Dynamics of gravity-type pneumatic percussion machine for drain hole drilling

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Abstract. The author presents and justifies a construction diagram of a gravity-type pneumatic percussion machine for drain hole drilling with efficient penetration of casing in soil. The working cycle parameters of the machine, efficient design variables, as well as the blow energy, blow frequency and compressed air flow rate depending on the line pressure are determined by the numerical modeling. The weight, size and dynamics of the gravity-type machine are compared with the known analogous facilities.

1. Introduction
Coalbed gas drainage is carried out in holes drilled in coal from ground surface [1]. Such holes can be drilled with simultaneous or subsequent casing and outside cementing. Aimed to enhance economic efficiency of coalbed drainage, the Institute of Mining, SB RAS has proposed some engineering solutions on shallow hole making by vertical ramming of steel pipe in soil with simultaneous removal of soil from the pipe to the ground surface. In this technology, a percussion machine drives steel pipes with diameter from 219 to 426 mm in soil [2]. Soil plug is removed from the pipe by compressed air fed along a special line to the bottom hole end of the pipe (Figure 1a). The penetration of the pipe is not less than 40 m. The pipes are driven in soil without a pit head frame. The vertical ramming technology includes: assembly of a setup for soil plug removal from a driven pipe by compressed air; connection of the pipe to the pneumatic percussion machine using a coupling and a cable; mechanical lift of the percussion machine with the pipe into vertical position; actuation of the percussion machine and ramming; cyclical removal of soil plug from the driven pipe by compressed air flow. In order to facilitate introduction of this technology, a simplest and efficient percussion facility is required.

2. Gravity-type air percussion machine
For rock fracture or driving of steel pipes in soil in the mining and construction industries, percussion machines with hydraulic, electric and pneumatic drives are used [3–5]. Pneumatic percussion machines feature simple design and reliable operation. In the construction industry, vertical driving of steel pipes with a diameter to 426 mm in soil to a depth of 15–20 m is implemented using double action air impact hammers (Typhoon-500, M-400) with a blow energy to 4000 J. These machines approved themselves in horizontal driving of steel pipes with a diameter to 1020–1220 mm in soil [6]. Efficient vertical ramming of pipes to a depth more than 40 m in soil requires higher blow energy (to 8 kJ) as soil becomes denser with depth. Tests of the double action pneumatic percussion machines shows that vertical ramming of pipes can be greatly complicated by the recoil of the percussion machine body, which weakens the stiff
connection between the machine and the pipe. The ramming safety and efficiency lower accordingly. The recoil can be eliminated through the inclusion of a gravity-type percussion device in the machine design.

To this effect, it is proposed to use a gravity-type pneumatic percussion facility with valve air control. Figure 1b demonstrates the component arrangement of such facility with connected pipes, Figure 2a shows the structure diagram of the facility.

![Diagram](image)

**Figure 1.** Ramming process diagram (a) and construction diagram of percussion machine coupled with pipes (b): 1 — soil; 2 — pipe; 3 — coupling; 4 — air percussion unit; 5 — pressure line; 6 — air feed to pipe; 7 — compressor; 8 — opening for soil removal; 9 — crane jib.

After actuation of the machine, compressed air enters power chamber 2 (Figure 2a) from the pressure line via chamber 1 and an opening in the anvil. The hammering piston is moved up from the low position by the air pressure. As elastic valve 4 passes by lateral holes in the body, air is exhausted from the chamber. At the same time, owing to differential pressure in chambers 1 and 2, flat valve 5 is moved upward and shuts the power chamber from the compressed air entrance. Then, the piston moves downward by gravity and hits anvil 1 of the body stiffly connected with the driven pipe. After that, the cycle is repeated.

