Hydrodynamic analysis of the efficiency of a pressure self-stabilizer work under conditions of hydraulic shock occurrence and distribution

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In this article, findings obtained from simulation of the hydraulic shock impact on a piping system and assessment of the stress strain behavior for a pipeline section are presented, and a technique for evaluation of the dynamic impact of the environment on the pipeline strength is suggested. The hydraulic shock action and its causes are considered. A description of the hydraulic shock and its physical model, and a method for determination of the hydraulic shock value and the velocity of its propagation in the medium are presented. Adverse effects on the piping system caused by the hydraulic shock impact are described. Using the hydrodynamic simulation on a hydraulic shock formation and propagation model, the operation mode based on dissipative damping effects and the efficiency of using a pressure self-stabilization device for elimination of adverse effects of hydraulic shock have been proposed and shown by the authors. As follows from the simulation results, if a pressure self-stabilization device is installed in the piping system, the dynamic load on the system decreases more than 13 times. Proven by field tests, these results do not contradict to the commonly known physics.

Piping systems used in oil production, nuclear power industry, chemical industry and elsewhere are the areas with an emphasis on particular attention and requirements to the durability and safety. Hydraulic shock is a primary dynamic action that leads to the destruction of the piping system and its equipment. The basic causes of hydraulic shock [1–3] include:

– Abrupt stop or failure of the pump;
– Uncontrolled operation of the shutoff devices; and
– Transient conditions (switch on/off of pumps concurrently operating in the system).

Hydraulic shock is a pressure surge caused by a sudden stop of the flow or rapid changes in the flowrate. As a result of a hydraulic shock, the pipeline, valves, fittings, supports and other components of the system are exposed to dynamic load.

The hydraulic shock problem was described in the late 19th century by N.E. Zhukovsky. In his works, he gave an explanation and a definition of hydraulic shock, and suggested a technique for the calculation of the pressure buildup and shock wave velocity [4]. He used the equations of motion and continuity to describe the physical effects of hydraulic shock. Those equations were solved with the following assumptions:

– Laminar flow of liquid in the pipeline;
– Absolutely rigid pipeline walls; and
– One-dimensional flow.

As a result of the transformation of the equations subject to the above assumptions, the formula given below was obtained

$$\Delta p_{UD} = \rho \cdot c \cdot \upsilon, \tag{1}$$

where: ρ is the fluid flow density;
c is the shock wave velocity (sound speed) in the given medium;
υ is the fluid flowrate (before the closure of the shutoff device); and
Δp_{UD} is the maximum pressure buildup caused by the hydraulic shock.

In his work [4], N.E. Zhukovsky describes methods for determination of shock wave velocity proven by experimental results presented in [9]. The shock wave velocity equation derived by him is shown below

$$c = \frac{1}{\sqrt{\rho \cdot \beta + \frac{2pr}{\delta E}}}, \tag{2}$$

where: β is the compressibility factor of the fluid;
r is the internal radius of the pipe;
δ is the thickness of the pipe's wall; and
E is the elastic modulus of the pipe's material.

If an absolutely rigid pipe material is considered, then

$$c = \frac{1}{\sqrt{\rho \cdot \beta}}. \tag{3}$$

Hydraulic shock may cause destruction of the pipeline's integrity, which, in turn, results in fluid leakage, biological contamination of the soil, loss of pressure, loss of profit, extra charges on the equipment/pipeline replacement and environmental reclamation.

A considerable number of design solutions and methods has been proposed to prevent the occurrence of such accidents. For the most part, however, the aforementioned design solutions require additional space or lack sensitivity to the shock-and-wave action (for example, relief valve), or require additional power sources not always available. The authors of this work suggest a new method and a completely new design that does not require additional power sources and enables to damp the critical pressure of hydraulic shock by using the pressure self-stabilization effect [5]. This design has proven its efficiency as a most versatile, compact and least expensive solution for prevention of the adverse impact of hydraulic shocks.

