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Inverse analysis method for mathematical modeling of hydrodynamic ballast in a drilling rig

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Introduction. When organizing drilling operations, one of the major problems is the accuracy and smoothness of lowering bundles of pipes into the shaft of the drilling rig. This depends on many factors, including the operation of the hydraulic brake of the lifting device. The objectives of this work are to create and study a mathematical model of hydrodynamic ballast in a drilling rig. Using the inverse analysis method, the effect of some performance indicators on the braking torque of the hydraulic brake is studied.

Materials and Methods. The experiments were performed using a laboratory setup, which is a model of a hydrobrake. Its valve was closed under various conditions to obtain several pressure values with the calculation of the braking torque when a certain weight was suspended. The real (field) operating conditions of the hydromatic brake were simulated, and the results obtained were compared. When creating a mathematical model, the inverse analysis method is used. It is based on the results of experimental measurements and provides expressing the totality of the effects of individual variables on the braking torque.

Results. A mathematical model of the hydraulic brake has been created and tested. The dependence of the braking torque on the pressure, density, and viscosity of the ballast fluid is determined. The influence of each variable is determined experimentally since the dependence under consideration cannot be represented as a direct relationship. The inverse analysis method is used to obtain a set of constant values that give the optimal solution. Taking into account the standard error array and the minimum standard error, the statistical errors made during experimental measurements are considered. The physically acceptable range of values of the proposed mathematical model is visualized. Using a basic (nonlinear) mathematical model, the auxiliary braking torque of a hydrobrake is calculated as a function of pressure, density, and viscosity. The proposed model validity is established. The calculated values of the braking torque were used as a criterion of correctness. The erroneous discrepancy did not exceed 6 %. For additional testing of the model, a computational experiment simulating field conditions was performed.

Discussion and Conclusions. For mathematical modeling of hydrodynamic ballast in a drilling rig, it is advisable to use the inverse analysis method. The model proposed in this paper relates the braking torque of a hydrobrake to the operating parameters of the fluid inside the ballast: pressure, viscosity, and density. The objectivity of the model is validated. An amendment to it is proposed to simulate the operation of the brake in the field. Based on the results obtained, in future studies it is advisable to test the created model in the field with a real payload.

Keywords: hydromechanical ballast, mathematical modeling, inverse analysis method**.**

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Introduction. When setting up drilling rigs, the operations of lifting and lowering drill pipes are provided with lifting gears (coils). They are equipped with two types of brakes, the main of which is mechanical. During the drilling process, the weight of the pipes exceeds the payload. To compensate for it, an auxiliary control ballast designed to reduce the speed when installing a group of drill pipes is introduced [1–4].

Ballasts are used to control the load and speed of the hook, as well as to absorb the kinetic energy of a group of borehole drill pipes. In addition, ballasts:

— reduce the effort on the drilling rig, especially under heavy loads;

— reduce the wear of the main mechanical ballast elements;

— help to slowly and smoothly stop the load attached to the hook.

One of the types of auxiliary brakes is hydrodynamic. In this case, the water in the ballast converts part of the absorbed kinetic energy of the lifting axis into heat, and due to this, the pipes are lowered and raised [5–7].

When manipulating the pipes, the brake axis is connected to the axis of the lifting gears. During the operation, the moving part of the installation displaces water in the direction of the inclined blades inside the stator, and rotates at a speed equal to the rotation speed of the axis of the lifting gears. Water falls on the stator ribs, and then on its parts. The process is repeated, and the forces that hinder the movement of the rotor increase. This creates a braking torque that reflects the rotational movement of the lifting gear axes and reduces the rotary speed. As a result, speed of lowering the group of drill pipes decreases [8–10].

The braking safety factor (if its value does not exceed the permissible value) coordinates the auxiliary hydrodynamic work and the effects of the operation of two main brake systems, as well as provides a longer service time for the main brake elements. This is done through reducing the wear of the friction discs and the planes of the brake wheels. Increasing the braking torque of the ballast provides the correct braking movement of the cylindrical hoist. This demonstrates the importance of studying the hydrodynamic brake.

The hydraulic power of the brake N is determined from the ratio:

$$N = \rho \cdot g \cdot Q \cdot H\ , \tag{1}$$

where ρ — the density of the working fluid (fresh water); g — gravity acceleration; Q — the amount of working fluid consumed, equal to the volume of the working fluid, which passes through the system of ribs in one working cycle; H — the height (level) of the working fluid in the ballast.

