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NUMERICAL SIMULATION OF THE UNSTEADY FLUID MOTION IN PIPELINES IN THE PUMP UNIT START-UP MODE

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From the experience of operating pipeline transport systems of various incompressible and compressible viscous flows, it is known that transportation conditions are constantly changing during their operation. In turn, this causes sharp fluctuations in the characteristics of such flows, which can be dangerous to pipeline equipment and personnel. In this regard, starting from the design stage, it is necessary to have convenient methods for calculating transient (unsteady) modes of flow movement, which allow determining possible pressure and flow fluctuations in different modes of operation of transport systems. Based on them, it is possible to obtain the necessary information for calculating and selecting the pipeline protection system against excessively high pressures, as well as for adjustment of the means of automatic control and protection of the systems under consideration. The paper proposes a simple and reliable mathematical model along with the numerical method for calculating the unsteady turbulent transport of liquid flows in pipelines in the pump installation start-up mode. The model is based on a system of quasi-one-dimensional equations for describing the motion of liquid in a pipe, supplemented by the empirical dependence of the value of hydraulic resistance on time and the Reynolds number. The main problems that arise when modeling the considered unsteady flows are stated. In particular, we discuss the possibility of using a simplified system of hydrodynamic equations, the choice of numerical solution methods, and the unsteadiness of the coefficient of hydraulic resistance for flows of the considered class. Typical dependencies of the pressure at the control point for the basic configuration of the pipeline and in the presence of a check valve in the system are demonstrated. The result obtained in this work summarizes the previous data of the authors regarding the possibility of using the applied approach to solve a broad class of problems related to unsteady fluid flows in pipelines.

KEY WORDS: unsteady liquid flows, pipeline hydraulic systems, numerical modeling, pump start

1. INTRODUCTION

It is known from the experience of pipeline transport systems for various types of incompressible and compressible viscous carriers that constant changes in the transport conditions are possible during the operation of such systems, and this, in turn, can cause sharp and often dangerous fluctuations in pressure and flow.

In this regard, even at the design stage, it is necessary to have convenient methods for calculating transient (unsteady) flows. Driving modes allow you to determine possible pressure and flow fluctuations in different modes of operation of transport systems. Based on these methods, it is possible to obtain the necessary information for calculating and selecting a pipeline protection system against excessively high pressures, as well as for calculating and setting up automatic control and protection systems for such systems [1–5].

2. DEVELOPMENT OF NUMERICAL ALGORITHMS FOR CALCULATING THE UNSTEADY MOTION OF HOMOGENEOUS FLOWS IN PIPELINES

As noted above, during the operation of industrial and, in particular, complex main pipeline systems, sharp pressure fluctuations can occur almost constantly, which sometimes leads to undesirable emergencies. Such transient (unsteady) operating modes occur when the main and auxiliary pumps are started and stopped, during a planned or emergency power outage, as well as when various control valves are switched on and off.

Full consideration of instability will ensure the stable operation of the entire transport system, and increase the reliability and durability of pressure pipelines and all hydro-mechanical equipment.

Currently, a sufficient number of works are known that relate to the solution of some of the problems formulated above. The approaches proposed in these works, as a rule, contain many assumptions, which in some cases may affect both the nature and magnitude of the determining hydrodynamic parameters.

In the most general form, the problem of the non-stationary motion of an incompressible homogeneous medium in the framework of the one-dimensional and isothermal approximation should be solved based on the general system of dynamical equations obtained, for example, in [6]. For a complete closure, the corresponding initial and boundary conditions must also be added to this system.

At the same time, our research has shown that several simplified approaches can also be used to solve several problems of interest related to transients, which also successfully allows us to model the flows of the class under consideration.

In this formulation, the general problem of the non-stationary motion of a homogeneous medium formulated in the framework of a one-dimensional problem is a special case of the system [6–8] and reduces to solving a system of two partial differential equations:

$$
\begin{gathered}
-\frac{\partial P}{\partial x} = \frac{\partial(\rho V)}{\partial t} + \rho\frac{\lambda|V|V}{2D},\\
-\frac{\partial P}{\partial t} = a^2\frac{\partial(\rho V)}{\partial x}.
\end{gathered}
\tag{1}
$$

Here V is the average velocity of the carrier medium over the cross-sectional area; P is pressure in the system; ρ is density of the carrier medium; a is sound propagation speed; λ

is a coefficient of hydraulic resistance; D is diameter of the pipeline.

