

Investigating external and internal working processes of mining machines when operating on “unclarified” water in underground conditions

Bulat M. Gabbasov^{1*}, Anton I. Kurochkin¹,
Artur M. Mazhitov¹, Sergei N. Kornilov¹

¹ Nosov Magnitogorsk State Technical University, Magnitogorsk, Russia

*e-mail: bulatg74@mail.ru

Abstract

Introduction. The article considers the issues of mine drainage study, including the working processes of the centrifugal pumps pumping “unclarified” water in arduous underground conditions. Such problems resolution is of high practical and scientific importance.

Methods of research. Scientific and practical experience in the field of mine drainage was analyzed and generalized. The centrifugal pump modes were considered and promising research tasks in this field were outlined. Thus, the continuous income of groundwater to mine workings requires the uninterrupted operation of pumps. One of the most common types of mine-drainage plants is a multistage centrifugal pump which fulfills its functions to the full if properly operated. However, “unclarified” water pumping requires a new technique for centrifugal pump optimal modes determination in such service conditions.

Result and analysis. The study of hydraulic, volumetric, and mechanical efficiency dependency on pump modes, the analysis of working processes within centrifugal pumps when operating on “unclarified” water, and the procedure and calculation of TsNS (multistage centrifugal pump) head-capacity curve were presented in the paper to justify the effectiveness of the presented solutions and conclusions.

Scope of results. It is recommended that the research results are introduced in all enterprises conducting underground mining operations with the mine drainage.

Keywords: shaft centrifugal pumps; optimal modes; efficiency factor; head-capacity curve; hydraulic losses; speed coefficient.

Acknowledgements: The research was funded from the Grant of the President of the Russian Federation no. MD-3602.2021.1.5.

Introduction. Effective and safe operation of underground mines depends on the proper operation of the mine-drainage system that dewateres the mine [1]. Mine drainage is the most important secondary process of underground ore mining because the malfunction of pumps reduces mining safety and increases the running expenses within the cost of production. It is common knowledge that the increase in mineral extraction volume results in a higher concentration of mining operations, increased extent of mine workings and the depth of mining, and, as a consequence, increased water inflow to mine workings. The efficiency of water drainage, in its turn, depends not only on hydrogeological and mine engineering factors but also on the mine-drainage plants applied. Underground water-collecting hollows (sumps) should be regularly clarified from debris so that the mine-drainage plant functions properly [2]. Untimely cleaning of pumping plant sumps results in the active capacity reduction and pumped water contamination with debris, which reduces the pump overhaul period by 2 or 3 times. Besides, sump cleaning is currently a complex and labor-intensive technological process associated with partial flooding of a mine working and requiring the removal of load and haul machines from service, which reduces the efficiency of mining [3]. The cost of power resources grows and creates a need for new energy-saving technologies

development for both mine water pumping and water-collecting hollows cleaning. So, it is highly relevant for the enterprises that perform underground mining operations to increase mine drainage operational efficiency and develop effective mine water pumping technologies and water-collecting hollows clean-up facilities [4].

Multistage centrifugal pumps with each stage consisting of an impeller and a guide vane are most common in mine drainage practices. Not a single existing pump is capable of transforming all the power obtained from the electric motor into useful efficiency. This circumstance is due to major losses caused by several negative effects within the pump [5].

The efficiency factor η indicates which part of shaft power N_{sh} (motor power) is used by the pump to obtain the useful efficiency N_u , i.e. for liquid motion through the pipeline:

$$\eta = \frac{N_u}{N_{sh}} = \frac{\rho g Q H}{N_{sh}}, \quad (1)$$

where ρ is the density of the liquid, kg/m^3 ; Q and H are the effective capacity and head of the pump, m^3/s and m respectively.

