E3S Web of Conferences **402**, 10026 (2023) https://doi.org/10.1051/e3sconf/202340210026
TransSiberia 2023

Analysis of dynamic processes in the impact system of drilling machines

Leonid Dvornikov[1*], *Darya* Levacheva[1], *Denis* Ischenko[1], and *Aleksey* Makarov[2]

[1]Scientific Research Centre "MachineStructure, Saint-Petersburg, 197372, Russia
[2]Siberian State Industrial University, Novokuznetsk, Russia

Abstract. Modern drilling machines designed for the destruction of rocks have a diverse structure, various performance characteristics and in most cases a well-defined limited scope of application. The most common for the destruction of rocks of high strength are impact drilling machines. One of the most effective ways to increase the productivity of such machines is to find rational geometric parameters of colliding elements. The article proposes and substantiates a methodology for studying shock systems based on the application of the shock wave theory. According to the calculation results, the parameters and design solutions of the elements of drilling machines are given, the use of which will allow to increase productivity and reduce energy consumption for the destruction of rock.

1 Introduction

Extraction and processing of solid minerals [1], construction of underground structures [2-4], extraction of building materials [5, 6] are associated with the need to destroy large volumes of rocks estimated in Russia at billions of cubic meters per year. A variety of machines and mechanisms [7-10] are created for the destruction of rocks, large labor resources are involved, the destruction of rocks requires huge expenses of energy and money.

Among the methods of rock destruction, and by now known, proven are thermal, thermomechanical, chemical, plasma methods and mechanical method which has a predominant application. It based on the direct impact of a drilling tool made of durable and hardwearing materials on the rock, by cutting, impact, cutting and impact simultaneously.

The greatest effect of destruction is achieved by impact on the rock, on this principle many mining machines are built, in particular, machines used for drilling boreholes and wells [11-14].

Impact drilling machines are rather complex mechanical systems, the justification of the optimal parameters of which is associated with significant economic benefits, consisting in increasing the productivity of drilling operations and reducing energy consumption for drilling.

* Corresponding author: tmmiok@yandex.ru

2 Solution methods

Schematically, drilling machines can be represented as a long rod with a small cross-section (Fig. 1), resting through the tip-drilling tool on the rock being destroyed. The striker generates a wave of longitudinal deformation in the bar-rod, which moves along the bar towards the face, loads the drilling tool and the rock, creating conditions for the destruction of the rock. The amplitude and duration of the stress waves are determined by the materials, shapes and sizes of the colliding bodies. Due to the fact that only a part of the rod is loaded along the length during impact, and the length of the loaded section can be meaningfully set, a very important situation is achieved when with significant forces arising between the sections of the waveguide and then between the tool and the rock, the longitudinal stability of the compressed rods is ensured at almost any length.

As a drive in such machines, various mechanisms are used: lever, cam [15], gear [16]. The specific working conditions of impact machines require the use of high-quality high-strength materials for the manufacture of parts, as well as careful selection of the type of their thermal and technological processing [17-19]. Dynamic impact processes are a feature of drilling machines. And despite the availability of modern automated means of solving engineering problems [20-24], the analysis of the dynamics of the collision of rods requires analytical approaches based on the application of the wave theory of impact.

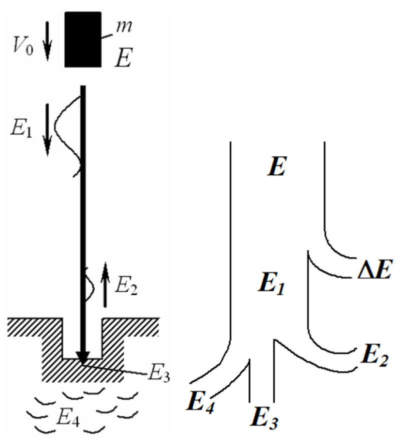

Fig. 1. Scheme of the drilling machines.

