https://doi.org/10.32056/KOMAG2022.4.3

Constructional changes of pneumopercussion machines for improving their efficiency

Received: 19.09.2022

Accepted: 29.12.2022

Published online: 30.12.2022

Author's affiliations and addresses:

[1] Institute of Geotechnical Mechanics of National Academy of Sciences of Ukraine, 2a Simferopilska str., 49000, Dnipro, Ukraine

[2] Dnipro University of Technology, 19 Yavornytskoho Ave., 49005, Dnipro, Ukraine

[3] Energy Economy Research Institute of the Polish Academy of Sciences, J. Wybickiego 7A, 31-261, Kraków, Poland

*** Correspondence:**

e-mail: arturdyczko@gmail.com

Volodymyr ANTONCHIK [1], Kostiantyn ZABOLOTNYI [2], Valentyn HANKEVICH [2], Vira MALTSEVA [1], Oleksandra KUTS [1], Artur DYCZKO [3*]

Abstract:

Pneumopercussion machines (pneumatic impact machines) are widely used in all areas of human activity. Also, they are widely used in the mining industry. Unfortunately, their operation is characterized by a low efficiency of compressed air energy usage. In some cases, this level of efficiency is calculated as 15-20%. Such a situation increases the cost of drilling operations significantly. In this article, due to an implementation of a new construction of the equipment, the efficiency of the pneumopercussion machines was increased. This problem is solved by combining the most effective thermodynamic processes of compressed air in the working chambers of machines. Also, a new technical solution for the construction of pneumopercussion machines is suggested by the Authors. The proposed new design is realized by a combination of the most effective thermodynamic processes in the chambers of pneumatic impact machines. A new pneumatic hammer is presented, which allows to reduce compressed air consumption twice during an operation on the surface (in comparison with hammers available on the market). The operation of pneumopercussion machines and the method of calculating geometric parameters are described. The economic performance of the equipment confirms the correctness of the proposed technological solutions.

Keywords: pneumopercussion machines, thermodynamic processes, hammer, compressed air

Publisher: **KOMAG Institute of Mining Technology, Poland**

1. Introduction

One of the most important indicators of the efficiency and operational cost of pneumatic hammers is a consumption of compressed air, the cost of which is up to 50% of the cost of a running metre of a well drilled with a pneumatic hammer. Their usage is caused by the need to introduce small mechanization in production. They are also quite often used in the mining industry [1]. Economical, technical and technological application of this equipment is indispensable in the complex use of mining equipment. Energy reserve as regards the parameters of parallel power active compensators is also very important for increasing the efficiency of the equipment operation in mining [2].

Available pneumopercussion machines, having a small, about 15-20%, efficiency, require a significant consumption of compressed air which increases the cost of drilling. One of the most promising ways to increase efficiency of pneumopercussion machines includes an improvement of their thermodynamic parameters. This can be done by increasing the useful work of thermodynamic processes and combining such processes into cycles [3]. The correct economic evaluation of the proposed technical and technological solutions, which are adopted in mining, is also important [4].

In available hammer designs compressed air is used twice for a full cycle of movement of the striker (hammer) - to accelerate the striker (hammer) at the stage of the working stroke, followed by the exhaust of compressed air into the atmosphere and to return the striker (hammer) to its original position, during the reverse stroke, with subsequent exhaust of compressed air into the atmosphere [5]. Thus, the full cycle of movement of the striker (hammer) of available pneumatic hammers consists of cycles of compressed air operation at the stage of isobaric and adiabatic expansion and acceleration of the striker (working stroke), followed by the exhaust of portions of compressed air into the atmosphere and cycles of isobaric and adiabatic expansion and acceleration of the striker in the opposite direction (reverse stroke) with the subsequent exhaust of portions of compressed air into the atmosphere [6]. Thermal processes, occurring during the operation of the equipment, can be presented using the Neumann principle [7]. This principle was also used for a determination of maximum stresses [8]. The required frequency of strikes of the striker, determined by the drilling process, does not enable a full expansion of compressed air to the atmospheric pressure in the known designs of pneumatic hammers due to a short period of acceleration of the striker. In this regard, a significant part of the compressed air, both during the working and reverse strokes of the striker, is emitted into the atmosphere after the striker reaches the required speed [9].