The braking torque M_B of the forced engine braking is determined from the ratio:

$$M_B = \rho \cdot g \cdot Q \cdot H / \omega\ , \tag{2}$$

where ω — angular velocity of rotation of the moving section of the ballast.

Indicators of kinematic braking of ribbed hydraulic machines are determined from the known theoretical ratios of the braking torque:

$$M_B = \lambda_M \cdot \rho \cdot \left(D^5 - d^5\right) \cdot \omega^2\ , \tag{3}$$

$$M_B = \lambda_M \cdot \rho \cdot \left(D^5 - d^5\right) \cdot \frac{n^2}{100}\ . \tag{4}$$

Here, D — the outer diameter of the “ring” of the working fluid formed during the rotation of the rotor wheel and assumed to be equal to the diameter of the wheel of the moving section; d — the inner diameter of the working fluid ring, which depends on the level of the ballast; n — the number of revolutions of the moving section of the ballast (the number of rotating axes), rpm:

$$\left(\omega = \frac{2\pi n}{60} \Rightarrow \omega^2 = \frac{n^2}{100}\right).$$

The hydraulic braking torque coefficient λ_M is a dimensionless value that takes into account the shape of the working cavities in the ballast, the parameters, and the number of brake ribs. In practical calculations, the average value of the hydraulic braking torque coefficient is assumed to be 0.3.

The objective of the study is to investigate the effect of some performance indicators on the braking torque of the hydraulic brake, which is the pressure inside the brake chamber, as well as on the density and viscosity of the working fluid inside the ballast. These indicators are absent in the ratio (4); therefore, we assume that the brake operates at an atmospheric pressure of 1 atmosphere, the fluid is fresh water with a density of 1 g/cm^2 and a viscosity of 1 stoke.

Materials and Methods. Laboratory experiments were conducted at the University of Aleppo (Syrian Arab Republic) on a device that is a model of hydrodynamic brake inhibition (Fig.1).

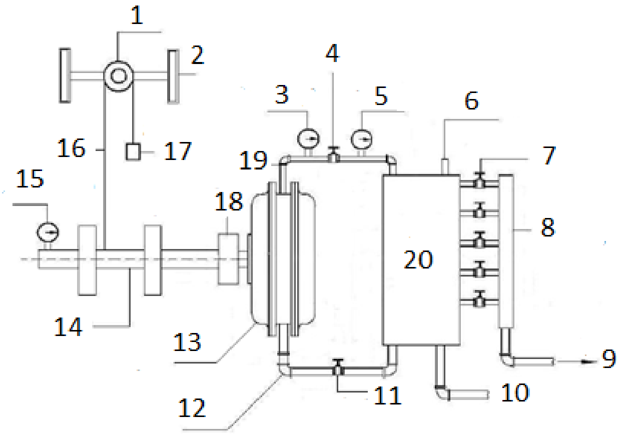


Fig. 1. Laboratory installation — hydrodynamic brake inhibition model:
1 — pulley, 2 — pulley armrest, 3 — pressure gauge, 4 — exhaust valve, 5 — thermometer, 6 — opening in the upper part of the tank, 7 — fluid level control valves, 8 — fluid outlet channel, 9 — fluid outlet from the tank, 10 — fluid drain valve, 11 — inlet valve, 12 — inlet line, 13 — hydraulic brake, 14 — coil, 15 — cycle number meter, 16 — cable, 17 — payload, 18 — communication hub, 19 — control line, 20 — fluid tank.

Stages of experiments

1. The tank and ballast are filled with fluid to the required level (0.106 m to the level of the first valve).
2. Using a hand lever connected to the coil, the weight suspended on the hook is raised by 0.317 m. It weighs 8 kg and is connected to the cable.
3. The bundle of pipes is allowed to fall under its own weight.
4. Readings from the indicator of the rotation of the axis of the coil are taken.
5. Then the impact of the indicators obtained during laboratory experiments on the study of hydraulic braking is recorded:

— the pressure created in the brake chamber (P);

— the density of the working fluid (ρ);

— the viscosity of the working fluid (μ).

First, the braking torque of the auxiliary hydrodynamic brake is calculated from the ratio (4). For this purpose, the values n, D, d are determined.