The energy stored by the striker E (after impact – E_1) is only partially consumed in the form of E_3 for the destruction of the face. The energy E_4 goes into the medium in the form of seismic waves and dissipates, E_2 is reflected in the form of a longitudinal wave of compression or stretching and rushes to the striking end of the rod. Efficiency of the system

$$\eta' = \frac{E_3}{E}. \quad (1)$$

It is often not possible to determine the value of the energy E_3, and therefore, instead of efficiency, the coefficient of transition of pulse energy into rock (KPI) is used to evaluate the efficiency of the process

$$\eta = \frac{E_1 - E_2}{E_1}. \quad (2)$$

The energy E_1 is easily determined experimentally by the incident shock pulse, and E_2 by the reflected pulse. The coefficient η can be taken in the first approximation as a criterion for the effectiveness of the impact system.

During the interaction of the tool with the face, the face is destroyed, and the tool penetrates into the medium to a certain depth *h*. This linear penetration of the tool is a criterion of distruction productivity.

According to the criteria *h* and *η*, it is possible to judge about the effectiveness of the entire process.

If introduce the concepts of the energy of a single impact $A_y = \frac{mV_0^2}{2}$ and the frequency of impacts, then the power of the impact system will be determined by their product. The stress wave in the rod caused by the impact of the striker on the rod is characterized by the shape of the stress pulse σ_x, the maximum amplitude of the stresses σ_M and the duration $a \cdot T$, where a is the speed of sound in the rod material – 5000 m/s. The same graph can be represented in relative strain coordinates $\varepsilon = \frac{\sigma}{E_y}$, where E_y is the elastic modulus of the waveguide material. By changing the mass m and the pre-impact velocity V_0 of the striker, as well as σ_x, σ_M and T, it is possible to find the most rational combinations of them for certain conditions of destruction.

Based on the above, the important tasks are to study: the influence of the shapes and sizes of strikers on the form of the wave impulse in the rod; the interaction of the rod with the destructible medium; finding conditions to ensure maximum values of *h* and *η*, find the strength of the waveguide and impact systems as a whole.

The problem of collision of elastic bodies is a classical problem of the elasticity theory, which has been developing and solving since the beginning of the last century. The first solutions to the problem were found by Navier and Saint-Venant. Navier used the Fourier method or the method of separation of variables to solve the problem and Saint-Venant used the so–called Dalembert method or the method of traveling waves.

The method used by Saint-Venant was quite effective and allowed to find solutions as N.A. Kilchevsky points out, in a form that allows practical applications. The Saint-Venant theory is usually called a wave or one-dimensional wave theory of impact. The reliability of Saint-Venant's analytical solutions was confirmed by the experiments of J. Sirs and V.M. Malyshev.

The longitudinal vibrations of the rod (Fig. 2) under the assumption that the rod is homogeneous, the cross-sections of the rod remain planar during the oscillations, move only along the axis of the rod and do not change in area, are described by a well-known homogeneous linear partial differential equation of the second order with constant coefficients of hyperbolic type

for a striker:

$$a^2 \cdot \frac{\partial^2 w(x,t)}{\partial x^2} + a^2 \cdot \frac{1}{S(x)} \cdot \frac{dS(x)}{dx} \cdot \frac{\partial w(x,t)}{\partial x} - \frac{\partial^2 w(x,t)}{\partial t^2} = 0 , \quad (3)$$

for a rod:

$$\frac{\partial^2 u(x,t)}{\partial t^2} = a^2 \frac{\partial^2 u(x,t)}{\partial x^2} , \quad (4)$$

where $u(x,t)$, $w(x,t)$ is the displacement function of the cross–section of the rod and striker respectively with the x coordinate at time t, $S(x)$ is a function of the cross-sectional area.

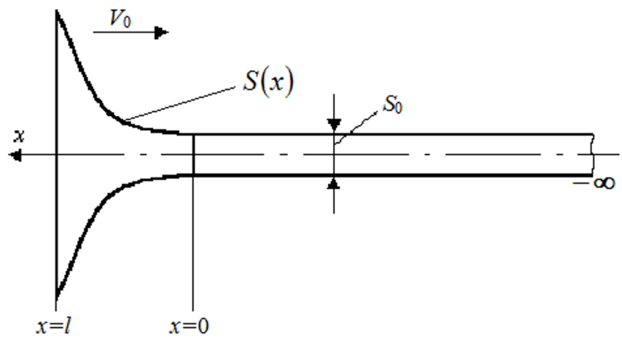

Fig. 2. Scheme of the impact.