2. Methods

The essence of the method for increasing the thermodynamic efficiency of pneumopercussion machines is described below. The highest thermodynamic efficiency of pneumopercussion machines is achieved due to thermodynamic processes in their working chambers by combining them into one cycle. Full acceleration of the striker (at the stage of the working stroke) is carried out only by an isobaric expansion of compressed air from the high-pressure line without exhausting any part of it into the atmosphere [10]. After the striker strikes the drilling bit, pressurized air should be supplied to the reverse stroke chamber and the striker should be accelerated to the required speed by an isobaric expansion in the opposite direction. After that, it is necessary to shut off the supply of compressed air to the reverse stroke chamber and continue moving the striker in the opposite direction by an adiabatic expansion of the compressed air remaining in the reverse stroke chamber. When the speed of the striker is sufficient to return it to its original position, the striker, moving by inertia, opens the exhaust holes and a portion of compressed air from the return chamber is released into the atmosphere [11]. Thus, the full cycle of operation of the pneumatic hammer can be carried out with two isobaric and one adiabatic expansion of compressed air, followed by one exhaust of its portion into the atmosphere.

The pneumatic hammer design that implements the specified cycle of thermodynamic processes must have the following design features:

– the compartment for the translational movement of the accelerator in the high-pressure chamber, where an isobaric expansion of compressed air occurs at the stage of the striker's working stroke;

– the compartment for the translational movement of the striker in the reverse stroke chamber, in which first isobaric, and then - adiabatic expansion of the portion of compressed air occur at the striker's reverse stroke stage;
– the striker movement section should be divided by the striker into two isolated chambers;
– the striker reverse stroke chamber and low-pressure chamber, which is permanently connected to the atmosphere by a through-passage channel;
– the accelerator must rigidly join with the striker during the working stroke of the striker to accelerate the striker together with the accelerator to the required speed, as well as at the stage of the striker's reverse stroke to decelerate it after the striker reaches the required reverse stroke speed and until the striker with the accelerator stop completely.

3. Results

3.1. The pneumatic hammer design that implements a new way of operation

The design of the DTH hammer (down-the-hole hammer) for an implementation of the specified method of operation is shown in Fig. 1.

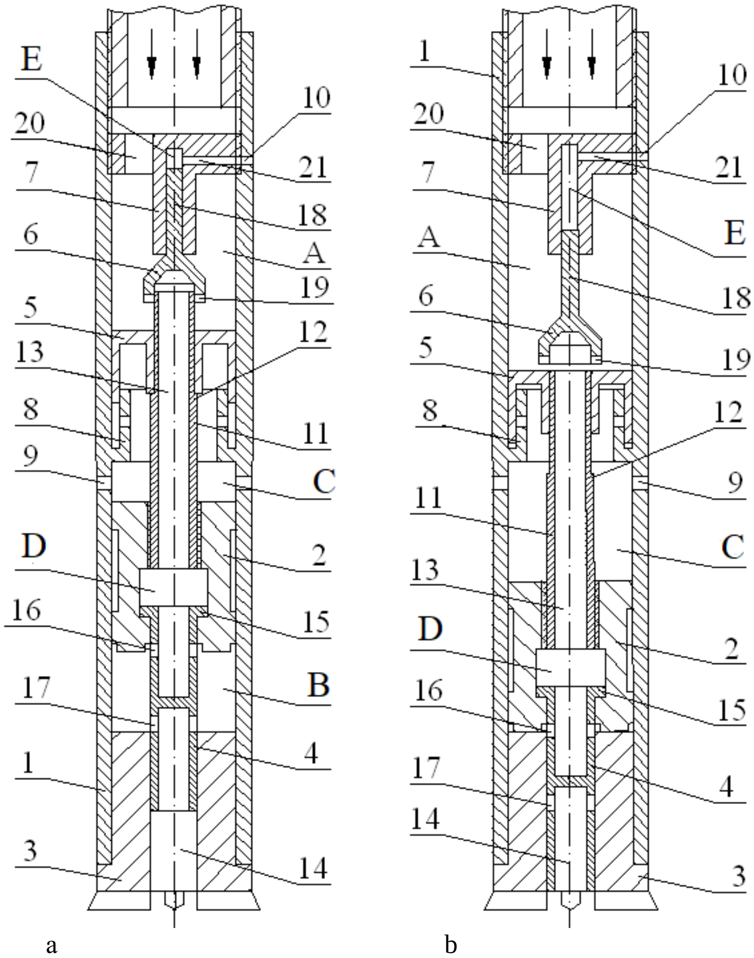

Fig. 1. DTH pneumatic hammer: a – original position, b – striker at the impact moment