The outer diameter of the “ring” of the working fluid D is formed when the wheel of the moving section (rotor) rotates and is assumed to be equal to its diameter. The diameter D of the propulsor of the laboratory unit in the practical experiments is 0.33 m.

The inner diameter of the working fluid ring d depends on the level of the ballast. To determine this diameter, the following volumes are aligned:

— working fluid inside the ballast;

— the fluid in the “ring”, which is formed when the movable section rotates inside the ballast and is limited by the height of the coolant in the tank (0.106 m).

At D = 0.33 m, the working volume of the ballast fluid is 2.85 liters. The calculations have shown that the inner diameter d = 0.283 m.

The number of revolutions n is associated with the suspended load. If the brake does not work, this is the number of revolutions of the coiler. If it is separated from the brake, then n is taken from the laboratory experiments: with a load weight of 8 kg, n = 240 rpm.

We calculate the hydraulic braking moment for the weight of 8 kg attached to the hook. During the experiment, the tank was filled with fresh water to the level of the first valve (0.106 m) at the following parameters: pressure $P = 1$ atm, fluid density $\rho = 1$ g/cm^3, and fluid viscosity $\mu = 1$ St. Therefore, the moment of hydraulic braking:

$$M_B = 0.3 \cdot 1000 \cdot \left(0.33^5 - 0.283^5\right) \cdot \frac{240^2}{100} = 355 \text{ Н·м.}$$

Consider the inverse relationship to (2). First of all, we are talking about the inverse relationship between the torque and the angular velocity (ω). A decrease in the number of revolutions n by 20% with an increase in the fluid density means an increase in the torque by 20%. Consider the permissible value of the braking torque 355 N·m. As a result, we get the desired value of the braking torque at the following parameters: pressure $P = 1$ atm, density $\rho = 1$ g/cm^3, and fluid viscosity $\mu = 1$ St.

With regard to this work, we note the following. When determining the braking torque of a hydraulic brake, it is required to consider the number of revolutions of the coil connected to the brake axis (i.e., during braking): a decrease in the winding speed by a certain percentage means the same increase in torque.

The effect of the pressure inside the ballast on the braking torque of the hydraulic brake. Fluid pressure was created inside the hydraulic brake by closing the valve in the fluid outlet line. We conducted a laboratory experiment and calculated the braking torque with a suspended load weighing 8 kg. The results are shown in Table 1.

Table 1

Change in the braking torque of the hydraulic brake when the pressure inside the ballast changes

Pressure inside the ballast, atm	1	1.25	1.4	1.55	1.7	1.85
Left torque limit, N·m	105	104	103	100	95	90
Right torque limit, N·m	355	359	363	372	388	405

Fig. 2 shows the change in the braking torque when the pressure inside the ballast changes.

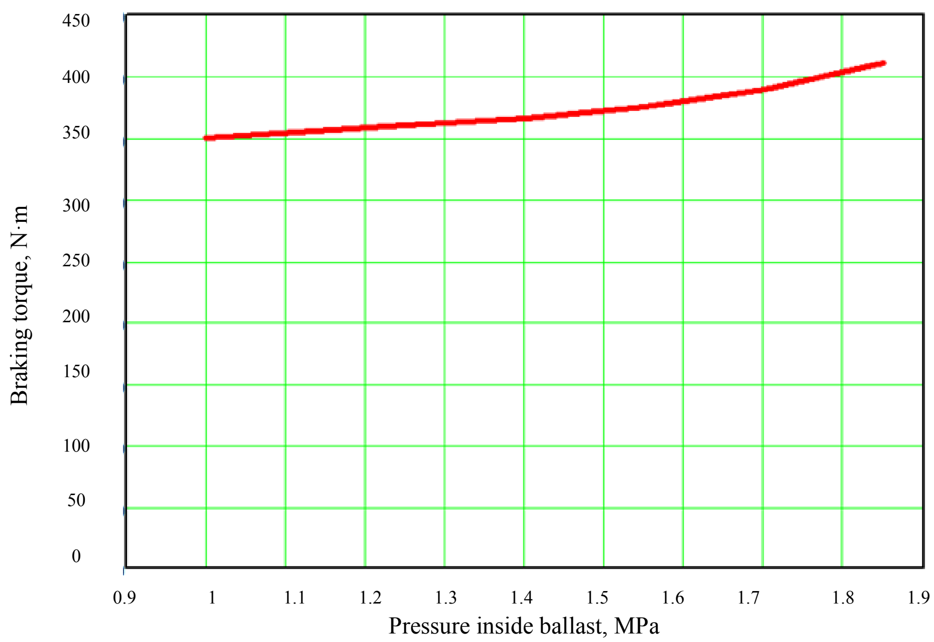


Fig. 2. Change in the braking torque at different pressures inside the ballast

The effect of the working fluid density on the braking torque. The experiments used chemicals that increase the density of water without corroding the elements of laboratory equipment.