In addition, initial and boundary conditions are recorded. In this case, the rod, as a rule, is considered semi–infinite due to the absence of a reflected pulse, and the coordinate system is taken in such way that its origin coincides with the place of collision of the striker and the rod, i.e. the non-strike end of the striker has a coordinate $x = l$, where l is the length of the striker.

The initial conditions are as follows:

– at the moment of the beginning of the interaction, the displacements of the sections of the rod $u(x,t)$ and the striker $w(x,t)$ are zero:

$$w(x,0) = 0, \quad u(x,0) = 0; \tag{5}$$

– the displacement velocity determined by the partial time derivative for the striker is equal to its pre-impact velocity V0, and for the rod is zero:

$$\frac{\partial w(x,0)}{\partial t} = V_0, \quad \frac{\partial u(x,0)}{\partial t} = 0. \tag{6}$$

Boundary conditions determining the state of the ends of the striker and the rod:

– in the process of interaction, the displacements at the boundary of the striker and the rod are equal:

$$w(0,t) = u(0,t); \tag{7}$$

– in the process of interaction, the interaction forces at the boundary of the striker and the rod are equal:

$$S(0)\frac{\partial w(0,t)}{\partial x} = S_0 \frac{\partial u(0,t)}{\partial x}, \tag{8}$$

where S_0 is the cross-sectional area of the waveguide (assumed constant);

– the non-impact end of the striker is free from deformations:

$$\frac{\partial w(l,t)}{\partial x} = 0; \tag{9}$$

– since the rod is semi-infinite, there are no deformations in the sections of the rod removed from the impact:

$$\lim_{x \to -\infty} \frac{\partial u(x,t)}{\partial x} = 0. \tag{10}$$

From the system of differential equations (1) and (2), taking into account the initial and boundary conditions, it is possible to determine the shock pulse, which is associated with the displacement function of the sections of the rod by the dependence:

$$F(x,t) = ES_0 \frac{\partial u(x,t)}{\partial x}, \tag{11}$$

and is determined in the section of the striker's and the rod contact at $x = 0$.

Depending on the striking body's shape, the forms of wave deformations in the rod differ significantly.

3 Results and discussion

Using the D'Alembert method, the graphical method of characteristics, and the Laplace-Carson integral time transformation, the forms of longitudinal vibration pulses for various forms of striking bodies have been found so far [25].

So, with constant energy stored by the striker of the impact mechanism before impact, the mass, pre-impact velocity and shape of the striking body can be significantly different.

Limiting the length of the wave pulse from the point of view of ensuring the longitudinal stability of the rod dictates the need to reduce the length of the striker, and, consequently, with limited transverse dimensions and its mass.

If the stress between the sections of the rod is within the permissible limits ($\sigma \leq 800$ **MPa**), then with regard to rods with a cross-section used for drilling holes, it is possible to allow the interaction force of the striker with the rod up to 600kN. At the same time, the critical length ensuring the stability of the rod will be about 1m, therefore, the firing pin should not exceed $l = 0{,}5$ m, and its mass with reasonable cross-sections should be limited to 15kg.

On the other hand, if we consider a vibro-impact system as a working organ that provides a constant supply of energy to the rod in the form of deformation waves, then its impact power at given constant values of the pre-impact velocity and the acting axial force F in the system depends inversely on the square root of the mass of the impactor, then the frequency of impacts

$$\omega = \sqrt{\frac{l}{m} \cdot F^3} \quad . \tag{12}$$

An increase in mass leads to an intense drop in the frequency of impacts and is therefore impractical.

In impact systems under consideration the pre-impact velocities cannot be randomly increased. Elementary reasoning shows that there is a connection between the stresses in the rod and the pre-impact velocity in the form

$$\frac{\sigma}{E_y} = \frac{V_0}{a} \, . \tag{13}$$

So that the stresses arising in the rod do not exceed the permissible ones, V_0 should not exceed the values

$$V_{0\max} \leq \frac{\sigma}{E_y} \cdot a = 10 \text{ m/s}. \tag{14}$$

Thus, it is possible to approximately find an objective limitation on the energy of a single impact of systems used for drilling holes in rocks. It is determined by the value

$$A_{y\max} = \frac{m_{\max} V_{0\max}^2}{2} = 750 \text{ N}\cdot\text{m}. \tag{15}$$

Studies conducted to date have shown that there is a definite relationship between the longitudinal displacement of the tool and the strength of the resistance to deepening. Studies have established that at the initial moment of deepening, some force other than zero is required, which increases exponentially as the tool invades. Such a model is described by the expression

$$R = R_0 \cdot e^{ku} \, , \tag{16}$$

where k is the coefficient determining the hardness of the face (rock).

