the immobility of the body, the pattern of operation of impact machines without a feed force has not been practically considered until now.

To determine the role of the body of an impact machine in its operation, the author studied the pattern of its movements in the absence of a feed force.

In the case of a horizontal position of the impact machine, during the working stroke, the piston-striker and the body produce opposite movements relative to the center of mass.

At the same time, the forces and time of the compressed air impact on the body and the piston-striker, in accordance with Newton's laws, are equal, and the momentum obtained during the forward stroke of the piston-striker is determined by the formula

$$I_{fs} = S_{re} \int_{0}^{T_{fs}} P_{fs} dt, \tag{31}$$

where I_{fs} is the impulse received by the firing piston during the forward stroke, S_{re} is the area of the rear end of the firing piston, P_{fs} is the average compressed air pressure during the forward stroke, and T_{fs} is the forward stroke time of the firing piston.
As a result of its forward stroke, the ram piston strikes the working tool.

At the same time, one part of the energy is transferred to the tool, and the other part is returned to the ram piston due to the elastic stresses during the impact.

Therefore, during the reverse stroke of the ram piston, its energy is determined not only by the action of compressed air, but also by the energy determined by the recovery coefficient.

At the same time, the parameters of the body's movement during the reverse stroke of the ram piston are determined by the compressed air pressure and the time of the reverse stroke.

As a result of its forward stroke, the ram piston strikes the working tool.

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Therefore, during the reverse stroke of the ram piston, its energy is determined not only by the action of compressed air, but also by the energy determined by the recovery coefficient.

At the same time, the parameters of the body's movement during the reverse stroke of the ram piston are determined by the compressed air pressure and the time of the reverse stroke.

The ratio of the body's displacement x_b and the piston-striker's displacement x_p is determined by the mass ratios:

$$\frac{x_b}{x_p} = \frac{m_p}{m_b}. \tag{32}$$

Let's consider the pattern of these movements during vertical drilling. Figure 15 shows a simplified diagram of the impact mechanism of a hammer drill.

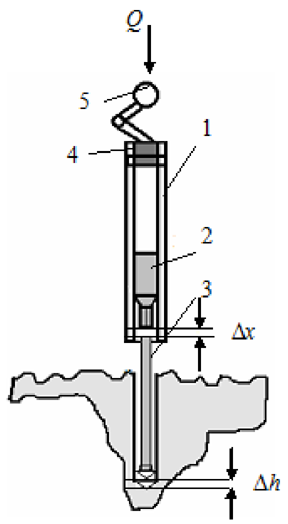

Figure 15 – Simplified diagram of the hammer drill's impact mechanism

As can be seen from Figure 15, the main elements of a pneumatic perforator are the body 1, the striking piston 2, the drilling rod with the drilling bit 3, the air distribution mechanism 4, and the vibration-damping handle 5.

During the operation of the perforator, when compressed air enters the rear chamber located above the piston-striker 2 in Figure 15, the piston-striker moves downward due to the force of compressed air pressure and its own gravity to deliver a blow. As mentioned above, in order to effectively break up the material, the drill rod with the drill bit must be in contact with the material before each impact.

Consider the dynamics of the perforator movement with a drill rod and a drill bit in the process of its operation.

Figure 16 shows the position of the perforator body 1, the drill rod 2 and the piston-striker 3 at the moment before the impact.

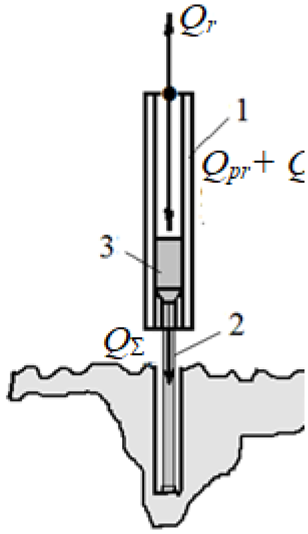

Figure 16 – Position of the perforator body, drill rod with drill bit, and impact piston-striker before the impact

As can be seen from Figure 16, several forces act on the perforator before the impact: Q_Σ is the total force that presses the perforator with the drill rod and drill bit against the rock, Q_{pr} is the weight of the perforator, Q_o is the operator's force, and Q_r is the reactive force acting on the body that occurs when the piston-percussion moves downward.