Both the more general system of equations of motion [6] and the simplified system of equations (1) belong to the hyperbolic type and the same analytical and numerical methods can be used to solve them.

In other words, by setting the initial and boundary conditions (also knowing the law of changeρ, λ and a), the system of equations (1) can be solved in principle for various modes of operation of high-speed pipeline systems.

At the same time, it should be noted that the correct setting of boundary conditions and the choice of a convenient method for solving system (1) is most often associated with certain difficulties. Let us dwell in more detail on the method of solving this system of equations.

As is known, both analytical and numerical methods for solving systems of hyperbolic equations are widely used in the literature. Due to the nonlinearity of system (1), when using analytical methods of solution, it is assumed to perform a preliminary linearization of the initial equations. Most authors, concerning such problems, performed linearization of system (1) by replacing the term $\lambda|V|V$ with its constant value equal to the average value over the coordinate and time. In addition, when solving specific problems $\rho = \mathrm{const}$, is usually assumed, which excludes the possibility of phase transitions. The above assumptions lead to the fact that it is not always possible to achieve a satisfactory correspondence between the calculated and experimental data.

In this connection, there is a need to develop a universal method for solving the system (1). It is quite obvious that in such a situation it is possible to use only numerical methods of calculation. The most popular method for solving system (1) is still considered to be the method of characteristics.

After analyzing various numerical algorithms, we adopted explicit Lax and Lax–Wendroff difference schemes, which are well-established in the field of gas dynamics. In this setting, we were able to use the initial system of equations (1) to solve some problems in the field of not only the motion of a homogeneous fluid but also the hydro transport of various solid dispersed materials in pipes in the developed turbulence mode [6–8].

The purpose of this paper is, first of all, to demonstrate the possibility of using the proposed numerical calculation method for modeling the poorly studied problem of unsteady motion of a homogeneous liquid, namely, starting pumping units when electric power is turned on in the system. Before proceeding directly to a discussion of the features of using the proposed numerical algorithm for solving specific problems, we'll clarify one fundamental issue, which until now has not been practically discussed in the literature.

In all numerical calculations for unsteady flows, even for homogeneous mixtures, it was usually assumed that the coefficient of hydraulic resistance λ included in the determining system of equations (1) is determined at each calculation step strictly from the conditions of the stationary problem. Separate research known from the literature, for example [9, 10], have shown that such assumptions are, as a rule, not sufficiently justified.

The problem of determining the coefficient λ for conditions strictly corresponding to the real regimes of unsteady motion of one-phase mixtures is of independent interest and requires additional research. This paper is mainly devoted to assessing the influence of the non-stationary coefficient on solving concrete problems of fluid movement in pipelines.

3. THE HYDRAULIC RESISTANCE COEFFICIENT FOR UNSTEADY LIQUID FLOW IN PIPELINE

In this case, according the [9, 10] the value of the hydraulic resistance coefficient λ was determined from the first equation of system (1):

$$\rho \frac{\partial V}{\partial t} = -\frac{\partial P}{\partial x} - \lambda \frac{\rho V^2}{2D}. \qquad (2)$$

In this case, the flow velocity was determined by an induction flow meter, and the pressure P was determined by pressure sensors.

For such conditions during laminar acceleration flow, it was found in the paper that the ratio λ_N/λ_{ST} is uniquely related to the ratio of the instantaneous value of the Reynolds Re number to the value of this Reynolds number Re_∞, corresponding to the end of the acceleration process:

$$\frac{\lambda_N}{\lambda_{ST}} = 4 \sum_{n=1}^{\infty} \frac{1 - \exp(-\mu_n^2 4\nu t/D^2)}{\mu_n^2} \frac{\mathrm{Re}_\infty}{\mathrm{Re}}. \qquad (3)$$

Here λ_N and λ_{ST} are the hydraulic resistance coefficients for non-stationary and stationary motion respectively; μ are the roots of the Bessel function of zero order; ν is a kinematic viscosity of the liquid; $\mathrm{Re}_\infty = V_\infty D/\nu$ is the Reynolds number at the end of the acceleration process; t is time.