Various concentrations of substances dissolved in water for the production of a ballast fluid of different densities are considered. The conditions are the same: the braking torque is investigated with a suspended load of 8 kg (Table 2).

Table 2

Dependence of the braking torque on the fluid density, the number of coil turns, and additives in the ballast fluid

Fluid density	1	1.065	1.09	1.11	1.13	1.15
Number of coil turns	105	104	102	99	95	91
Sodium chloride salts						
Braking torque	355	359	366	375	388	402
Food sugar						
Braking torque	355	359	363	368	378	388

A very weak change in the viscosity with an increase in the concentration of the sodium chloride salt was observed. We can assume that the viscosity of the fluid is approximately equal to 1 St.

Fig. 2 shows the dependence of the braking torque on the density of the working fluid when using sodium chloride salt and food sugar.

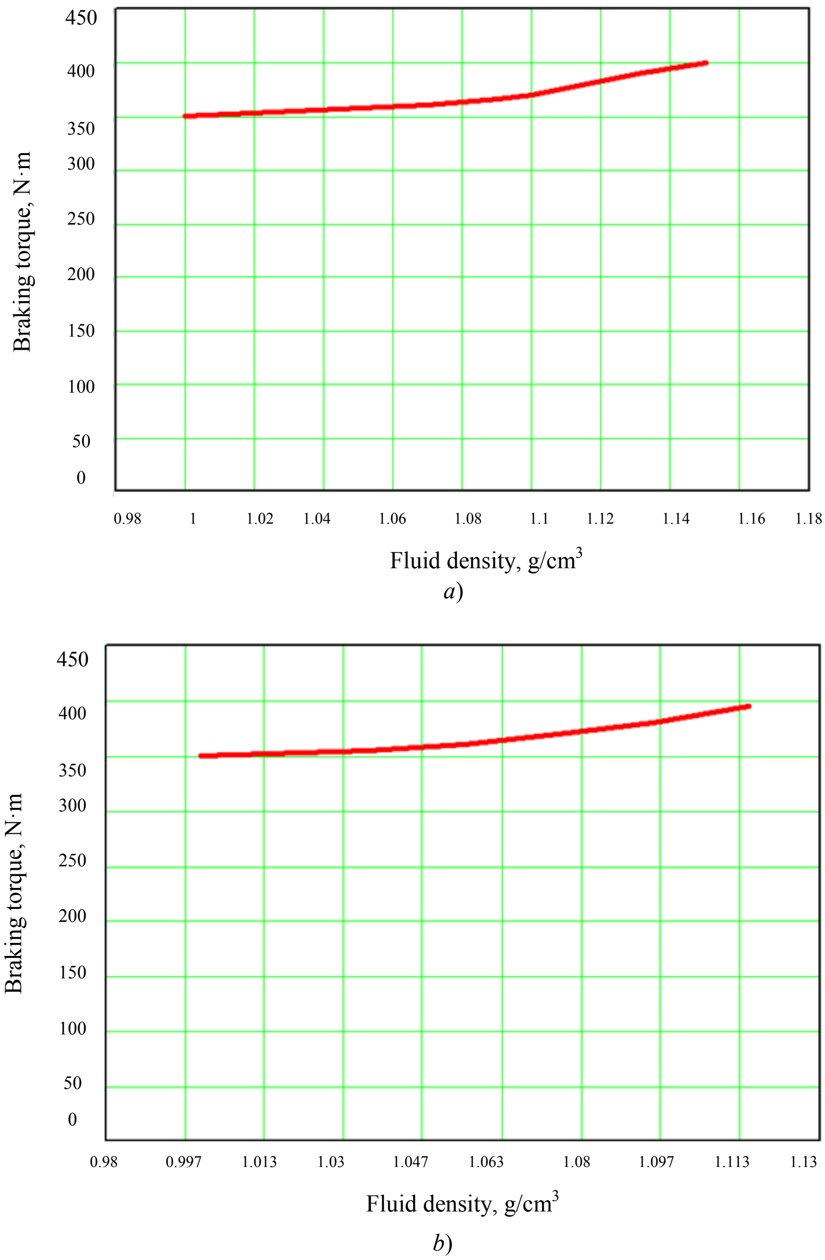


Fig. 3. Dependence of the braking torque on the density of the working fluid with the introduction of sodium chloride salt (*a*) and food sugar (*b*)