Since the piston-striker develops maximum speed before impact, the speed of movement of the perforator body in the opposite direction also becomes maximum.

It follows that in order to ensure contacts of the perforator body with the flange of the drill rod, and the drill rod with the rock, the total feed force of the perforator Q_Σ must meet the following conditions:

$$Q_\Sigma = Q_{pr} + Q_o \geq Q_r \quad (33)$$

After the impact of the piston-striker on the drilling rod, it bounces upward under the action of elastic forces.

At the same time, using the air distribution mechanism of the punch, the direction of compressed air supply is switched, and it begins to flow into the front chamber located under the firing piston-striker, as a result of which the firing piston-striker continues to move upward.

When the piston-striker reaches the upper end position, the direction of compressed air supply switches back to the opposite and the processes are repeated.

When the piston-striker with a mass of m_p moves downward under the action of compressed air and gravity of the piston-striker, its kinetic energy increases, taking the maximum value of j_{kel}. in the lowest position at which its velocity reaches the value of $v_{p.max}$.

Its kinetic energy is determined from the equation:

$$j_{kel} = \frac{m_p v_{p.max}^2}{2}. \quad (34)$$

Based on the fact that the forces of compressed air pressure in the cylinder act on the piston-striker and the back wall of the perforator simultaneously and equally, we determine the amplitude of vibrations of the perforator body.

The force acting on the body of the puncher upwards is equal to Q_r, the mass of the perforator with accessories is m_{pr},

The period of strokes of the T_s is:

$$T_s = \frac{1}{n}, \tag{35}$$

where n is the frequency of strokes.

Since the forces acting on the piston impactor during forward and reverse travel are different, the forward and reverse travel times are also different.

In this regard, we introduce the coefficient η, which characterizes the ratio of the forward stroke time of the piston impactor t_{fs}. for the entire period of his movements equal to the period of T_s strokes:

$$\eta = \frac{t_{fs}}{T_s}. \tag{36}$$

Then the forward stroke time of the firing piston will be:

$$t_{fs} = \frac{\eta}{n} \tag{37}$$

Since t_f is the time during which the perforator tends upward, we determine the amount of its movement.

According to Newton's 2nd law

$$Q_r = m_{pr} a_{pr}, \tag{38}$$

where a_{pr} is the average effective acceleration of the hammer body under the action of Q_r force.

The final speed of the $v_{pr.max}$ hammer body when moving upwards is:

$$v_{pr.max} = a_{pr} \cdot t_f. \tag{39}$$

Hence, the amplitude of the movements of the perforator body x_{pr} will be:

$$x_{pr} = \frac{Q_r}{2m_{pr}} t_f^2. \tag{40}$$

As can be seen from formula 40, the path of the x_{pr} body is inversely proportional to its mass m_{pr}. Since the mass ratio of the m_{pr} body and the m_p piston in existing designs of pneumatic impact machines is about 8:1, the pattern of movements of the hammer body does not play a significant role in the working process of reciprocating movements of the piston-striker.

As it moves upward, the perforator loses contact with the drilling rod, which experiences damping vibrations after being hit by the piston-striker, as a result of which it is likely to lose contact with the destructible material at the moment of impact.

Thus, when the hammer piston moves downwards - during its working stroke, the hammer body must be held in the lower position to ensure its contact with the collar of the drill rod and the contact of the latter with the destroyed material and to perform a full impact on it with the hammer piston.

Considering the effect on the movement of the hammer body of the piston-striker on the drill rod, we note the following.

The impact frequency of the piston-striker on the drilling rod n is 38 Hz;

Hence, the period between T_s hits is:

$$T_s = \frac{1}{n} = \frac{1}{38} = 26 \cdot 10^{-3} s. \tag{41}$$

As mentioned above, the velocity of propagation of shock pulses in a steel drill rod corresponds to the speed of sound ~ 5200 m/s.

Hence, the time of passage of the shock pulse along the drilling rod, depending on its length, is:

- 1 m – $171 \cdot 10^{-6}$ s;
- 2 m – $342 \cdot 10^{-6}$ s;
- 4 m – $684 \cdot 10^{-6}$ s.

The duration of the shock pulse passing through the drill rod is ~ 50×10-6 s, as can be seen from Figure 8 above.