The DTH hammer consists of body 1, striker 2, drill bit 3, bushing 4, accelerator 5, valve 6, valve stop 7, accelerator stop 8. Body 1 has discharge holes 9, 10. Striker 2 has a shank 11 rigidly connected to it with an annular shoulder 12 and the axial bore of the shank 13. The drill bit 3 has an axial bore of the bit 14. The bushing 4 has an annular ledge 15, inlet holes 16 and outlet holes 17. The valve 6 has a stem 18 and a cylindrical ledge with radial slots 19. The valve stop 7 has feed holes 20 and discharge channel 21.

Publisher: **KOMAG Institute of Mining Technology, Poland**

The hammer has a high-pressure chamber A, a reverse stroke chamber B, and a low-pressure chamber C. Striker 2 has a cavity D. Valve stop 7 has a cavity E.

3.2. The DTH hammer operation

The hammer works in the following manner. In the original position, the shank of the striker 11 with an annular ledge 12 butts with the accelerator 5 and the valve 6, which closes the axial bore of the shank 13 and the supply of compressed air to the reverse stroke chamber B (Fig. 1a). Bushing 4 is in the uppermost position. After supplying compressed air to the hammer through the supply holes 20 at the stop of the valve 7, due to the difference in air pressure between the high-pressure chamber A and chambers B and C (where the pressure is atmospheric, since they are connected to the atmosphere by the outlet holes 17, the axial bore of the bit 14 and hole 9), the compressed air in chamber A expands and accelerates the accelerator 5 (which abuts against the annular ledge 12 of the shank 11) together with the striker 2, the shank 11 and the valve 6 in the direction of chambers B and C, making a working stroke. Air from chamber B through the outlet holes 17 of the bushing 4 and the axial bore of the bit 14 is exhausted into the atmosphere. During the acceleration of the striker 2, the bushing 4 remains in the extreme upper position when the outlet holes 17 remain open and connected to the atmosphere (Fig. 1a). The distance between the accelerator 5 and the stop of the accelerator 8 is calculated so that the accelerator 5 stops at the stop of the accelerator 8 at the moment when the striker 2 has received the required speed and impact energy (Fig. 1b). Further, the striker 2 together with the valve 6 move until the valve 6 collides with the accelerator 5. After that, the valve 6 stops, and the striker 2, having passed another 3 mm, collides with the drill bit 3 (Fig. 1b). At the moment of separation of the valve 6 from the shank 11, the striker 2 with the end wall of the cavity D pushes the bushing 4, which, having received an impulse, moves in the outlet channel 14 of the drill bit 3, blocking the outlet holes 17 and subsequently the air outlet from the reverse chamber B. After separating the valve 6 from the shank 11, compressed air from the high pressure chamber A through the cylindrical ledge with radial slots 19 and the axial bore 13, the shank 11 enters the cavity of the striker D and presses the annular ledge 15 of the bushing 4 to the wall of the cavity D of the striker 2, opening the inlet holes 16 of the bushing 4, and leaves it in this ledge at the moment the striker 2 hits the drill bit 3. At the moment of impact of the striker 2 on the drill bit 3, compressed air from the high-pressure chamber A through the axial bore of the shank 13, the cavity of the striker D and the inlet holes 16 of the bushing 4 enters the reverse stroke chamber B, where the air pressure increases to a high pressure equal to the pressure in the line compressed air supply [9].