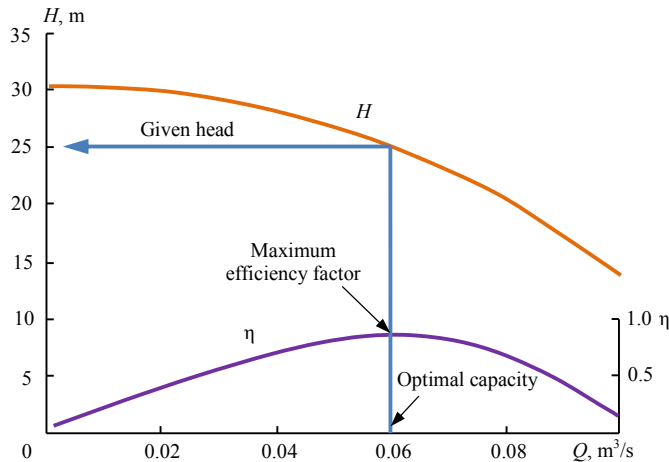


Figure 1. Determining the optimal mode of a pump
Рисунок 1. Определение оптимального режима работы насоса

In its turn, the efficiency factor of the centrifugal pump depends on pump capacity. Pump capacity at which the efficiency factor is maximum is called optimal. The regime of pump operation under the optimal capacity is also called optimal (Figure 1).

To calculate the general efficiency factor of the pump more accurately, it is necessary to consider the negative effects within the pump that reduce its power. The essence of such processes is exacting and is going to be considered further in the paper. The main aim at the stage of pump design is to reduce the negative effects by developing a design that ensures lower power losses [5–7].

The impact of the negative effects on the pump efficiency is estimated with the help of the volumetric, hydraulic, and mechanical efficiency factors and calculated by the formula:

$$\eta = \eta_v \eta_h \eta_m. \quad (2)$$

Volumetric efficiency η_v is the ratio between the effective capacity Q , i.e. the volume of the liquid flowing from the pump to the pipeline per second, and the theoretical capacity

Q_{th} which is the volume of the liquid flowing through the impeller per second, and is determined by the formula:

$$\eta_v = \frac{Q}{Q_{th}}. \quad (3)$$

Effective capacity is lower than the theoretical since a part of liquid passing through the impeller doesn't enter the pipeline but percolates between the body and the impeller and re-enters the impeller. Thus, a particular amount of liquid uselessly circulates within the pump. It is possible to improve the efficiency factor by applying an improved groove seal [6–8].

The preliminary value of volumetric efficiency for centrifugal pump operating in optimal regime is determined by the formula:

$$\eta_v = \left(1 + 0.68 n_s^{-2/3}\right)^{-1}, \quad (4)$$

where n_s is the speed coefficient.

Speed coefficient is the criterion of two pumps similitude that operate in optimal regimes [9]. Speed coefficient of the pump which develops head H , m, in the optimal regime, and capacity Q , m³/s, and the impeller of which rotates with frequency n , rpm, is determined by the formula:

$$n_s = 3.65n \left(\sqrt{Q}/H^{0.75}\right). \quad (5)$$

Speed coefficient is connected with the ratio of the impeller inlet and outer diameters, i.e. it defines the structural type of the pump.

Speed coefficient provides an idea of the efficiency factor of the pump working in an optimal mode. More precisely, the higher the speed coefficient, the more space-saving the pump and its efficiency factor are. Pumps with high speed coefficient are therefore more efficient.

Hydraulic efficiency factor is the ratio between the effective capacity H actually developed by the pump and the theoretical head H_{th} that would be developed by the pump without any hydraulic losses:

$$\eta_h = \frac{H}{H_{th}}. \quad (6)$$

Hydraulic losses are conditioned by liquid friction against the surface of the pump and friction in the vortex flux perturbations. Vortex perturbations are conditioned by the following:

- flux impact on the blades. Liquid flow initially moving progressively runs against the rotating blades and undergoes the impact, which results in strong vorticity;
- blade stall. In the ducts between the blades, liquid outflow from the surface of a blade towards an adjacent blade is recorded, which conditions the development of vortices in the ducts between the blades;
- turns, narrow and wide spots inside the body. The variation of velocity and direction of liquid fluxes in turns, narrow and wide spots inside the body cause the development of vortices.