Thus, the duration of the shock pulse passing through the drill rod and the duration of the shock pulse itself are ten times shorter than the period of impacts on the drill rod. However, there are damped resonant vibrations of the drill rod associated with the elastic properties of its material.

It follows that the influence of these processes on the movements of the hammer body can be very significant.

The application of the necessary force to hold the perforator in the lower position, in which it is pressed against the collar of the drill rod, is not accompanied by work (and energy consumption), since the path of the perforator is close to zero, but it ensures the necessary technological regime for an efficient drilling process.

In an idealized form, the equation of motion of the hammer piston, taking into account the above conditions, has the form:

$$m_p \frac{d^2 x}{dt^2} = F(t), \tag{42}$$

where x is the distance from the hammer piston to the drill rod, F is the force acting on the impactor piston, m_p is the mass of the piston-striker.

Here

$$F(t) = F_a(t) - m_p g, \tag{43}$$

where F_a is the force of compressed air pressure, g is the acceleration of gravity.

The equation of the compressed air force function $F_a(t)$ in relation to serial punchers, taking into account the actual area of its impact, has the form:

$$F_a(t) = P_{fc}(t) \cdot \left(S_{af} - S_{ta}\right) - P_r(t) \cdot (S_{re} - S_{ta}), \tag{44}$$

where $P_{fc}(t)$ is the pressure of compressed air in the front chamber, $P_r(t)$ is the pressure of compressed air in the rear chamber, S_{re} is the area of the end face of the rear part of the piston impactor, S_{fc} is the area of the end face of the front part of the piston-striker, S_{ta} is the area of the hole in the piston-striker under the water or air tube.

The average piston-striker speed v_{pa}, taking into account the initial rebound speed after hitting the drill rod, will be:

$$v_{pa} = \frac{dx_p}{dt} = v_{rb} + \frac{1}{T_s}\int_0^t F_a(t)dt, \tag{45}$$

where v_{rb} is the rebound velocity at the beginning of the return stroke of the piston-striker, T_s is the period between impacts.

Hence, the energy of the reciprocating motion of the body will be:

$$j_{pr} = \frac{v_p^2 m_p^2}{2m_{pr}}. \tag{46}$$

This energy is partially transferred to the drill handles, and to a greater extent to the impacts of the body on the collar of the drill rod when the piston-striker moves in the direction opposite to the drilling direction.

Thus, when drilling boreholes with insufficient feed force, the impact energy from the piston-striker to the drilling rod and further to the rock being destroyed is transmitted only partially and depends on the magnitude of this force.

Hence, it is obvious that in the applied schemes of impact machines, the movements of the body, as an intermediate element between the handle and the working tool, directed in the opposite direction relative to the movement of the piston-striker, prevent the working tool from striking the destroyed material.

To solve this problem, the author's analysis of kinematic relationships in pneumatic impact machines has shown that, since the housing and the piston impactor are not kinematically connected axially, it is possible to ensure a low feed force if their design conforms to the following principle:

For effective transmission of impacts of the piston-striker to the working tool, the presence of tool contact with the destroyed material and the corresponding velocity of the impactor piston before impact relative to the tool (destroyed material) are necessary and sufficient. In this case, the machine body and the working tool should not have kinematic connections along their axis.

Based on the research results obtained, in order to ensure productive operation, low feed force and low vibration, the author proposed a new scheme for the mechanism of impact machines, in which the feed force is applied not to the machine body, but directly to the working tool, as can be seen from Figure 17.

Figure 17 shows: handle 1, housing 2, guides 3, tubes 4, piston-striker 5, working tool holder 6, working tool 7.

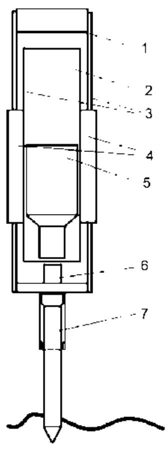

Figure 17 – Diagram of a percussive machine with a movable body

When this machine is operating, the piston-striker 5 makes reciprocating movements, striking the shank of the working tool 7. At the same time, the body 2 makes counter-phase movements, sliding along the guides 3 through the tubes 4. The handle 1, through the guides 3 and the working tool holder 6, presses the tool 7 against the material being broken. When the piston-striker 5 strikes the working tool 7, the latter moves forward in the cavity in the tool holder 6 without transferring the impact impulse to the handle. In this case, the operator only needs to move the working tool in the direction of the material to be destroyed for the next impact.