Due to the difference in air pressures in chambers B and C (where the air pressure is atmospheric), an unbalanced force appears from the side of the reverse chamber B, which accelerates the striker 2 in the opposite direction. Valve 6 after collision with accelerator 5 and separation from the shank 11 due to the difference in air pressures in the high-pressure chamber A and the cavity E in the stop of the valve 7 (where the air pressure is atmospheric, since the cavity E is constantly connected to the atmosphere by the discharge channel 21 and the discharge hole 10), starts to move in the opposite direction, compressing the air in cavity E. The distance between the annular ledge 12 of the shank 11 of the striker 2 and the accelerator 5 is calculated so that they join at the moment when the striker 2 receives speed and kinetic energy sufficient to complete the reverse stroke. Further, the striker 2 with the accelerator 5 (which move faster than the valve 6) catch up with the valve 6 and the shank 11 joins the valve 6, blocking the flow of compressed air into the reverse chamber B. Further movement of striker 2 with shank 11, accelerator 5 and valve 6 occurs by inertia and under the action of adiabatic expansion of a portion of compressed air in chamber B. On the other side, in the high-pressure chamber A, the accelerator 5 and valve 6 are acted upon by the high-pressure force of compressed air, which breaks the accelerator 5 together with the striker 2, the shank 11 and the valve 6 (Fig. 1 a).

At the stage of the reverse stroke, the striker 2 pulls the bushing 4 by the annular ledge 15 and at the end of the reverse stroke, the bushing 4, leaving the axial bore of the bit 14, opens the outlet holes 17 through which the compressed air from chamber B is exhausted into the atmosphere. The kinetic energy of striker 2, shank 11 with accelerator 5 and valve 6, together with the total action of compressed air pressure forces on them in chambers A and B, are calculated so that at the end of the reverse stroke, striker 2 with shank 11, accelerator 5 and valve 6 return to the original position considering the exhaust of compressed air from chamber B into the atmosphere. After that, the cycle of the working and reverse stroke is repeated.

3.3. Calculation of the geometric parameters of the pneumatic hammer

The calculation of the geometric dimensions of the hammer is made by successive solving of the equations of motion of the striker 2 and the accelerator 5 and the valve 6 at different stages of the thermodynamic cycles of the compressed air expansion. Since the time of the working cycle of the striker (working and reverse stroke of the striker) is approximately 0.03-0.05 s, the energy losses of the compressed air due to the transfer of heat to the environment can be neglected due to their small value, therefore the process of expansion of the portion of compressed air in the reverse stroke chamber B of the hammer can be considered adiabatic [12]. Small volumes of the hammer chambers in comparison with the volume of the compressed air supply line, as well as the speed of propagation of compressed air (in the order of the speed of sound) in the hammer cavity allows the process of striker acceleration with the expansion of compressed air permanently supplied from the line to be considered isobaric [13].

The diameter of the hammer and the dimensions of the cross-sections of its parts, bores, cavities and chambers are set constructively, based on the diameter of the drilled hole. The linear dimensions of the accelerator 5 are also set constructively. The linear dimensions of the body 1, striker 2, bit 3 and bushing 4 and valve 6 are specified after determining the working and reverse stroke of striker 2, accelerator 5 and valve 6.

An acceleration of striker 2 with accelerator 5 and valve 6 to a given speed and impact energy from the initial position at the stage of isobaric expansion of compressed air is determined as follows:

$$K_{\mathrm{c}} = \int_0^V p_m\,(V)dV \tag{1}$$

$$K_c = K_y + K_b + K_k \tag{2}$$

where:

K_{c} - the total kinetic energy of striker 2, accelerator 5 and valve 6 at the end of striker acceleration, J;

K_k - kinetic energy of the valve at the end of the acceleration of the striker, J;

K_y - kinetic energy of the accelerator at the end of the acceleration of the striker, J;

K_b - kinetic energy of the striker 2 required to hit the drill bit and destroy the rocks, J. This value is set depending on the hardness of the rocks. Knowing the mass of the striker from the formula: $K_b = \frac{m_b \vartheta_b{}^2}{2}$;

where: m_b - weight of the striker, kg; ϑ_b- the speed of the striker at the moment of its impact on the drill bit.

The kinetic energies of the accelerator 5 and the valve 6 are determined by the speed of the striker with which they have the same speed. The masses of the accelerator 5 and the valve 6 are determined constructively based on the dimensions and materials of these parts.