The experiments [10] performed on a series of large-scale stands with various liquids have shown that dependence (3) is well satisfied in the initial period of the acceleration section corresponding to the laminar flow regime. For a flow in a turbulent region, the direct use of dependence (3) becomes problematic. Despite this, the authors of [9, 10] derived that this dependence after partial modernization can also be used to describe the initial acceleration period (even in the turbulent region):

$$\frac{\lambda_N}{\lambda_{ST}} = 1 - \frac{1.6\left(1 - \dfrac{\mathrm{Re}}{\mathrm{Re}_\infty}\right)}{1 + \left(1 - \dfrac{\mathrm{Re}}{\mathrm{Re}_\infty}\right)^2}. \qquad (4)$$

Thus, using relation (3) or (4), it is possible to take into account the unsteady coefficient λ for accelerating fluid motion in the pipeline. The ratios thus obtained can be used in calculating the start-up (acceleration) modes of pumping units.

4. THE START-UP MODE OF PUMPS FOR HYDRAULIC PIPELINE SYSTEM: CALCULATION EXAMPLE

Let us take a closer look at this problem. It is clear that when the power is turned on, the pump gradually reaches its performance. In this case, as for other problems of this kind, the question of setting boundary conditions remains fundamental. Setting the boundary conditions together with the system of equations (1) will allow us to formulate an algorithm for solving the problem considered here in a closed form.

In the most general form, the qualitatively studied transient process can be traced in Fig. 1. Assume that at a given time the pump speed is equal n to, and the operating point

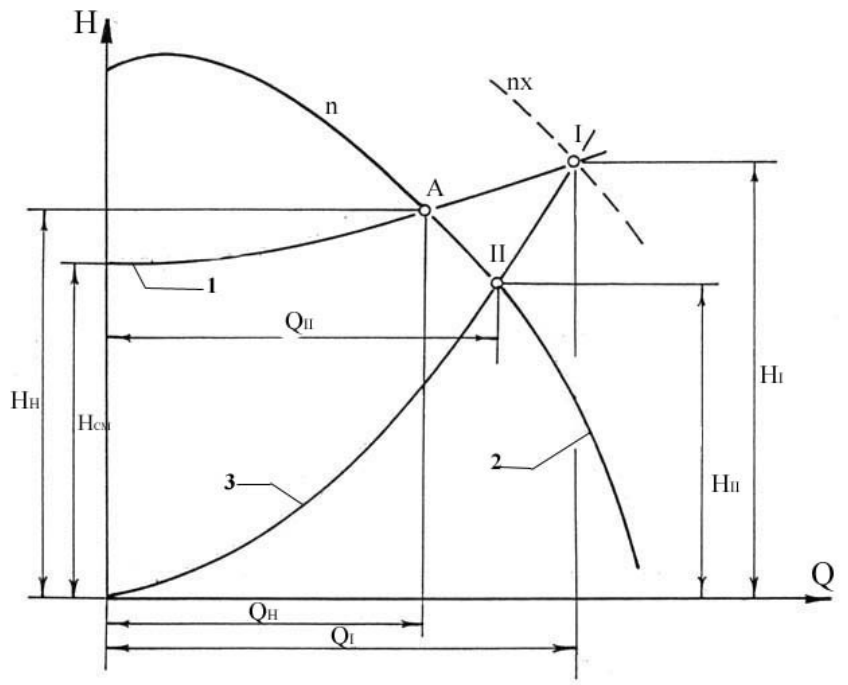

Fig. 1. On calculating the boundary conditions for starting pumping units:
1 — installation characteristics; *2* — pump characteristics; *3* — parabola of similar modes

of the system corresponds to point A. The pump head is equal to H_H, and the flow rate is Q_H. You need to determine the new pump speed n_x, if the system flow rate has become equal Q_I. Knowing Q_I the characteristics of the installation, you can determine the new operating point I of the system (see Fig. 1).

The characteristic of the pump at the initial speed must pass through this point n_x. To determine n_x, it is necessary to draw a parabola of similar modes through the point first and find point II of the intersection of this curve with the given pump characteristic. For points I and II, you can get the following relationship:

$$n_x = n\frac{Q_{\mathrm{I}}}{Q_{\mathrm{II}}} \quad \text{or} \quad n_x = n\sqrt{\frac{H_{\mathrm{I}}}{H_{\mathrm{II}}}}. \tag{5}$$

Thus, to solve the problem (starting the pump unit), it is enough to set the law of rotation speed change over time.