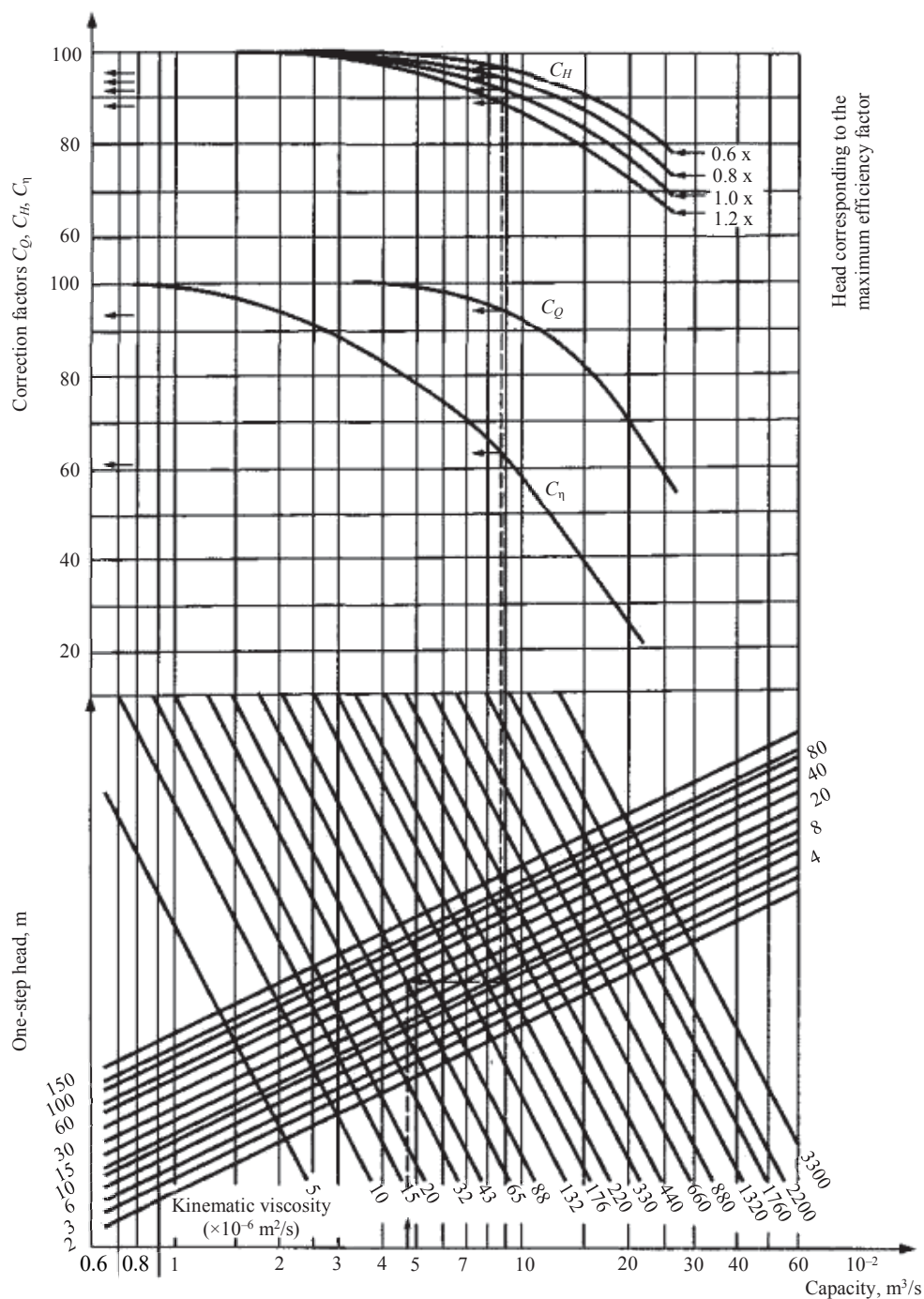


Figure 2. The coefficients of pump characteristics recalculation from water to viscous liquids
 Рисунок 2. Коэффициенты пересчета характеристик насоса с воды на вязкие жидкости

Mechanical efficiency factor is the ratio between the blades power and shaft power or, in other words, mechanical losses composed of friction losses in bearings, stuffing boxes, and balancing rings of the impeller, as well as losses of the impeller external surface friction against the liquid:

$$\eta_m = \frac{\rho g Q_{th} H_{th}}{N_{sh}}. \quad (7)$$

Statistical data reveals that the values of the mechanical efficiency factor of the centrifugal pumps operating in the optimal mode are within the interval of 0.92 and 0.99. For this reason, for preliminary calculation, it is accepted that $\eta_m = 0.96$. However, this value is correct for water pumping [10, 11].