Considering that in the isobaric process $p_m = const$, as well as for the cylindrical shape of the impactor chambers $V = S \cdot X$, we obtain:

$$K_{\mathrm{c}} = p_m \int_0^{V_1} dV = p_m(V - 0) = p_m \cdot S_y \cdot X_P \tag{3}$$

$$X_{\mathrm{p}} = \frac{K_{\mathrm{c}}}{p_m \cdot S_y} \tag{4}$$

where:

p_m - pressure of supplied compressed air, Pa;

V - volume of compressed air in the thermodynamic cycle of its expansion, m^3;

S_y - the cross-sectional area of the accelerator, m^2;

X_P - the path traveled by the striker together with the accelerator and valve, m.

Publisher: **KOMAG Institute of Mining Technology, Poland**

A determination of the return stroke of the striker in an isobaric expansion of compressed air in chamber B of the return stroke is determined by the formula:

$$K_0 = \int_0^{V_1} p_m\,(V)dV \tag{5}$$

$$K_0 = p_m \int_0^{V} dV = p_m(V-0) = p_m \cdot S_b \cdot X_1 \tag{6}$$

where:

K_0 - the kinetic energy of the striker at the end of the section of isobaric expansion of compressed air, J;

S_b - the area of the butt end of the striker in the return chamber B, m^2;

X_1 - the return path of the striker in isobaric expansion of compressed air in the return chamber B, m.

A determination of the reverse stroke of the striker with accelerator 5 and valve 6 in adiabatic expansion of compressed air in the reverse stroke chamber B.

From the moment valve 6 docks with the striker shank, the compressed air supply to the return chamber B stops and the striker 2 with accelerator 5 and valve 6 moves under the action of the following forces: from the side of chamber B - the force of adiabatic expansion of compressed air in chamber B and the inertia force that the striker 2 with accelerator 5 acquired at the stage of their acceleration due to the isobaric expansion of compressed air in chamber B, and on the other hand, the pressure force of compressed air in the line and energy incoming air flow from the high pressure line, which was formed as a result of the consumption of compressed air to fill the return chamber B.

The movement of the striker with the accelerator at this stage is described by the following equation:

$$A_a + K_o = A_m + K_g \tag{7}$$

where:

A_a - work of the adiabatic expansion of compressed air in chamber B of the reverse stroke.

K_o - kinetic energy of the striker and accelerator by the beginning of the adiabatic expansion cycle.

A_m - the work of pressure forces of compressed air in the line.

A_g - kinetic energy of the dynamic pressure of compressed air on the striker 2 and the accelerator 5.

Equation (1) can be rewritten as:

$$\int_{V_1}^{V_2} P_a\,(V)dV + \int_0^{V_1} P_m\,dV = \int_{V_1}^{V_2} P_m\,dV + \frac{\dot{m}\vartheta^2}{2}t_2 \tag{8}$$

where:

V_1 - volume of compressed air in chamber B at the beginning of the adiabatic expansion, m^3;

V_2 - volume of compressed air in chamber B by the end of the adiabatic expansion, m^3;

P_a - current value, pressure in chamber B during adiabatic expansion, Pa;

$\dot{m}$ - mass flow rate per second of compressed air, kg/s;

ϑ- compressed air flow rate, m/s;

t_2 - the time of the return stroke of the striker with the accelerator 5 and the valve 6 at the stage of the adiabatic expansion of the compressed air in the chamber B.

After integrating the terms of equation (8), we obtain:

$$\frac{P_1V_1}{k-1}\left[\frac{(V_2)^{k-1}-(V_1)^{k-1}}{(V_2)^{k-1}}\right] + P_mV_1 = P_m(V_2-V_1) + \frac{\dot{m}\vartheta^2}{2}t_2 \tag{9}$$

where: k - adiabatic exponent (dimensionless quantity).

For the adiabatic expansion of a gas (air), an equality is true, which is the Poisson equation.

$$P_1(V_1)^k = P_2(V_2)^k = P_3(V_3)^k = \cdots = const \tag{10}$$

Based on this:

$$\frac{P_1}{P_2} = \left(\frac{V_1}{V_2}\right)^k \tag{11}$$

In the known designs of the pneumatic hammer, the stealing pressure decreases before the exhaust by 2.5-2.7 times, which, in accordance with (10), corresponds to an increase in the initial volume of compressed air by about 2 times, i.e.