Currently, in the literature [1, 10] there are two fundamental approaches to determining the frequency characteristics of a pump.

Some authors [1] have performed special experimental studies not only of the time but also of the nature of changes in the pump speed when they reach the operating mode. Unfortunately, the number of pumps studied in this regard is very limited, and, therefore, approximate dependences for the value are more often used in calculations $n = f(t)$. For example, in [1], a fairly simple method for calculating this value is presented. According to [1], when the pump unit is switched on, the rotation of the rotor depends on the torque consumed by the pump M_H and the torque consumed by the pump engine M_{PE}.

Therefore, the basic equation of motion can be written as:

$$M_{PE} - M_H = I\frac{d\omega}{dt} = I\frac{2\pi}{60}\frac{dn}{dt}, \tag{6}$$

where I is the pump's flywheel moment, and ω is the rotational speed. Replacing in (6) the value $I = GD^2/(4g)$, where GD^2 is the swing moment of the pump unit rotor, we can finally obtain:

$$\frac{dn}{dt} = \frac{375}{GD^2}(M_{PE} - M_H), \tag{7}$$

By specifying the law of change M_{PE} and M_H in time, we can get the dependence from equation (7) $n = f(t)$. This, in turn, allows you to determine the value of the pressure P and velocity V values near the pump (in other words, the boundary conditions). for the case of starting pumping units discussed above.

Taking into account the now defined boundary conditions, we can solve the system of equations (1). In this case, the value of the pressure and velocity values near the pump (in other words, the boundary conditions) it can be determined according to the scheme proposed above.

Thus, the general problem of starting pumping units is formulated in a closed form and can be solved for both simple and branched pipelines. Without dwelling in detail on the specifics of calculating complex pipelines, we only note that the main conditions for coupling solutions are the conditions for maintaining flow rates and pressures at the network's nodal points. Of absolute interest is not only the calculation but also the comparison of the developed calculation scheme with the experimental data available in the literature. Unfortunately, there is practically no similar experimental data in the literature related to the launch of pumping units on large industrial and main pipeline systems. In this regard, we conducted a comparison with experimental data obtained on fairly large-scale laboratory stands.

The issue of starting pumping units was considered in more detail in the paper [10]. These studies were carried out by the author on the installation, the schematic diagram of which is shown in Fig. 2.