Thus, the theoretical analysis performed and the new scheme of operation of the impact machine proposed on its basis ensure its efficient operation without large feeding efforts and vibrations. Regardless of the scheme of operation of the pneumatic punch, the pattern of dependence of the drilling speed v_d on the feeding effort Q_t in general can be described by a 2nd order equation:

$$v_d = c_1 Q_f^2 + c_{2.} \tag{47}$$

From here, the dependence of the drilling speed v_d on the perforator parameters and the feed force Q_f is as follows:

$$v_d = \frac{dh}{dt} = \psi \cdot j_h \cdot n\left(c_1 Q_f^2 + c_2\right), \tag{48}$$

where j_h is the impact energy, ψ is the loss coefficient, n is the frequency of impacts, and c_1 and c_2 are the correction factors.

5. Analysis of the dynamics of mutual movements of the piston-striker and the perforator body in the absence of feed forces

As follows from the above materials, the mutual movements of the piston-striker and the body of the perforator have a significant effect on the speed of drilling the holes, as a result of which a significant feed force is required for effective operation.

At the same time, the body of the perforator in the process of its operation performs the functions of forming and discharging the pressures of compressed air by means of the corresponding air channels, directing its receipt relative to the piston-striker, determining the axial directions of the movements of the piston-striker, coupling with the handle by means of elastic elements and fixing the shank of the drill rod.

At the same time, the piston-striker, which moves in the housing, does not have any kinematic connections with other parts and assemblies of the perforator and moves along its axis and strikes the drill rod only under the influence of compressed air pressure.

Thus, the perforator housing does not have a direct effect on the energy of the piston-striker's impact on the drill rod, but by pressing it against the material being destroyed, it ensures the transfer of impact energy to the drill rod.

As mentioned above, when describing the operation of a pneumatic hammer drill, the force that presses the drill rod against the face of the rock is the weight of the hammer drill and the operator's force when drilling downward, as well as the pneumatic support force and the driller's force when drilling horizontally or at an angle.

Thus, the perforator body performs only the functions of an intermediate link between the perforator handle and the drill rod, while actively preventing the full-fledged transfer of the energy of the drill rod and drill bit impacts to the rock being destroyed due to the reactive forces directed in the opposite direction to the directions of the impacts.

To determine the ways to resolve this contradiction, let us consider the pattern of mutual movements of the piston-striker and the perforator body in the absence of feed efforts.

The effect of the compressed air forces on the piston-striker and the perforator body are the components of its basic working process.

Since the result of the perforator work is the kinetic energy of the impact of the piston-striker on the drill rod, depending both on the mass of the piston-striker and on its final speed before the impact, consider the conditions under which this energy will be maximum.

To do this, determine the dependence of the impact energy on the mass and final speed before the impact.

The accelerations of the percussion piston-striker and the perforator body under the force F_p will be, respectively:

$$a_p = \frac{F_p}{m_p}, \tag{49}$$

$$a_{pr} = \frac{F_p}{m_{pr}} \tag{50}$$

where m_p is the mass of the piston-striker; m_{pr} is the mass of the housing, F_p is the force exerted by compressed air on the piston-striker and the housing.

The final velocities of the piston-striker v_p and the perforator housing v_{pr} over the time t_{Fp} will be:

$$v_p = a_p \cdot t_{F_p}, \tag{51}$$

$$v_{pr} = a_{pr} \cdot t_{F_p}, \tag{52}$$

where t_{Fp} is the time of force F_p's effect.

Taking into account (38) and (39), we write:

$$v_p = \frac{F_p}{m_p} t_{F_p}, \tag{53}$$

$$v_{pr} = \frac{F_{pr}}{m_p r} t_{F_p}, \tag{54}$$

We will determine the ratio of amplitudes of movements of the piston-striker s_p and the body of the perforator s_{pr}.

The value of sp will be:

$$s_p = \frac{a_p t_{Fp}^2}{2} \tag{55}$$

The value of s_{pr} will be:

$$s_{pr} = \frac{a_{pr} t_{Fp}^2}{2}. \tag{56}$$

Then

$$\frac{s_p}{s_{pr}} = \frac{m_{pr}}{m_p}. \tag{57}$$

From here

$$s_{pr} = \frac{m_p}{m_{pr}} = s_p k_m. \tag{58}$$

Here, k_m is the ratio of the mass of the piston to the mass of the perforator body.