The effect of the tool's interaction with the rock is determined not only by the stiffness in contact, but also by the pulse parameters. It is very convenient to consider and analyze this process depending on the complex parameters s and β **,** which are defined as

$$s = \frac{kaT}{E_y \cdot S_0}, \ \beta = \frac{R}{E_y \cdot S_0}. \tag{17}$$

It is the parameter s that is the main one determining the coefficient of use of the pulse energy for destruction.

Theoretically, situations are possible when η from (2) will take values from 0 to 1. The condition $\eta = 0$ is typical for cases when all the energy of the incident pulse returns to the waveguide either in the form of a compression wave (the face is absolutely rigid) or in the form of a stretching wave (there is no resistance to the face). The condition $\eta = 1$ is possible when all the energy of the incident pulse goes to the destruction of the face. Then the energy of the reflected wave will be zero.

The task of finding conditions that ensure $\eta = \max$ r minimum the energy reflected from the face becomes very important.

Studies show that, depending on s and β, defined by expression (17), the incident pulse must have a special shape of the leading edge. The incident pulse in amplitude should begin with a certain value and increase with an intensity corresponding to the intensity of the growth of the rock resistance to invasion.

It is natural to assume that a striker forming an impulse in which the amplitude increases according to a linear law or with an intensity increasing over time can give the best results in terms of the efficiency of the destruction of the face. A striker satisfying this condition must obviously be of variable cross-section, the area of its cross-section must increase from the impact end and the forming part of the striker must be in the area between the forming parts of the striker, made in the form of a truncated cone and a stepped shape. These requirements are met by the generatrix in the form of a curve concave towards the longitudinal axis of the striker. One of the functions describing this kind of generatrix is the catena - chain line. It is the semi-catenoid of rotation that can be taken as the form of a striker that best meets the requirements of the efficiency of the face destruction.

4 Conclusions

Finding rational forms of strikers provides not only an increase in the productivity of rock destruction, it leads to a decrease in the stress level in the waves reflected from the face, and thereby, consequently, reduces the dynamic impact on the drill rods – waveguides and on the machines driving the strikers.

Drilling machines are complex highly loaded multi-link systems with bearing assemblies, the reliability of which is very low. The increase in their reliability is, firstly, due to decrease in the level of loads acting on the parts, and, secondly, due to the creation of scientifically based methods of strength calculation of components at the design phase. Lowering the level of loads can be achieved by using rational forms of strikers. The types and parameters of the deformation waves reflected from the face can be the source material for the guaranteed calculation of the strength of machines. Rational forms of strikers allow to reduce the level of loads on waveguides, reduce vibration and noise in machines, change the dangerous alternating loading of waveguides to less dangerous, which will increase the fatigue resistance of the rods.

The above approach makes it possible to choose a rational shape and geometry of the drilling tool equipment through the coefficients of the face rigidity reasonably, which contributes to the maximum transfer of energy of incident pulses to the face.