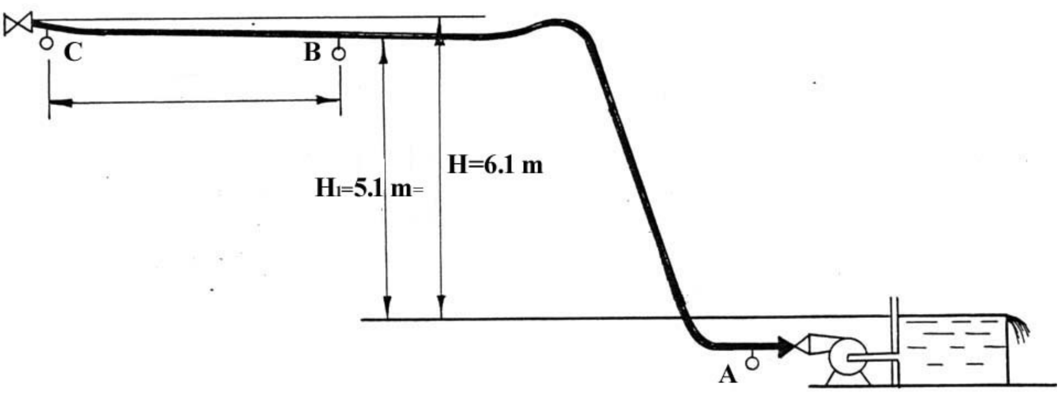

Fig. 2. Diagram of the experimental setup

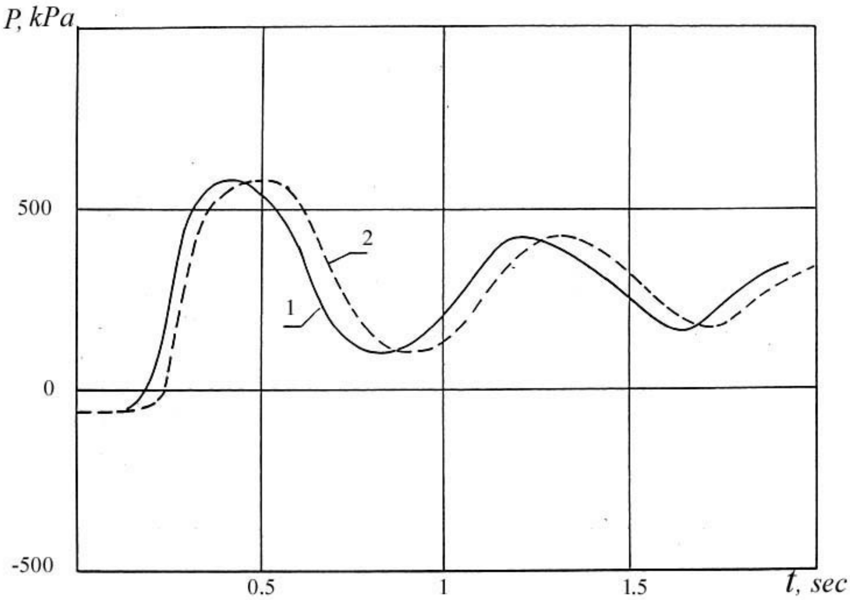

Fig. 3. Pressure dependence at point C from start-up time pump in a pipeline filled with water: *1* — experimental data; *2* — calculation

Fig. 3 shows a comparison of experimental and calculated data when starting a pump in a pipeline filled with water when the valve is closed at point C. The nature of the pressure change in the pipeline can be explained as follows. The pressure generated by the pump propagates at a speed along the pipeline. Taking into account that the valve is completely closed at the end of the pipeline (point C), the incoming pressure wave with the coefficient is reflected near this valve $r_r = 1$. The reflected wave begins to propagate in the direction of the pump. If there is no check valve in the water intake with atmospheric pressure, the secondary pressure shock wave is reflected with a coefficient $r_b = -1$. The resulting negative pressure wave propagates again to the end of the pipeline, etc. The pressure change in the pipeline section before the valve closes at its end occurs along a curve close to a sinusoid, where the impact phase is traced $\tau_* = 2l/a$. Here l and a are the length of the pipeline and the speed of sound, respectively.

As can be seen from Fig. 3, there is a fairly good match between the experimental and calculated data, and some divergence of the curves in phase can be explained by the inaccuracy of setting the acceleration time of the pump unit.

The picture of the transition process under consideration will be somewhat less clear if there is a non-return valve near the pump. In this case, the wave reflected from the valve closed at the end of the pipeline, almost equal to the pressure developed by the pump when approaching the non-return valve, closes it, after which a closed system is formed without communication with the atmosphere. In such a system, it follows from experience and calculations; the pressure fluctuation occurs along a curve that has only positive amplitude. The importance of the result obtained is since the secondary positive pressure wave, which has approached from the closed valve to the non-return valve, reflects from it, in this case, does not lead to a doubling of pressure and takes the place of a half-cycle with negative

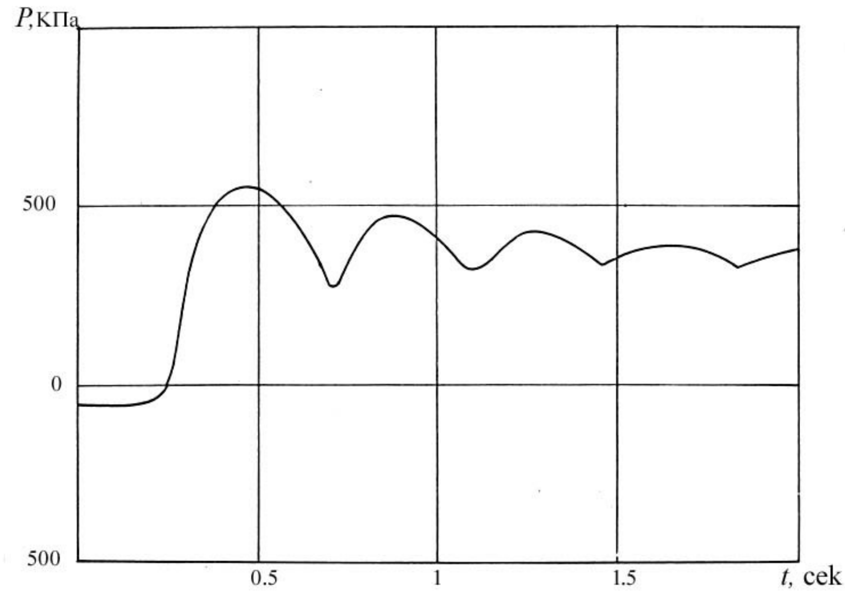

Fig. 4. Pressure dependence (at point C) from start-up time pump in a pipeline filled with water (there is a check valve near the pump)