For the analysis of the machine serviceability, operating cycle and efficiency, as well as in order to determine efficient design and dynamics, a simulation model of the air percussion machine is constructed using a computer program [7, 8]. The model was inserted with physical parameters and design variables of the machine (weight of piston, effective area of piston, volumes of compressed air chambers, etc.), and for each set of values, a solution was obtained in the diagram form: pressures, piston travels and velocities, instant air flow. The operating cycle efficiency was estimated by the blow energy (pre-blow velocity) and average air flow rate in the steady-state operation mode. Interpretation of the diagrams and evaluation of the output characteristics of the operating cycle was performed using formulas (1)–(5) below.

Blow energy, J:

$$ A = \frac{mv^2}{2}, $$  

where $m$ is the weight of the piston, kg; $v$ is the pre-blow velocity of the piston, ms.

Blow frequency, Hz:

$$ f = \frac{1}{T_c}, $$
where $T_c$ is the cycle duration, s.

Figure 2. Design models of air percussion machine: (a) construction diagram; (b) pneumatic connections; (c) mechanical forces: $V_i$—volumes of chambers; $J_i$—areas of air passage cross-sections; $m_i$—masses of movable elements; $S_{ij}$—effective areas of masses $m_i$ from the side of an $i$-th chamber; $T_i$, $T_{pl}$—absolute air temperatures in the $i$-th chamber and pressure line, respectively; $p_{pl}$, $p_i$, $p_a$—absolute air pressures in the pressure line, $i$-th chamber and in the atmosphere.

Blow capacity, $W$:

$$N = A \cdot f.$$  

(3)

Absolute air flow rate, $m^3/min$:

$$Q = \frac{60RT}{pT_u} \int_{t_0}^{t_f} G(t) dt,$$  

(4)

where $R$ is the absolute gas constant; $T$ is the absolute air temperature in the pressure line; $p$ is the atmospheric pressure.

Specific flow rate, $m^3/J$:

$$q = \frac{Q}{60N}.$$  

(5)

Modeling of the operating cycle used the input parameters from the set ranges: absolute pressure in the pressure line of 0.6–1.2 MPa; blow energy of 4–8 kJ; blow velocity of 4–6 m/s; piston weight of 600 kg; piston travel to 1400 mm; area of the pressure line of 5.3–8.5 cm$^2$, body diameter to 400 mm. Figure 2 shows the design models of the construction diagram with the main elements (Figure 2a), generalized pneumatic connections (Figure 2b) and mechanical connections of forces and impacts between the movable elements ($m_1$, $m_2$) and the body $m_3$ (Figure 2c). Figure 3 presents the operating cycle diagrams for the efficient design variables: $m_1 = 600$ kg, $m_2 = 7$ kg, $S_{21} = 583$ cm$^2$, $S_{12} = S_{22} = 200$ cm$^2$, $S_{41} = 779$ cm$^2$, $J_{p_0} = J_{01} = 5.3$ cm$^2$ (8.5 cm$^2$), $J_{a_0} = 210$ cm$^2$, $V_0 = 22670$ cm$^3$, $V_1 = 5800$ cm$^3$, $V_2 = 2700$ cm$^3$. The diagrams prove the efficient operation of the machine since the operating cycle is stable, the power stroke of the piston is performed by gravity, the back run of the piston takes place under the power chamber pressure, and the plate valve acts promptly before the exhaust.
Figure 3. Assumption diagrams of operating cycles of air percussion machine under main pressure of 0.6 MPa: (a) time variation in absolute compressed air pressure $p_i(t)$ in $i$-th chamber; (b) travels of piston $x_1(t)$ and valve $x_2(t)$; (c) piston velocity $v_1(t)$ and valve velocity $v_2(t)$; (f) instant air flow rate $G(t)$; $t$—time, s; $T_c$—cycle duration; $T_{back}$—back run duration; $T_{pow}$—power stroke duration.