References

1. A.V. Teplyakova, I.A. Zhukov, N.V. Martyushev, Sust. Dev. of Mount. Ter., **14,** 3, 501-511 (2022), doi: 10.21177/1998-4502-2022-14-3-501-511.
2. D.A. Yungmeister, M.I. Timofeev, A.I. Isaev, E.E. Gasymov, MIAB, **1**, 107-118 (2023), doi: 10.25018/0236_1493_2023_1_0_107.
3. A.V. Mikhailov, Yu.A. Kazakov, D.R. Garufullin, O.Yu. Korotkova, A. Agaguena, MIAB, **6-1**, 251-267 (2022), doi: 10.25018/0236_1493_2022_61_0_317.
4. P.A. Demenkov, V.V. Komolov, MIAB, **6**, 97-110 (2023), doi: 10.25018/0236_1493_2023_6_0_97.
5. D.A. Zadkov, V.V. Gabov, N.V. Babyr, A.V. Stebnev, V.A. Teremetskaya, MIAB, **6-1**, 317-330 (2022), doi: 10.25018/0236_1493_2022_6_0_46.
6. V.V. Gabov, D.A. Zadkov, V.X. Nguyen, M.S. Hamitov, V.V. Molchanov, MIAB, **6-2**, 205-222 (2022), doi: 10.25018/0236_1493_2022_62_0_205.
7. I.A. Zhukov, N.V. Martyushev, D.A. Zyukin, A.M. Azimov, A.I. Karlina, Metallurgist, **66**, 11-12б 1644-1652 (2023), doi: 10.1007/s11015-023-01480-w.
8. D.A. Yungmeister, A.I. Isaev, E.E. Gasymov, Gor. Zhur., **7**, 72-77 (2022), doi: 10.17580/gzh.2022.07.12.
9. D.A. Yungmeister, A.I. Isaev, E.E. Gasymov, MIAB, **6-2**, 251-267 (2022), doi: 10.25018/0236_1493_2022_62_0_251.
10. O. Su, O. Yarali, N.A. Akcin, Rock Mech. and Rock Eng., **46**, 1577-1587 (2013), doi: 10.1007/s00603-012-0358-3.
11. V.I. Bolobov, V.A. Plashchinsky, MIAB, **3**, 78-96 (2022), doi: 10.25018/0236_1493_2022_3_0_78.
12. H. Aldannawy, A. Rouabhi, L. Gerbaud, Rock Mech. and Rock Eng., **55**, 3, 1555-1570 (2022), doi: 10.1007/s00603-021-02707-5.
13. H. Zhi-Qiang, Z. Cao, M. Ya-Chao, X. Dou, Z. Jing, W. De-Gui, Chin. J. of Eng. Des., **28**, 5, 662-670 (2021), doi: 10.3785/j.issn.1006-754X.2021.00.068.
14. K.O. Afebu, Y. Liu, E. Papatheou, Neu. Net., **148**, 266-284 (2022), doi: 10.1016/j.neunet.2022.01.021.
15. A. Teplyakova, I. Zhukov, E3S Web of Conf., **326**, 00023 (2021), doi: 10.1051/e3sconf/202132600023.
16. L.T. Dvornikov, I.A. Zhukov, Russ. Engin. Res., **42,** 6, 541-547 (2022), doi:10.3103/S1068798X22060090.
17. V.V. Maksarov, A.I. Keksin, I.A. Filipenko, Tsvet. Met., **7**, 82-87 (2022), doi: 10.17580/tsm.2022.07.09.
18. V.V. Maksarov, J. Olt, A.I. Keksin, R.A. Shcheglova, Cher.Met., **2**, 49-55 (2022), doi: 10.17580/chm.2022.02.09.
19. J. Olt, V. Maksarov, G. Petrishinc, E. Panteleenko, M. Liskovich, Russ. Eng. Res., **43**, 3, 314-318 (2023), doi: 10.3103/S1068798X23030243.
20. I.A. Zhukov, N.S. Golikov, N.V. Martyushev, Sust. Dev. of Mount. Ter., **14**, 1, 142-150 (2022), doi: 10.21177/1998-4502-2022-14-1-142-150.

21. E.G. Timofeev, A.V. Teplyakova, I.A. Zhukov, N.S. Golikov, MIAB, **12-2**, 257-269, doi: 10.25018/0236_ 1493_2022_122_0_257.
22. A.M. Slidenko, V.M. Slidenko, J. of Phys.: Conf. Ser., **1902**, 012024 (2021), doi: 10.1088/1742-6596/1902/1/012024.
23. A. Slidenko, V. Slidenko, E3S Web of Conf., **371**, 03047 (2023), doi: 10.1051/e3sconf/202337103047.
24. W.-S. Ding, L. Tian, K. Liu, Huan. Lig. Dax. Xue., **44**, 11, 63-70, doi: 10.3969/j.issn.1000-565X.2016.11.010.
25. A.V. Teplyakova, A.M. Azimov, L. Alieva, I.A. Zhukov, MIAB, **9**, 120-132 (2022), doi: 10.25018/0236_ 1493_2022_9_0_120.