amplitude (Fig. 4).

The results obtained give some information about the feasibility of placing fittings on such water pipelines. It is clear that safety valves must be installed near, for example, a gate valve (closed end of the pipeline) to avoid shock pressures (if the latter exceeds the norm) and the possibility of vacuum formation. The latter circumstance may further contribute to the creation of an emergency, since in this case the flow continuity may break.

5. CONCLUSION

In conclusion, we note that the specific example considered here of a non-steady mode of fluid movement (starting pumping units) to a certain extent once again proves the possibility of using the algorithm proposed in the paper for calculating various types of transient processes in simple and complex pipelines. The result obtained here to a certain extent generalizes our earlier data [8] on the possibility of using this approach to solve a wide class of problems of unsteady fluid motion, in particular, hydraulic impact, accelerating and decelerating fluid motion, etc. In addition, as we showed earlier, this approach can also be used to model the movement of slurry in pipes. This approach helps also get the necessary information for calculating and selecting a system for protecting pipelines from excessively high pressures, as well as for calculating and adjusting automatic control systems and protection of such systems.

REFERENCES

[1] J. A. Fox, *Hydraulic analysis of unsteady flow in pipe*. London: Red Globe Press, 1977. DOI: https://doi.org/10.1007/978-1-349-02790-3

[2] E. M. Al-Safran, *Applied multiphase flow in pipes and flow assurance*, J. P. Brill, Ed.

Richardson: Society of Petroleum Engineers, 2017.

[3] V. V. Tarasevich, "Steady flows and unsteady ones in pipes: models link and quasistationarity hypothesis," *E3S Web of Conferences*, vol. 102, pp. 01 007 (1–7), 2019. DOI: https://doi.org/10.1051/e3sconf/201910201007

[4] G. Vataj and X. Berisha, "Modeling of transient fluid flow in the simple pipeline system," *ARPN Journal of Engineering and Applied Sciences*, vol. 13, no. 13, pp. 4160–4166, 2018.

[5] T. Wiens and E. Etminan, "An analytical solution for unsteady laminar flow in tubes with a tapered wall thickness," *Fluids*, vol. 6, no. 5, pp. 170 (1–14), 2021. DOI: https://doi.org/10.3390/fluids6050170

[6] S. I. Kril' and V. P. Berman, "Equations of turbulent gas/solids flow," *International Journal of Fluid Mechanics Research*, vol. 27, no. 1, pp. 43–55, 2000. DOI: https://doi.org/10.1615/interjfluidmechres.v27.i1.40

[7] E. Bournaski and V. Berman, "Unified method of numerical simulation of one-phase and two-phase solid-liquid non-steady turbulent pipeline flows," in *6th IAHR Europe Congress*. Warsaw, Poland: IAHR, 2020, pp. 165–166.

[8] V. Berman, S. Kril, and E. Bournaski, "Some examples of numerical modeling of non-steady liquid flows in simple and branched pipelines," *Electronic Journal of Polish Agricultural Universities*, vol. 23, no. 4, pp. 1–12, 2020. DOI: https://doi.org/10.30825/5.ejpau.194.2020.23.4

[9] B. S. Baibikov, A. M. Prudovsky, and O. F. Oreshkin, "Hydraulic characteristics of straight cylindrical pipelines and restrictive inserts with accelerating flow of incompressible fluid," *Proceedings of Hydroproject*, no. 91, pp. 66–78, 1983.

[10] O. F. Oreshkin, "Study of unsteady fluid flows in pipes in relation to emergency cooling systems of nuclear reactors," Ph.D. dissertation, V. V. Kuibyshev Moscow State University of Civil Engineering, Moscow, 1981.