Considering the expressions, the general efficiency factor of the pump in the design is calculated by the formula [12]:

$$\begin{aligned} \eta &= \left(1 + 0.68 \left(3.65n \frac{\sqrt{Q}}{H^{0.75}} \right)^{-2/3} \right)^{-1} \frac{H}{H_{th}} \frac{\rho g Q_{th} H_{th}}{N_{sh}} = \\ &= H \rho g Q_{th} \left(N_{sh} + 0.68 N_{sh} \left(3.65n \frac{\sqrt{Q}}{H^{0.75}} \right)^{-2/3} \right)^{-1}. \end{aligned} \quad (8)$$

The content of debris in water influences each separate indicator of the efficiency factor. Their impact on the total efficiency factor, therefore, requires practical and scientific estimates.

Materials and methods. The head-capacity curve of a pump gives an idea of the pump's capabilities and depends not only on the pumped liquid density but also on its viscosity. The higher the viscosity, the lower the head-capacity curve is constructed. Manuals present head-capacity curves for water pumps. Characteristics from a manual should therefore be recalculated (reconstructed) according to a particular method to pump the liquid with the viscosity different from the viscosity of water [9–11, 13] (*Hydraulic vehicle design guide (for construction norms and regulations SNiP 2.05.07-85) / Promtransniiproekt. Moscow: Stroizdat; 1988*).

The characteristics of the pump tested on water are recalculated to determine its indicators when pumping liquids with higher viscosity in accordance with GOST 6134 – 2007 “Rotodynamic pumps. Test methods”.

Figure 2 presents the nomogram from GOST 6134 – 2007 for determining the obtained water characteristic recalculation coefficient when pumping viscous liquid. Values presented in Figure 2 are obtained as a result of Hydraulic Institute Standards (HIS) testing.

The procedure of working with the nomogram is as follows: it is necessary to find the value corresponding to the optimal capacity on the lower scale of the nomogram (capacity Q , m^3/s) and go up to the value of the head (per one pump stage) in the optimal capacity mode, then move horizontally (left or right) up to the required viscosity value, and after that in an upward direction again to the intersection with the curves of the recalculation coefficient C_Q , $C_H = f(Q)$, as shown by the dotted line in Figure 2. The points of the dotted line and the indicated dependencies intersection will determine the values of coefficients C_Q , C_H .

Pump capacity, head, and efficiency factor for viscous liquid is calculated by the formulae:

$$Q_{\text{viscous}} = QC_Q; \quad (9)$$

$$H_{\text{viscous}} = HC_H. \quad (10)$$

An example of constructing a head-capacity curve is presented below. Initial data for viscous liquid at a 260 m horizon are as follows: the normal water inflow is 426.5 m³/h, the maximum water inflow is 527 m³/h.

Table 1. Rated data of the 260 m horizon external network
Таблица 1. Данные расчета внешней сети горизонта 260 м

Indicator	Point number				
	1	2	3	4	5
$Q, \text{ m}^3/\text{h}$	0	75	150	225	300
$H, \text{ m}$	267.00	288.21	351.83	457.86	606.30

The design head of multistage centrifugal pump TsNS(K) 300-360 installed at a given horizon is calculated using the following formula (with certain restrictions):

$$H_d = \frac{H_g}{\eta} = \frac{H_{\text{horizon}} + H_{\text{suction}} + H_{\text{overwind}}}{\eta}, \quad (11)$$

where H_g is the geodetic pressure head, m; H_{horizon} is the depth of the water level, $H_{\text{horizon}} = 260$ m; H_{suction} is the geodetic suction head, $H_{\text{suction}} = 2.5$ m; H_{overwind} is the overwind height, $H_{\text{overwind}} = 2.5$ m.

The calculations found that $H_d = 300$ m.