$$V_2 = 2V_1 \tag{12}$$

Substituting in (8) instead V_2 of its value from (11) we get:

$$\frac{P_1V_1}{k-1}[(V_2)^{k-1} - (V_1)^{k-1}] + P_mV_1(V_2)^{k-1} = P_m(2V_1 - V_1)(2V_1)^{k-1} + \frac{\dot{m}\vartheta^2}{2}t_2 \tag{13}$$

$$\frac{P_1{V_1}^k}{k-1}[(2)^{k-1} - 1] + P_m2^{(k-1)}{V_1}^k = P_m2^{k-1}{V_1}^k + \frac{\dot{m}\vartheta^2}{2}t_2 \tag{14}$$

As you know, from gas dynamics, the one-time second flow rate of gas (air) is equal to:

$$\dot{m} = \rho\vartheta S \tag{15}$$

where:

ρ - density of the gas (compressed air), kg/m^3;

ϑ - gas flow rate, m/s;

S - cross-sectional area of the gas flow, m^2.

The air flow rate is equal to the speed of the air coming from the line into the chamber B (return stroke of the striker) and is equal to the speed of the striker at the end of the section X_1 of the isobaric expansion of compressed air.

According to the well-known formulas of mechanics at $\vartheta_0 = 0$,

$$\vartheta = at_2 \tag{16}$$

where:

a - acceleration of the object, m/s^2; ($a = F/m_b$, where: F- force acting on the striker, N; m_b- weight of the striker, kg);

t - time of movement of the object with a given acceleration, s;

ϑ_0 - initial speed, m/s.

In section X_1 of the isobaric expansion of compressed air:

$$F = p_mS_b \tag{17}$$

where is the line pressure p_m

$$\vartheta = \frac{p_mS_b}{m_b}t_2 \tag{18}$$

where S_b - area of the butt (cross-section) of the striker, m^2.

Substituting (16) and (17) into (15) and further into (14) we get:

$$\dot{m} = \rho\frac{p_mS_b}{m_b}t_2S \tag{19}$$

In our case, the cross-sectional area of the gas flow is equal to the area of the end face of the cross-section of the striker, taking this into account, we obtain:

$$m^{\cdot} = \rho\frac{p_mS_b}{m_b}t_2 \tag{20}$$

As you know, the cycle of operation of the hammer striker consists of the time of forward and reverse stroke of the striker. The working stroke time of the striker is equal to the time of its acceleration and is determined as follows:

$$t_c = t_{\mathrm{p}} + t_{ob\mathrm{p}} \tag{21}$$

where the time t_c of one cycle of movement of the striker is set from the drilling conditions. Since the striker with the accelerator and the valve moved from a state of rest, then:

$$\vartheta_{kr} = a_pt_p,\ \text{from this}\ t_p = \frac{\vartheta_{kr}}{a_p} \tag{22}$$

where:
t_p - acceleration time of the bike with the accelerator 5 and the valve 6 to the set speed, s;
a_p - acceleration of the accelerator bike and the valve at the stage of the working stroke m/s^2;
ϑ_{kr} - the final specified speed of the accelerator striker and valve at the stage of the working stroke, m/s;

$$\vartheta_{kr} = \sqrt{\frac{2K_b}{m_b+m_y+m_{kl}}} \tag{23}$$

At $F = p_m S_y$, where: F - pressure force of compressed air on the accelerator, N; S_y - end area of the end section of the accelerator, m^2. We get from here $a_p = \frac{F}{m_b+m_y+m_{kl}} = \frac{p_m S_y}{m_b+m_y+m_{kl}}$, where m_y - mass of the accelerator, kg; m_{kl} - valve weight, kg.

Substituting (23) into (22) we get:

$$t_p = \frac{\sqrt{\frac{2K_b}{m_b+m_y+m_{kl}}}(m_b+m_y+m_{kl})}{p_m S_y} \tag{24}$$

The striker return time consists of the acceleration time with isobaric expansion of compressed air - t_1 and the movement time of the striker with adiabatic expansion of compressed air - t_2.

Based on the known formulas of mechanics:

$$X_1 = \frac{a_{obp1} {t_1}^2}{2} \tag{25}$$

from here

$$t_1 = \sqrt{\frac{2x_1}{a_{obp}}} \tag{26}$$

for $a_{obp} = \frac{F}{m_b}$ and $F = p_m S_b$, we get:

$$a_{obp} = \frac{p_m S_b}{m_b} \tag{27}$$

where:
X_1 - striker path at the stage of isobaric expansion of compressed air, m;
a_{obp1} - striker acceleration at the stage of isobaric expansion of compressed air, m/s^2.
t_1- time of movement of striker on the segment, s;
F- force of action on the striker, N.