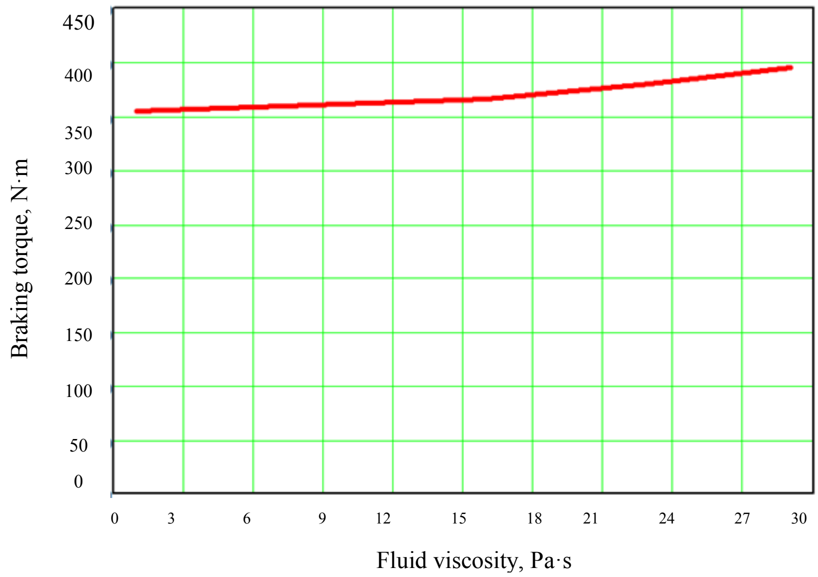
The effect of the working fluid viscosity on the hydraulic system braking. Various combinations of concentrations of glycerin and sodium silicate dissolved in water were used to produce ballast fluids that differ in viscosity. The conditions are the same: the braking torque is investigated with a suspended load of 8 kg (Table 3).

Table 3

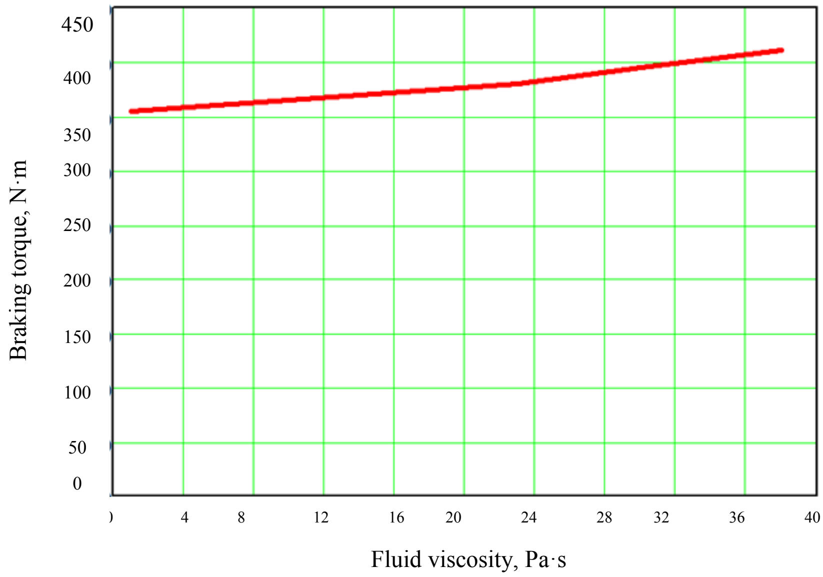
Dependence of the braking torque on the viscosity of the fluid and additives in the ballast fluid

Fluid density	1	16	22	29
Number of coil turns	105	102	98	94
Glycerin				
Braking torque	355	365	378	392
Sodium silicate				
Braking torque	355	372	388	408

Fig. 4 shows the changes in the braking torque with the change in the viscosity of the working fluid when using glycerin and sodium silicate.



a)



b)

Fig. 4. Dependence of the braking torque on the viscosity of the working fluid inside the ballast with the introduction of glycerin (*a*) and sodium silicate (*b*)

The almost linear dependence obtained in this way should be checked using a mathematical model.