Table 1 gives the main characteristics of the operating cycle from the diagrams. As the pressure in the pressure line is increased from 0.6 to 1.2 MPa, we see the increase in the cycle duration from 0.77 to 0.95 s, in the travel of the piston from 800 to 1370 mm, in the piston velocity from 3.86–5.35 m/s and in the absolute air flow rate from 6.3–7.8 m$^3$/min.

| Characteristics          | Value            |
|--------------------------|------------------|
| Pressure $p$, MPa        | 0.70 0.9 1.10 1.30 |
| Cycle duration $T_c$, s  | 0.77 0.82 0.87 0.95 |
| Back run duration $T_{back}$, s | 0.40 0.41 0.44 0.50 |
| Power stroke duration $T_{pow}$, s | 0.37 0.39 0.41 0.45 |
| Power stroke travel $H_p$, cm | 800 1000 1180 1370 |
| Blow velocity $v$, m/s   | 3.86 4.46 4.95 5.35 |
| Absolute air flow rate $Q$, m$^3$/min | 6.30 6.66 7.07 7.80 |

Table 2 presents the calculated characteristics of the gravity-type air percussion machine from formulas (1)–(5) and specifications of the known analogs—double action machines designed at the Institute of Mining, SB RAS. At the increasing pressure line pressure of 0.6–1.2 MPa, the blow energy of the gravity-type hammer grows from 4470 to 8587 J, air flow rate—from 6.3 to 7.8 m$^3$/min and the blow capacity—from 5811 to 9016 W. The specific air flow rate is $(13.9–18.0)\times10^{-6}$ m$^3$/min and the blow frequency is 1.05–1.30 Hz. Air hammers M-400 and Typhoon-500 have similar weights and sizes as the gravity-type air percussion machine, and they operate at the pressure line pressure of 0.6 MPa. They have lower specific flow rates and blow energies. The gravity-type air percussion machine has the higher blow capacity than Typhoon-500. At the pressure of 1.3 MPa, the blow capacity of the gravity-type air percussion machine is comparable with the capacity of air hammer M-400 owing to high blow energy, while the gravity hammer has 2.5 times less absolute air flow rate (7.8 m$^3$/min), which essentially improves competitive ability of the gravity-type machine.
Table 2. Specifications of air percussion machines

| Characteristics                        | Gravity-type air percussion machine (calculation) | Air hammer (certificate data) |
|----------------------------------------|--------------------------------------------------|-------------------------------|
|                                        | M-400   | Typhoon-500 | Typhoon-740 |
| Pressure $p$, MPa                      | 0.6     | 0.6          | 0.6          |
| Blow energy $A$, J                     | 4470    | 3700         | 3700         |
| Blow frequency $f$, Hz                 | 1.3     | 2.8          | 2.8          |
| Air flow rate $Q$, m$^3$/min           | 6.30    | 20           | —            |
| Capacity $N$, Wt                       | 5812    | 10360        | 10360        |
| Specific air flow rate $q$, m$^3$/J    | $18.0 \times 10^{-6}$ | $13.9 \times 10^{-6}$ | $32.2 \times 10^{-6}$ |
| Piston weight $m$, kg                  | 600     | 500          | 740          |
| Machine weight $M$, kg                 | 1710    | 1800         | 1350         |
| Sizes, mm:                             | 3290    | 2590         | 2000         |
| length                                 | 448     | 2590         | 2650         |
| diameter                               | 448     | 448          | 400          |

3. Conclusions
In the drain hoe drilling technology, it is advisable to use the gravity-type air percussion machine as it ensures driving of casing pipes at high blow energy and lower air flow rate. The in-situ tests of the gravity-type air hammer prototype prove the competitive ability of the new gravity-type air percussion machine. The numerical modeling of the operating cycle of the gravity-type machine with valve air control shows that this machine provides the same energy performance as compared with the known analogs of the double-action pneumatic percussion machines. Finally, design and introduction of the new gravity-type air percussion machine can enable higher efficiency of steel pipe ramming in soil at lower energy consumed during this process.

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