The characteristic of the external network for viscous liquid (for the lengthiest branch of the external network):

$$H_{\text{network}} = H_g + \frac{P_2 - P_1}{\rho_{\text{liquid}} g} + 0.0083 \times \left(\lambda_{\text{suction}} \frac{l_{\text{suction}}}{d_{\text{suction}}^5} + \lambda_{\text{pressure}} \frac{l_{\text{pressure}}}{d_{\text{pressure}}^5} + \frac{\sum \xi_{\text{suction}}}{d_{\text{suction}}^4} + \frac{\sum \xi_{\text{pressure}}}{d_{\text{pressure}}^4} \right) \frac{Q^2}{3600^2 \pi^2 g}, \quad (12)$$

where P_1, P_2 are the pressures in feed and receiver tanks, kPa; ρ_{liquid} is the density of the pumped liquid, t/m³; $\lambda_{\text{suction}}, \lambda_{\text{pressure}}$ is the friction factor of the suction and pressure pipelines respectively; $l_{\text{suction}}, l_{\text{pressure}}$ is the length of the respective pipelines, m, $l_{\text{suction}} = 164.85$ m; $l_{\text{pressure}} = 1730.28$ m; $d_{\text{suction}}, d_{\text{pressure}}$ is the diameter of the suction and pressure pipelines, m; $\sum \xi_{\text{suction}}, \sum \xi_{\text{pressure}}$ is a total of the coefficients of all local resistances in the suction and pressure pipelines.

The rated value of linear hydraulic resistance coefficients for the pipelines is found by the formula:

$$\lambda = \frac{0.0195}{\sqrt[3]{D}}, \quad (13)$$

where D is the diameter of the pipeline, m. Thus, $\lambda_{\text{suction}} = 0.0347$; $\lambda_{\text{pressure}} = 0.02878$.

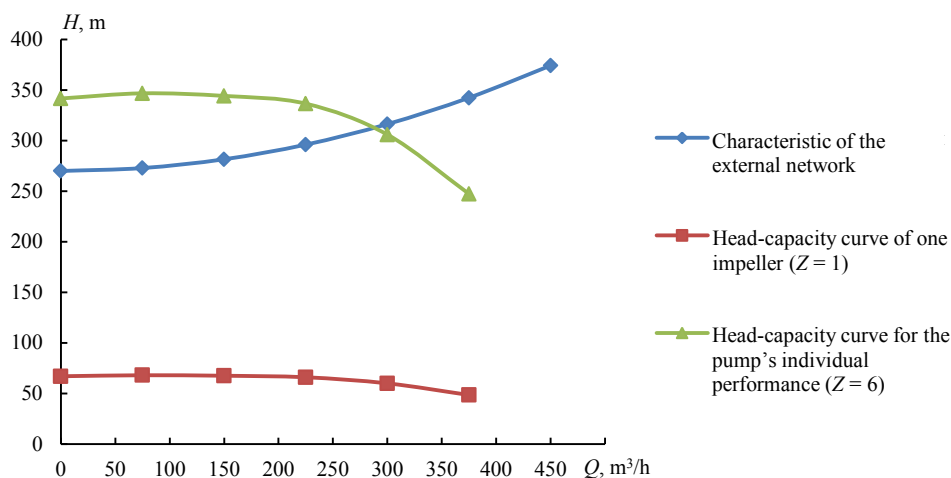


Figure 3. The head-capacity curve of the 260 m horizon for “clarified” water

Рисунок 3. Напорная характеристика горизонта 260 м при перекачивании «осветленной» воды

As a result, we get a dependency of the mine-drainage plant network for one pump per column: $H_c = 267 + 0.00377Q^2$.