During the study of the viscosity index, it was found that the density and viscosity parameters do not depend much on each other, and the density is not equal to 1 g/cm^2 (Table 4).

Table 4

Fluid density and viscosity ratio

Fluid viscosity	16	22	29	23	30	38
Density	1.008	1.014	1.021	1.017	1.025	1.034

The density influence coefficient did not exceed 6 %. It was calculated as the difference between the densities of a viscous fluid and fresh water. For example, if the viscosity was 30 St, the density was 1.025 g/cm^2. Hence, the density impact factor:

$$\Delta = \frac{1.025-1}{1} \cdot 100 = 2.5\% .$$

Research Results

Creating a mathematical model of a hydraulic brake. The dependence of the braking torque on pressure, density, and viscosity cannot be represented as a direct relationship. The effect of each variable on the braking torque is determined experimentally (see Fig. 2–4).

The inverse analysis method. In the scientific and reference literature, there are no recorded indicators of the relationship of the braking torque and pressure, density, and viscosity. Therefore, the inverse analysis method was used [4]. It is efficient for creating mathematical models based on experimental measurements. It can be used to show how the combination of the above variables affects the braking torque. This can be expressed by the relation:

$$D = F\left(P, C\right), \tag{5}$$

where F — a function that relates the considered phenomenon D to a set of variable values and a set of values for constants of the mathematical model of the phenomenon.

The inverse analysis provides finding a set of values of the assumed constants of the model C through the inverse dependence:

$$P = F^{-1}\left(D_m, C\right), \tag{6}$$

where D_m — a set of experimental values for the phenomenon under study [4, 5].

It is assumed that a direct solution to the relation (6) is impossible. Therefore, an iterative system should be used to get a set of constant values that give an optimal solution. The generalization specifies a set of P values for the relations:

$$D_C = F\left(P, C\right), \tag{7}$$

$$\left|D_m - D_C\right| \le \varepsilon, \tag{8}$$

where D_C — a set of calculated values for the phenomenon under study; ε — the required accuracy in accordance with the calculation of the phenomenon under study.

The proposed method takes into account the statistical errors made during the experimental measurements, focusing on the standard error array (S_t) and the minimum standard error. Thus, according to the statistical Gaussian distribution, the dependence of the density of statistical data:

$$f_1\left(P\right) = P_1 = const.\exp\left(-\frac{1}{2}\left[\left(D_C - D_m\right)^t C_d^{-1}\left(D_C - D_m\right)\right]\right). \tag{9}$$

Here, C_d — standard error array. The symbol *t* stands for the matrix:

$$C_d = \begin{bmatrix} S_1^2(1)..........0..................0 \\ 0.............S_1^2(2).............0 \\ 0...............0,,,,,,,,,,,,S_1^2(n) \end{bmatrix}, d, \tag{10}$$

where *n* — the number of control points.

D_c is a function of *P*, so, the problem is related to determining such a value of *P*, that D_c gives the maximum value P_t.

$$S_d = (D_C - D_m)^t \cdot C_d^{-1}(D_C - D_m). \tag{11}$$

The minimum value S_d can be obtained for more than one set of *P* values. Therefore, a correct set of parameters is formed using actual physical modeling of the value of these parameters (Fig. 5) [4, 5].