As can be seen from equation (58), the amplitude of the perforator body's vibrations is proportional to k_m.

In existing perforator designs, as well as in other impact machines, $k_m \ll 1$.

However, as experimental studies have shown, the granular structure of the rock being broken leads to a large spread of interactions between the drill bits and the rock being broken during impacts. This results in frequent large bounces of the drill rods, which cause them to strike the bush of the perforator body with their flange, causing the perforator to bounce back a significant distance, during which the piston-strikes the drill rod ineffectively. As a result, the drilling speed is significantly lower than the maximum possible.

Based on the data obtained, the author developed a new system for pneumatic impact machines, in which the hammer body is movable to ensure a low level of vibration, and the working tool is pressed against the material being destroyed by the handle directly connected to it [25, 26]. Figure 18 shows one of the proposed systems, in which the body is used as the striking element.

Based on the data obtained, the author developed a new system for pneumatic impact machines, in which the hammer body is movable to ensure a low level of vibration, and the working tool is pressed against the material being destroyed by the handle directly connected to it [25, 26]. Figure 18 shows one of the proposed systems, in which the body is used as the striking element.

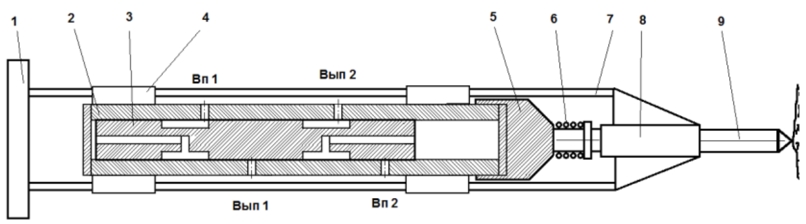

Figure 18 – Diagram of a pneumatic hammer with a body and a striker

As can be seen from Figure 18, the hammer contains a handle 1, a body 2, a piston 3, guides 4, a striker 5, a spring 6, a guide rod 7, a sleeve 8, and a working tool 9.

The operation of the hammer consists of two phases.

1st phase. The piston is in the leftmost position. The left channel of the piston is connected to the Bp1 hole, through which compressed air enters the left cavity. The right channel is connected to the outlet 2, through which the air is connected to the atmosphere. Under the influence of compressed air, the piston moves to the right position.

2nd phase. The piston is in the extreme right position. The right channel of the piston is connected to the outlet 2, through which compressed air enters the right cavity. The left channel is connected to the outlet 1, through which the air is connected to the atmosphere. Under the influence of compressed air, the piston moves to the left position.

Simultaneously with the piston movements, but in the opposite direction, the body-striker moves.

Consider the basic dependencies of the operation of a pneumatic impact machine. At the initial position:

$$F = P_{fs} \cdot S_{\text{re}}, \tag{59}$$

where F is the force acting on the piston, P_{fs} is the average acting pressure of the compressed air pressure, S_{re} is the diameter of the piston.

At the same time, the cylinder begins to move to the left, and the piston moves to the right relative to the center of mass. Their total path x_{Σ} is:

$$x_{\Sigma} = x_{max} - 2x_{ac}, \qquad 60)$$

where x_{max} is the maximum piston displacement in the housing, and x_{ac} is the amount of air cushions formed in the housing at the left and right extreme positions of the piston.

In this case, the path of each of them relative to the center of mass will be:

$$x_p = x_b = \frac{x_{max} - 2x_{ac}}{2}. \quad (61)$$

At a constant pressure of compressed air in the cylinder, the acceleration of the piston ap and the acceleration of the cylinder ac will be equal and opposite in direction:

$$a_p = \frac{F_p}{m_p} \quad (62)$$

$$a_b = \frac{F_p}{m_b}. \quad (63)$$

From here, the acceleration of the piston relative to the a_{pb} cylinder is

$$a_{pb} = 2a_p = 2a_b. \quad (64)$$

The path of the body will be:

$$x_b = x_{max} - 2x_{ac} \quad (65)$$

The final speed of the body $v_{b.max}$ during uniformly accelerated motion ab and the traveled distance will be determined from the formula:

$$v_{b.max} = \sqrt{2a_b \frac{x_{max} - 2x_{ac}}{2}} = \sqrt{a_b(x_{max} - 2x_{ac)}}. \quad (66)$$

From here, the kinetic energy of the body before the impact j_b will be determined as:

$$j_b = m\frac{a_b(x_{max} - 2x_{ac})}{2}. \tag{67}$$

Consider the case when the masses of the piston m_p and the housing m_b are different. Since the force acting on the housing and the piston F в and the time of action t_p are the same, we will determine the dependence of the ratio of their kinetic energies, respectively, j_b and j_p, on their masses.