Substituting (27) into (26) we get:

$$t_1 = \sqrt{\frac{2X_1 m_b}{p_m S_b}} \tag{28}$$

Striker return time:

$$t_{obp} = t_1 + t_2 \tag{29}$$

Consists of t_1- time of movement in the section, X_1- path in the section of the adiabatic expansion of compressed air.

From (21) we obtain t_{obp} and from (29) t_2 we obtain:

$$t_2 = t_{obp} - t_1;\ t_2 = t_c - t_p - t_1 \tag{30}$$

Substituting into (13) the values m, ϑ and t_2 from formulas (17), (20) and (30) we obtain:

$$\frac{P_1 {V_1}^k}{k-1}[2^{k-1} - 1] + P_m 2^{(k-1)} {V_1}^k = P_m 2^{k-1} {V_1}^k + \rho \frac{\dot{p}_m {S_b}^2}{m_b} t_2 \left(\frac{\dot{p}_m {S_b}^2}{m_b}\right)^2 t_2 \cdot 0{,}5 \tag{31}$$

Given that $V_1 = S_b X_1$ and substituting into (31) instead V_1 of its value $S_b X_1$ we obtain (32).

$$\frac{P_1 (S_b X_1)^k}{k-1}[2^{k-1} - 1] + P_m 2^{(k-1)} (S_b X_1)^k = P_m 2^{k-1} (S_b X_1)^k + \\ +0{,}5\rho \left(\frac{\dot{p}_m {S_b}^2}{m_b}\right)^3 \left(t_c - \frac{\sqrt{\frac{2K_b}{m_b+m_y+m_{kl}}}(m_b+m_y+m_{kl})}{p_m S_y} - \sqrt{\frac{2x_1 m_b}{p_m S_b}}\right) \tag{32}$$

There X_1 is only one unknown in this equation.

This equation does not have an analytical solution and is solved by numerical methods. Having obtained X_1 the value from relation (11), we obtain $X_2$and the entire length of the reverse motion of the bike, from where $V_2 = 2V_1,\ x_2 = 2x_1.\ L_{obp} = x_2 + x_1 = 3x_1$.

Knowing the acceleration time of the striker with the accelerator and the valve, as well as their acceleration, we determine the path of the bike acceleration.

Accelerator striker acceleration path and valve: $L_{\mathrm{p}} = \frac{a_p(t_p)^2}{2} = \frac{1}{2}\frac{p_m S_y}{m_b+m_y+m_{kl}} \cdot \frac{K_b(m_b+m_y+m_{kl})}{(p_m S_b)^2}$;

$$L_{\mathrm{p}} = \frac{K_b}{p_m S_b} \tag{33}$$

Next, the linear dimensions of all parts of the DTH hammer can be determined.

The valve motion equation is compiled based on the knowledge of its location by the time of docking with the striker and accelerator. The path traveled by the valve from the moment of its cutoff to the moment of docking with the striker shank is equal to:

$$X_k = a_k {t_k}^2/2 \tag{34}$$

where:

X_k - path traveled by the valve from the moment of its cut-off to the moment of docking with the striker shank, m;

a_k - valve acceleration, m/s^2;

t_k - valve movement time from the moment of its cut-off to the moment of docking with the striker shank, s;

presenting $a_k = F/m_k$ and substituting into equation (9) we get:

$$F/m_k = 2X_k L {t_k}^2 \tag{35}$$

where:

F_k - resulting force acting on the valve, H;

m - mass of the valve, kg.

To fulfill the equality, the ratio of the resulting force acting on the valve to its mass is selected. The resulting force acting on the valve is controlled by changing the flow area of the discharge channel 21 (in the stop of the valve 7), and the mass of the valve by selecting the material from which it is made.

4. Conclusions

The problem of increasing the efficiency has been solved pneumatic impact machines due to a combination of thermodynamic cycles of compressed air expansion in the working chambers of pneumatic impact machines. A fundamentally new design of a down-the-hole hammer has been developed, in which a theoretically substantiated combination of thermodynamic cycles of compressed air expansion is implemented.