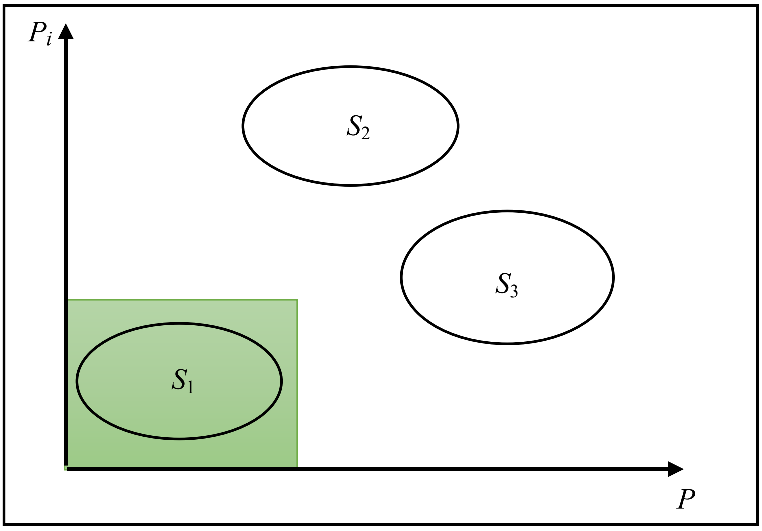


Fig. 5. Determining a physically acceptable range of values (green area):
S_1 — physically acceptable mathematical solution, S_2, S_3 — mathematical solutions

Thus, we use a set of primary values for the parameters P_0, which are derived from the average load of the physical field with account for the standard deviation matrix S_2.

When using new Gaussian distribution, we obtain the following statistical intensity parameter:

$$f_2(P) = P_2 = const.\exp\left(-\frac{1}{2}\left[(P - P_0)^t \cdot C_p^{-1}(P - P_0)\right]\right). \tag{12}$$

Here, C_p — standard deviation matrix:

$$C_p = \begin{bmatrix} S_2^2(1)............0.................0 \\ 0..................S_2^2(2)...........0 \\ 0..................0.............S_2^2(r) \end{bmatrix}. \tag{13}$$

The problem expressed in the relation (6) is solved by inverse analysis. In this case, a set of values of *P* is within the limits indicated in (7) and (8). Hence, it is possible to determine a general region of the parameters P_1 and P_2 from the Gaussian distribution:

$$f(P) = P_1 \cdot P_2 = const.\exp(-S), \tag{14}$$

$$S = \frac{1}{2}\left[(D_C - D_m)^t \cdot C_d^{-1}(D_C - D_m) + (P - P_0) \cdot C_p^{-1}(P - P_0)\right]. \tag{15}$$

So, for the solution, it is required to find the maximum or minimum values of *S* of the function *f* (*P*).

The only way to find the minimum value of S is to use numerical methods, such as Gauss–Newton. This approach is based on the transformation of analytical relations into digital iterative ones with account for the error made due to ignoring some restrictions in the analytical relations (Fig. 6).

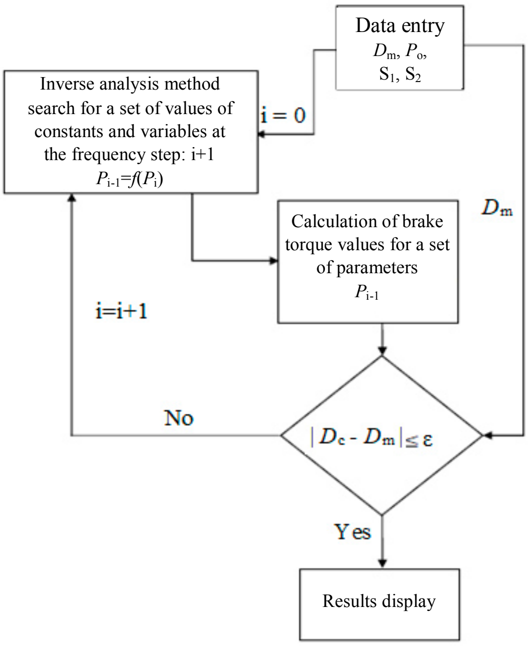


Fig. 6. Inverse analysis method algorithm

At the next stage of the work, the auxiliary braking torque of the hydrodynamic brake was calculated using a basic (nonlinear) mathematical model. It depends on the pressure, density, and viscosity:

$$M_B = f\left(P,\ \rho, \mu\right) + C\,, \tag{16}$$

$$M_B = a \cdot p^{n1} + b \cdot p^{n2} + d \cdot \mu^{n3} + C\,. \tag{17}$$

The mathematical model (17) is a general nonlinear model if the values n_1, n_2, n_3 are not equal to one. The set of constants to search are a, b, d, and n_1, n_2, n_3.

After applying the digital iterative method with an accuracy of 0.001, the parameters presented in Table 5 are found.

Table 5

Parameter values of the proposed model

Parameters	n_1	n_2	n_3	a	b	d	C
Value	1	1	1	58.5	284.6	1.29	10.62

According to Table 5, the physical phenomenon under study can be represented as the linear model (n_1, n_2, n_3) = 1:

$$M_B = 58.5P + 284.6\rho + 1.29\mu + 10.62\,. \tag{18}$$

The mathematical model is based on the results of laboratory experiments and implies the homogenization of units of measurement in accordance with the value of constants.