ЛІТЕРАТУРА

[1] Фокс Д. А. Гидравлический анализ неустановившегося течения в трубопроводах. — Москва : Энергоиздат, 1981. — 248 с.

[2] Al-Safran E. M. Applied multiphase flow in pipes and flow assurance / ed. by Brill J. P. — Richardson : Society of Petroleum Engineers, 2017. — 358 p.

[3] Tarasevich V. V. Steady flows and unsteady ones in pipes: models link and quasistationarity hypothesis // E3S Web of Conferences. — 2019. — Vol. 102. — P. 01007 (1–7).

[4] Vataj G., Berisha X. Modeling of transient fluid flow in the simple pipeline system // ARPN Journal of Engineering and Applied Sciences. — 2018. — Vol. 13, no. 13. — P. 4160–4166.

[5] Wiens T., Etminan E. An Analytical Solution for Unsteady Laminar Flow in Tubes with a Tapered Wall Thickness // Fluids. — 2021. — Vol. 6, no. 5. — P. 170.

[6] Kril' S. I., Berman V. P. Equations of turbulent gas/solids flow // International Journal of Fluid Mechanics Research. — 2000. — Vol. 27, no. 1. — P. 43–55.

[7] Bournaski E., Berman V. Unified method of numerical simulation of one-phase and two-phase solid-liquid non-steady turbulent pipeline flows // 6th IAHR Europe Congress. — Warsaw, Poland : IAHR. — 2020. — P. 165–166.

[8] Berman V., Kril S., Bournaski E. Some examples of numerical modeling of non-steady liquid flows in simple and branched pipelines // Electronic Journal of Polish Agricultural Universities. — 2020. — Vol. 23, no. 4. — P. 1–12.

[9] Байбиков Б. С., Прудовский А. М., Орешкин О. Ф. Гидравлические характеристики прямых цилиндрических трубопроводов и ограничительных вставок при разгонном течении несжимаемой жидкости // Сборник научных трудов Гидропроекта. — 1983. — № 91. — С. 66–78.

[10] Орешкин О. Ф. Исследование нестационарных потоков жидкости в трубах применительно к системам аварийного охлаждения ядерных реакторов : дис. ... канд. наук ; Московский инженерно-строительный институь им. В. В. Куйбышева. — Москва, 1981. — 182 с.

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Чисельне моделювання нестаціонерного руху рідини в трубопроводі в режимі пуску насосного агрегату

З досвіду експлуатації систем трубопровідного транспорту різних типів нестисливих і стислих в'язких потоків відомо, що при їх роботі умови транспортування постійно змінюються. Своєю чергою, це викликає різкі коливання характеристик таких потоків, що може становити небезпеку для обладнання трубопроводу та персоналу. У зв'язку з цим ще на етапі проєктування необхідно мати зручні методи розрахунку перехідних (нестаціонарних) режимів руху потоків, які дозволяють визначати можливі коливання тиску й витрати в різних режимах роботи транспортних систем. На їх основі можна отримати необхідну інформацію для вибору системи захисту трубопроводу від надмірно високих тисків, а також для розрахунку та налаштування засобів автоматичного керування й захисту розглянутих систем. У статті запропоновано просту та надійну математичну модель, а також чисельний метод розрахунку нестаціонарного турбулентного транспортування потоків рідини в трубопроводах у режимі пуску помпової установки. Модель базується на системі квазіодновимірних рівнянь для опису руху рідини в трубі, доповненій емпіричною залежністю величини гідравлічного опору від часу й числа Рейнольдса. Обговорено основні проблеми, які виникають при моделюванні таких нестаціонарних течій. Зокрема, розглянуто можливість використання спрощеної системи гідродинамічних рівнянь, вибір чисельних методів розв'язання та врахування нестаціонарності коефіцієнта гідравлічного опору для течій розглянутого класу. Продемонстровано характерні залежності тиску в контрольній точці для базової конфігурації трубопроводу та при наявності в системі зворотного клапану. Отриманий у цій роботі результат узагальнює попередні дані авторів щодо можливості використання застосованого підходу для розв'язання широкого класу задач, пов'язаних із нестаціонарними течіями рідини в трубопроводах.

КЛЮЧОВІ СЛОВА: нестаціонарна течія рідини, трубопровідна гідросистема, чисельне моделювання, пуск насоса