The use of simple and effective improvements in constructions makes it possible to increase the efficiency of pneumopercussion machines. These solutions are especially relevant for the mining industry. Within the limited geometrical parameters of the mine workings, it is possible to increase the efficiency of small-scale mechanization significantly, to reduce the cost of final products, and to obtain a sufficient economic effect. The advantages of the innovative solution are as follows:

1. The proposed design of the pneumatic hammer makes it possible to reduce the compressed air consumption by 1.7-1.9 times at the same values of the frequency and energy of blows as in the available pneumatic hammers.

2. Based on this technical solution, DTH pneumatic hammers, rock drills, and other pneumopercussion machines can be created, the use of which in various fields of industry can reduce the cost of drilling and of other types of work significantly.

References

[1] Griadushchiy Y., Korz P., Koval O., Bondarenko V., Dychkovskiy R.: Advanced Experience and Direction of Mining of Thin Coal Seams in Ukraine. Technical, Technological and Economical Aspects of Thin-Seams Coal Mining, International Mining Forum 2007: 2-7. https://doi.org/10.1201/noe0415436700.ch1

[2] Kolb A., Pazynich Y., Mirek A., Petinova O.: Influence of voltage reserve on the parameters of parallel power active compensators in mining. E3S Web of Conferences 2020, 201: 01024. https://doi.org/10.1051/e3sconf/202020101024

[3] Hankevich V., Moskalova T., Kabakova L., Livak O.: The feasibility evaluation of using cyclic thermal effect in the rock-cutting tools during drilling hard rock. E3S Web of Conferences 2019, 109: 00026. https://doi.org/10.1051/e3sconf/201910900026/

[4] Dychkovskyi R.O., Avdiushchenko A.S., Falshtynskyi V.S., Saik P.B.: On the issue of estimation of the coal mine extraction area economic efficiency. Naukovyi Visnyk Natsionalnoho Hirnychoho Universytetu2013, 4: 107-114

[5] Lipin A.A.: Down-the-hole hammers with combined air distribution. Machine science 2007, 2: 43-47

[6] Nadutyi V. P., Kurilov V. S., Cholyshkina O. G., Hankevych V. F.: Analytical studies on constrained particle settling velocity in a water suspension of fly ash from thermal power plants. Naukovyi Visnyk Natsionalnoho Hirnychoho Universytetu 2021, 1: 32–38. https://doi.org/10.33271/nvngu/2021-1/032

[7] Dychkovskyi R.O.: Forming the bilayer artificially created shell of georeactor in underground coal well gasification. Naukovyi Visnyk Natsionalnoho Hirnychoho Universytetu 2015, (5): 37–42

[8] Golovchenko A., Dychkovskyi R., Pazynich Y., Edgar C. C., Howaniec N., Jura B., Smolinski A.: Some Aspects of the Control for the Radial Distribution of Burden Material and Gas Flow in the Blast Furnace. Energies 2020, 13(4): 923. https://doi.org/10.3390/en13040923

[9] Alekseev S.E., Repin A.A., Pyatnin A.A.: Creation and introduction into production of powerful DTH pneumatic hammers. Machine science2007, 2: 9-13

[10] Dychkovskyi R., Tabachenko M., Zhadiaieva K., Dyczko A., Cabana, E.: Gas hydrates technologies in the joint concept of geoenergy usage. E3S Web of Conferences 2021, 230: 01023. https://doi.org/10.1051/e3sconf/202123001023

[11] Kosolapov D.V.: Features of dynamic calculations of details of downhole pneumatic hammers. Safety and survivability of technical systems 2009, 9: 171-175

[12] Sobolev V., Bilan N., Dychkovskyi R., Caseres Cabana E., Smolinski A.: Reasons for breaking of chemical bonds of gas molecules during movement of explosion products in cracks formed in rock mass. International Journal of Mining Science and Technology 2020, 30(2): 265–269. https://doi.org/10.1016/j.ijmst.2020.01.002

[13] Lipin A.A.: Promising pneumatic hammers for drilling wells. Physical and technical problems of mining. 2005, 2: 176 -199

Publisher: **KOMAG Institute of Mining Technology, Poland**