Adequacy of the mathematical model. The first stage of determining the adequacy of the model is a braking test at:

— pressure P = 1 atm,

— fluid density ρ = 1 g/cm^3,

— fluid viscosity μ = 1 St.

In this case, the braking was equal to the initial braking torque $M_B = 355$ N·m. This is logical, given the experimental values.

The second stage: three laboratory experiments with an 8-kilogram weight suspended on a hook. They are briefly described below.

The first experiment. We took the maximum values of the variables and the braking torque, and then performed the calculation using a mathematical model based on the ratio (18). We determined the error rate through comparing the experimental and mathematical results.

The second experiment. We took random values of variables that are relatively far from the experimental values and from the calculated values for the braking torque (according to the mathematical model). We calculated the error rate through comparing the experimental and mathematical results.

The third experiment. We took random values for variables that are relatively far from the experimental values and from the calculated values for the braking torque (according to the mathematical model). We calculated the percentage of errors through comparing the experimental and mathematical results.

Table 6 shows the results of validating the adequacy of the mathematical model.

Table 6

Results of checking the model adequacy

No.	Pressure	Fluid viscosity	Fluid density	Braking torque (calculation)	Braking torque (mathematical model)	Errors, %
1	1.85	1.15	38	490	495	1
2	1.60	1.10	13	406	434	6
3	1.45	1.12	26	423	447	5

So, the result validates the model adequacy. First, the level of its fallibility is found to be acceptable. Secondly, the identified errors have a scientific explanation. The fact is that the density parameter does not depend on the viscosity, and this is taken into account in the model. But the experiments conducted earlier to confirm this point of view revealed a correlation between these indicators at the level of 6 % — and this is approximately equivalent to the percentage of errors made when using a mathematical model (in comparison with the calculated data).

To express the cumulative effect of the fluid density and viscosity on the hydraulic brake operation, you can enter the parameter $\mu\cdot\rho$ in the model. However, this is not necessary, given the relatively low error rate. Otherwise, the proposed model will become much more complicated.

Simulation of field conditions. The proposed mathematical model expresses a physical phenomenon identified and studied under the laboratory conditions at the initial braking torque 355 N·m.

In the field, ballasts are characterized by an initial braking torque value M_{B0}. It is proposed to introduce into the mathematical model a parameter that will not change its shape, but, presumably, will reflect the "field":

$$M_B = 58.5P + 284.6\rho + 1.29\mu + 10.62 + \left(M_{B0} - M_{Bm}\right), \quad (19)$$

where M_B — the braking torque of the "field" ballast after applying the conditions (pressure, density, and viscosity); M_{B0} — the initial braking torque for the field ballast; M_{Bm} — the initial braking torque of the laboratory ballast, equal to 355 N·m.

This is the main hypothesis. It is substantiated as follows. The dimensions of the laboratory inhibitor are not chosen randomly, but so as to correspond to the actual smaller dimensions of the brakes produced by *Parmac L.L.C* (model 112–500)[1].

[1] Original HYDROMATIC® Brake 620 251-5000. Parmac L.L.C // www.parmacbrake.com.

Discussion and Conclusions

1. The mathematical model is created using the inverse analysis method, which relates the braking torque of the hydrodynamic brake to the operating parameters (pressure, viscosity, and density) of the fluid inside the ballast.

2. The experiments with various random values of variables validated the adequacy of the model. The values of the braking torque determined experimentally and using the created model were compared. The error rate did not exceed 6 % (Table 6).

3. An amendment to the model for simulating the operation of the hydrodynamic brake in the field is proposed.

4. Based on the results obtained, it is advisable to test the created model in the field with a real payload in future studies.

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Claimed contributorship

I. R. Antipas: academic advising; statement of the problem; determination of the research methodology; collection and analysis of the analytical and practical materials on the research topic; critical analysis and finalization of the solution; computer implementation of the problem solution. B. I. Saed: statement of the problem; determination of the research methodology; collection and analysis of the analytical and practical materials on the research topic. A. G. Dyachenko: analysis of the scientific sources on the research topic; critical analysis and revision of the text.

All authors have read and approved the final manuscript.