Under the influence of the force F the acceleration of the piston will be:

$$a_p = \frac{F}{m_p} \tag{68}$$

and the acceleration of the body:

$$a_b = \frac{F}{m_b}. \qquad 69)$$

The acceleration of the piston relative to the body will be

$$a_{pb} = a_p + a_b. \tag{70}$$

Then the maximum speed of the piston relative to the center of mass will be:

$$v_{p.max} = a_p \cdot t_p, \tag{71}$$

and body:

$$v_b = a_b \cdot t_p \tag{72}$$

After some simple transformations, we get:

– for the piston energy:

$$j_p = \frac{m_p v_p^2}{2} = \frac{m_p}{2} \cdot (\frac{Ft_p}{m_p})^2 = \frac{F^2 t_p^2}{2m_p}, \tag{73}$$

– for the energy of the body:

$$j_b = \frac{m_b v_b^2}{2} = \frac{m_b}{2} \cdot (\frac{Ft_b}{m_b})^2 = \frac{F^2 t_b^2}{2m_b}. \tag{74}$$

In real pneumatic impact machines, the pressure of the compressed air entering the housing during the working stroke of the piston-striker, called the indicator pressure [22], changes as the piston moves. The pattern of this movement is determined by their design and technical features.

However, in this case, we consider the actual pressure of the compressed air, which determines the final velocity of the striker before the impact. Taking into account the losses during the impact of the striker housing on the working tool, the actual energy transferred to the blade of the working tool is:

$$j_{ei} = j_b \eta_j, \tag{75}$$

where η_j is a coefficient that takes into account the loss of impact energy during its transfer to the blade of the working tool. The obtained formulas allow us to determine the main parameters of pneumatic hammers of the "body-striker" type, depending on their purpose, required energy and frequency of impacts, mass, operating conditions, and various designs [24].

6. Analysis of the effect of feed force on the speed of drilling of blast holes

The typical graph of the dependence of the drilling speed on the feed force, obtained as a result of numerous experiments, has the form presented in Figure 19.

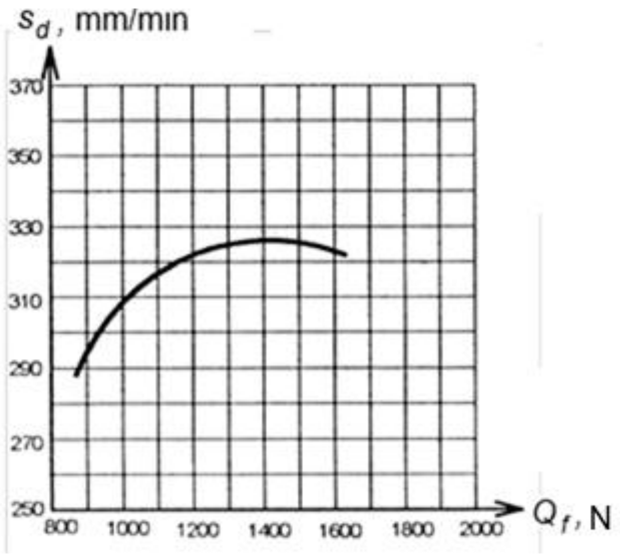

Figure 19 – Typical graph of drilling speed versus feed force

As can be seen from the graph, as the feed force Q_f increases to a certain value (1400 N on the graph), the drilling speed also increases according to a certain law.

If we consider that the energy of the piston-striker before the impact does not depend on the feed force, we can conclude that this pattern describes the probability of effective impacts of the piston-striker on the drill rod, where the drill bit is pressed against the material being drilled.

It should also be noted that during drilling, the drill rod rotates by a certain angle during the idle stroke.