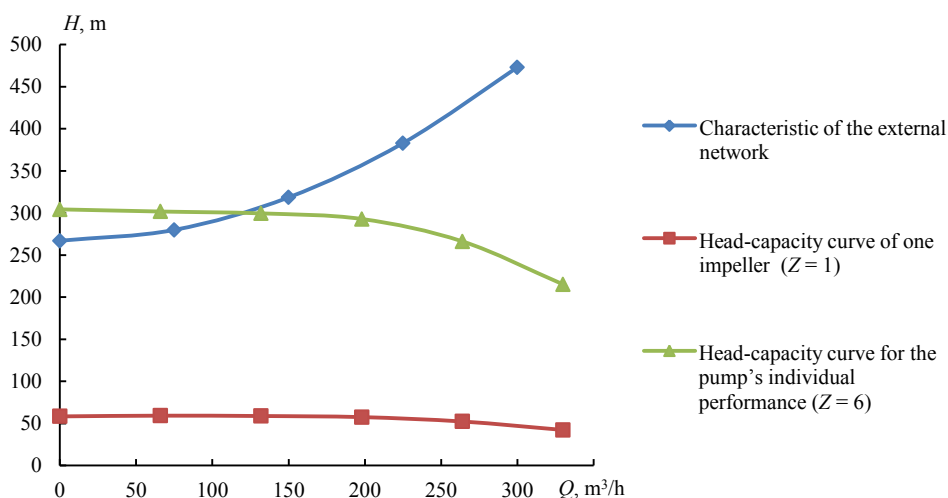


Figure 4. The head-capacity curve of the 260 m horizon for “unclarified” water

Рисунок 4. Напорная характеристика горизонта 260 м при перекачивании «неосветленной» воды

We summarize the calculation data for the external network in Table 1.

Further, correction factors are determined to recalculate the head-capacity curve of the TsNS(K) 300-360 pump operating on water.

According to the graph (Figure 3), at the point where external network and pump characteristics intersect, head and capacity values per one pump stage are as follows: $H_1 = 60$ m, $Q_1 = 290$ m³/h = 0.8 m³/s.

We determine the correction factors $C_Q = 0.88$; $C_H = 0.87$ through the nomogram (Figure 2).

Results and discussion. After the correction factors and external network data have been determined, the pressure characteristic of the 260 m horizon is reconstructed when pumping "unclarified" water (Figure 4). Pump performances are recalculated, and the results are summarized in Table 2.

Table 2. Recalculating the operating parameters of TsNS(K) 300-360 multistage centrifugal pump

Таблица 2. Пересчет эксплуатационных показателей насоса ЦНС(К) 300-360

Indicator	Point number					
	1	2	3	4	5	6
<i>When pumping water for one step</i>						
Q , m ³ /h	0	75	150	225	300	375
H , m	67.0	68.0	67.5	66.0	60.0	48.5
<i>When pumping water for six steps</i>						
Q , m ³ /h	0	75	150	225	300	375
H , m	341.70	346.80	344.25	336.60	306.00	247.35
<i>When pumping "unclarified" water for one step</i>						
Q , m ³ /h	0	66	132	198	264	330
H , m	58.3	59.2	58.7	57.4	52.2	42.2
<i>When pumping "unclarified" water for six steps</i>						
Q , m ³ /h	0	66	132	198	264	330
H , m	304.3	301.7	299.5	292.8	266.2	215.2

It should be noted that to construct the head-capacity characteristic curve (Figure 4), the calculations were carried out under the condition that the pump was new and not subject to wear. When operating on "unclarified" water [12–15], the real characteristic of the pump will rest even lower along the y-axis.

Conclusion. The research results are as follows:

- when the pump operates on "unclarified" water, the actual mode is beyond the optimal zone of the pump;
- if productivity falls, the operating time of all dewatering pumps increases by three times due to the constant water inflow;
- if the depth and productivity of the mine and therefore water inflows increase, the risk and threat of flooding increases (the time for pumping out normal water inflow is 20 hours, and the enterprise will require additional measures to increase the drainage capacity, which will lead to additional capital costs).

As a result of the calculation, it can be concluded that the multistage centrifugal pumps TsNS operation on "unclarified" water is unsafe and requires technical and technological measures to clarify the water before it enters the pumps.

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Received 6 April 2021

Information about authors:

Bulat M. Gabbasov – PhD (Engineering), associate professor of the Department of Mining Machines and Transport and Technological Complexes, Nosov Magnitogorsk State Technical University. E-mail: bulatg74@mail.ru

Anton I. Kurochkin – PhD (Engineering), associate professor of the Department of Mining Machines and Transport and Technological Complexes, Nosov Magnitogorsk State Technical University. E-mail: a.kurochkin@magtu.ru

Artur M. Mazhitov – PhD (Engineering), Associate Professor, Head of the Department of Mining Machines and Transport and Technological Complexes, Nosov Magnitogorsk State Technical University. E-mail: a.mazhitov@magtu.ru; <https://orcid.org/0000-0001-8219-4264>