If the feed force is too large and the rebound force of the drill bit from the material being destroyed is insufficient during the idle stroke, the drill bit remains pressed against the material being destroyed during the idle stroke, resulting in friction between the drill bit and the material being destroyed, which in turn leads to energy loss in the ram piston and a decrease in the drilling speed (when $Q_f > 1400$ N).

This relationship between the drilling speed s_d and the feed force Q_f can be described by a second-order equation in general:

$$s_d = aQ_f^2 + c, \tag{76)}$$

where a and c are correction factors.

From here, the dependence of the drilling speed s_d on the perforator parameters and the feed force Q_f is as follows:

$$s_d = \frac{dh}{dt} = \psi j_e \cdot n\left(aQ_f^2 + c\right), \tag{77}$$

Depending on the actual physical effort exerted by the operators during the drilling process, their performance varies significantly throughout the work shift. Consequently, the drilling speed also changes to a considerable extent. This occurs regardless of their level of adaptation to the working conditions [27, 28, 29].

Figure 20 shows a graph of the change in operator feed force during a work shift.

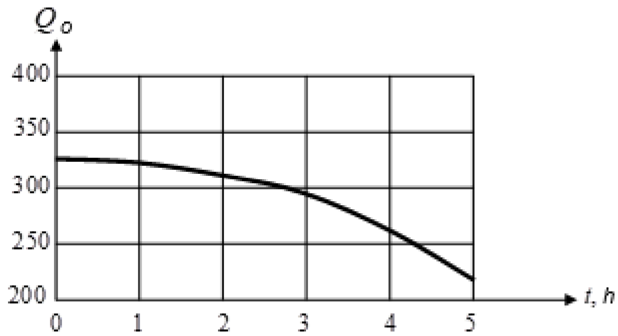

Figure 20 – Graph of the change in feed force of Qbur operators during a work shift

As can be seen from Figure 20, in general, the dependence of the operator's feed force on the drilling time during a work shift can be described by the following equation:

$$Q_o = Q_{el} \cdot k_Q \cdot e^{-k_R t}, \tag{78}$$

where Q_o is the drill driver's feed force, Q_{el} is the initial force, k_Q, k_R are correction factors obtained by experiment, t is the time of the drill driver's operation. Therefore, the real feed force Q_f applied to the perforators can be represented as a function

$$Q_f = Q_{pr} + Q_o = Q_{pr} + Q_{el} \cdot k_Q \cdot e^{-k_R t}, \tag{79}$$

where Q_{pr} is the feed force provided by the gravity of the perforator when drilling downwards or by pneumatic support when drilling horizontal and inclined holes. Therefore, the dependence of the drilling speed sd on the perforator parameters and the feed force Q_f is as follows:

$$\frac{dh}{dt} = j_p \cdot \psi \cdot n\left[c_1\left(Q_{pr} + Q_{el} \cdot k_Q \cdot e^{-k_R t}\right)^2 + c_2\right], \tag{80}$$

7. Analysis of the general pattern of the system functioning «operator - perforator - drilling rod - drilling crown – destroyed material»

As a result of the above analysis of the patterns of functioning of each of the links in the "driller – perforator – drilling rod – drilling crown – destroyed material" system, we will create a general formula for the dependence of drilling speed on the parameters of each of the links, which is its mathematical model:

$$\frac{dh}{dt} = \frac{F_{in}}{\Delta h f_s \cdot S} \cdot \psi \cdot \frac{4V_v}{\pi d_d^2} \cdot n\left[c_1\left(Q_t + Q_{el} k_Q e^{-k_R t}\right)^2 + c_2\right] \tag{81}$$

To simplify the problem, we introduce a generalized coefficient k_Σ:

$$k_{\Sigma} = \frac{F_p}{\Delta h f_s S} \cdot \psi \cdot \frac{4V_v}{\pi d_d^2} \cdot n. \qquad (82)$$

Then equation (85) becomes:

$$\frac{dh}{dt} = k_{\Sigma}[c_1(Q_t + Q_{el}k_Q e^{-k_R t})^2 + c_2]. \qquad (83)$$

After simple transformations, we get:

$$\frac{dh}{dt} = k_{\Sigma}c_1 Q_t^2 + 2k_{\Sigma}c_1 Q_t Q_{el} k_Q e^{-k_R t} + k_{\Sigma}c_1 Q_{el}^2 k_Q^2 e^{-k_R t} + k_{\Sigma}c_2. \qquad 84$$

To solve this equation, we integrate the left and right sides

$$\int dh = \int k_{\Sigma}c_1 Q_t^2 dt + \int 2k_{\Sigma}c_1 Q_t Q_{el} k_Q e^{-k_R t} dt + \int k_{\Sigma}c_1 Q_{el}^2 k_Q^2 e^{-2k_R t} dt + \int k_{\Sigma}c_2 dt. \qquad (85)$$

As a result of integration, we obtain a mathematical model of the drilling process using the "driller-perforator-drilling rod-drilling crown-destructible material" system:

$$h = k_{\Sigma}(c_1 Q_t^2 - 2c_1 Q_t Q_{el} k_Q \frac{e^{-k_R t}}{k_R} - c_1 k_Q Q_{el}^2 \frac{e^{-k_R t}}{2k_R} + c_2. \qquad (90)$$

As can be seen from equation (90), it takes into account the main conditions that affect the speed of drilling holes: the parameters of the perforators, drill rods, and bits, the hardness of the rock, the feed force, and many others.

As the author's research on the drilling of blast holes has shown, the specific operating conditions of pneumatic perforators in a particular location, such as their operation in a specific granite quarry for an extended period of time, are characterized by constant values of these parameters that are practically independent of time t.

Under these conditions, the role of the operator, who applies the feed force during the drilling of the holes, and who changes his physiological parameters significantly as a result of fatigue during one working shift, is very important.

Figure 21 shows the dependence of the drilling speed v of the "driller-perforator-drilling rod-drilling crown-rock" system on the feed force, where the actual feed force zone is shown, at which the pneumatic perforators operate.

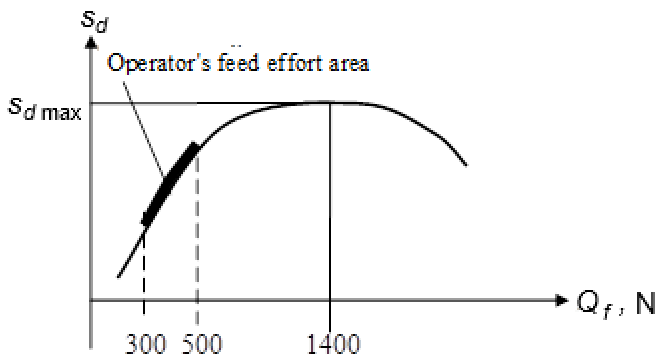

Figure 21 – Typical graph of the dependence of drilling speed s_d on drilling from the effort of the Q_f feed

As can be seen from Figure 21, the actual drilling speed of boreholes with portable pneumatic drills is significantly lower than the maximum.

At the same time, the feed force exerted by the operators significantly affects the drilling speed, which forces the operators to apply maximum effort to the perforator handles, which leads to their rapid fatigue.

Thus, the problem of efficient operation of portable pneumatic drills can be solved either by mechanizing the feed force while maintaining the weight of the drills and their properties as relatively lightweight and mobile devices for drilling holes, or by developing a different design of drills that do not require significant feed forces.

8. Conclusions

1. The theoretical analysis of the functioning of the system "operator – perforator – drilling rod – drilling crown – destroyed material" determined the influence of each of the links, including the operator, on the speed of drilling holes, as a result, a mathematical model of this system was compiled.

2. As a result of the analysis of the operation of existing designs of pneumatic machines of impact action, it was found that the reason for the need for a large feed force and high vibration of the handles is the use of the body as an intermediate link between the handle and the working tool.

3. As a result of analyzing the functioning of an impact machine as a thermodynamic system in which compressed air energy is converted into kinetic energy, it has been established that during operation, this system always tends to a stable non-equilibrium thermodynamic state.

. As a result of summarizing the research results of a pneumatic impact machine as a thermodynamic system, the dependence of their properties on the processes taking place in them has been established: when energy is converted, the systems tend to an unstable equilibrium state, and when energy is redistributed, the systems tend to equilibrium.

5. The main practical result of the conducted research is the provision of lighter pneumatic perforators with lightweight feed devices to ensure high drilling speed while maintaining mass, and ensuring that there are no kinematic connections between the handle, body, and drill rod, which ensures a low level of vibration.

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