Sergei N. Kornilov – DSc (Engineering), Head of the Department of Logistics and Transport Systems Management, Nosov Magnitogorsk State Technical University. E-mail: kornilov_sn@mail.ru

УДК 622.23.054.72

DOI: 10.21440/0536-1028-2021-6-13-23

Исследование внешних и внутренних рабочих процессов горных машин при работе на «неосветленной» воде в подземных условиях

Габбасов Б. М.¹, Курочкин А. И.¹, Мажитов А. М.¹, Корнилов С. Н.¹

¹ Магнитогорский государственный технический университет им. Г. И. Носова, Магнитогорск, Россия.

Реферат

Введение. В статье рассмотрены вопросы исследования горно-шахтного водоотлива, включающие изучение рабочих процессов центробежных насосных установок, обеспечивающих перекачку

«неосветленной» воды в тяжелых подземных условиях. Решение этих вопросов в настоящее время имеет высокую практическую и научную значимость.

Методология проведения исследования. Выполнены анализ и обобщение научно-практического опыта в области горно-шахтного водоотлива, рассмотрены режимы работы центробежных насосов и обозначены перспективные задачи исследования в данной области. Так, непрерывное поступление грунтовых вод в выработки шахты требует обеспечения бесперебойной работы насосных установок. Одним из самых распространенных типов водоотливных установок являются центробежные секционные насосы, которые при правильной эксплуатации в полной мере выполняют свои функции. Однако перекачка «неосветленной» воды требует разработки методики по определению оптимальных режимов работы центробежных насосов в данных условиях работы. **Результаты и их анализ.** Для обоснования эффективности предложенных решений и выводов представлены: исследования по определению зависимостей гидравлического, объемного и механического КПД от режимов работы насосов; анализ рабочих процессов, происходящих в центробежных насосах при работе их на «неосветленной» воде; методика и расчет напорной характеристики центробежного насоса типа ЦНС (центробежный насос секционный).

Область применения результатов. Результаты исследований, выполненных в работе, рекомендованы к внедрению для всех предприятий, ведущих подземные горные работы с применением шахтного водоотлива.

Ключевые слова: шахтные центробежные насосы; оптимальные режимы работы; коэффициент полезного действия; напорная характеристика; гидравлические потери; коэффициент быстроходности.

Работа выполнена при финансовой поддержке гранта Президента Российской Федерации № МД-3602.2021.1.5.

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Сведения об авторах:

Габбасов Булат Маратович – кандидат технических наук, доцент кафедры горных машин и транспортно-технологических комплексов Магнитогорского государственного технического университета им. Г. И. Носова. E-mail: bulatg74@mail.ru

Курочкин Антон Иванович – кандидат технических наук, доцент кафедры горных машин и транспортно-технологических комплексов Магнитогорского государственного технического университета им. Г. И. Носова. E-mail: a.kurochkin@magtu.ru

Мажитов Артур Маратович – кандидат технических наук, доцент, заведующий кафедрой горных машин и транспортно-технологических комплексов Магнитогорского государственного технического университета им. Г. И. Носова. E-mail: a.mazhitov@magtu.ru; <https://orcid.org/0000-0001-8219-4264>

Корнилов Сергей Николаевич – доктор технических наук, заведующий кафедрой логистики и управления транспортными системами Магнитогорского государственного технического университета им. Г. И. Носова. E-mail: kornilov_sn@mail.ru

Для цитирования: Габбасов Б. М., Курочкин А. И., Мажитов А. М., Корнилов С. Н. Исследование внешних и внутренних рабочих процессов горных машин при работе на «неосветленной» воде в подземных условиях // Известия вузов. Горный журнал. 2021. № 6. С. 13–23 (In Eng.). DOI: 10.21440/0536-1028-2021-6-13-23

For citation: Gabbasov B. M., Kurochkin A. I., Mazhitov A. M., Kornilov S. N. Investigating external and internal working processes of mining machines when operating on “unclarified” water in underground conditions. *Izvestiya vysshikh uchebnykh zavedenii. Gornyi zhurnal = News of the Higher Institutions. Mining Journal*. 2021; 6: 13–23. DOI: 10.21440/0536-1028-2021-6-13-23