The self-stabilization principle is based on the following dissipative damping effects:

– Hydraulic shock wave dissipation using a group of holes;

– Flow section geometry modification enabling the controllable modification of the volume; and

– Residual damping of pressure waves in the rebound chambers due to compliance of the piston and compressibility of the fluid itself.

The efficiency of the hydraulic shock damping using a pressure self-stabilization device was determined in a simulation experiment on the equipment of Power Plant Research and Test Center in Kashira. Hydrodynamic simulation results obtained from that experiment have been proven in the field on a pilot plant of LUKOIL Co.

The hydrodynamics of the pipeline and pressure self-stabilization device were studied using a code based on the numerical solution of partial differential equations describing the behavior of the liquid flow and acoustic effects in compressible and incompressible media.

In the paper, the simulation results and the data of hydraulic shock wave propagation in the pipeline with an installed pressure self-stabilization device of a rated diameter DN10 are presented to determine the pressure fluctuation at the cross-section before and after the self-stabilizer.

The pressure self-stabilizer model SSD.A.T.010 compliant to the standard requirements [7] is used for the simulation. The rated diameter was 10 mm. The general view of the pressure self-stabilization device is shown in Figure 1.

The simulation was made subject the following conditions and assumptions:

– Simulation area (Fig. 2) was defined as a 90° angular segment in the flow section of the pressure self-stabilizer with adjacent flow sections of the pipeline. Such assumption is valid, if the axial symmetry principle is used, so that the program code enables reconciliation between the equation solution results for the cells located in both planes of symmetry;

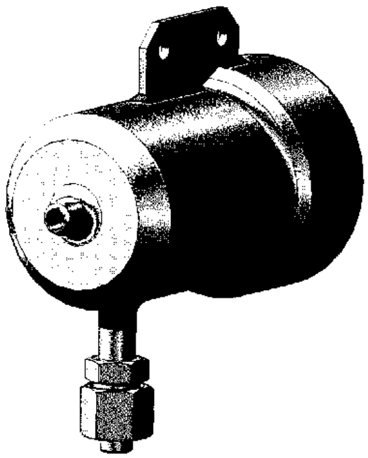

Fig. 1. *General view of the pressure self-stabilization device* DN10

– Length of each adjacent part was 100 mm to avoid the influence of the simulation area boundaries;

– In the description of dissipative damping effects, movement of each piston was considered as a free motion with one degree of freedom (in the longitudinal direction);

– Medium material was supposed to be the water at a temperature of 20°C. Initial parameters of the working medium:

Temperature, T .... 20 °C ,

Density, ρ_0 .... 968,7 kg/m^3 ,

Compressibility factor, β .... 5.8·10–10 1/Pa ,

Speed of sound in the medium (according to (3)), c_0 .... 1330 m/s;

– Pipeline material was supposed to be absolutely rigid that corresponds to the critical simulation conditions (maximum increase of pressure for the given parameters), i. e. the formula (3) was used for calculation of the speed of sound, c_0, in the medium;

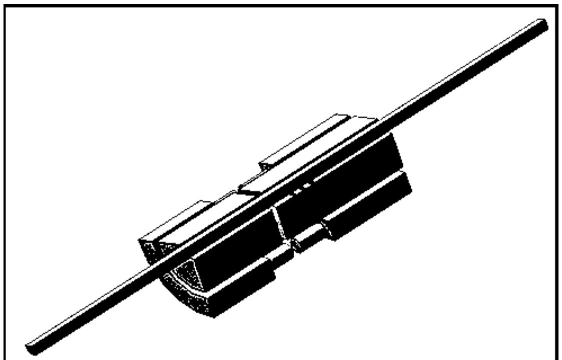

Рис. 2. *Расчетная область*

Fig. 2. Simulation area

– Initial (absolute) pressure, p_0, in the system equal to 8 MPa was chosen as that for the upper pressure boundary conditions of testing the self-stabilization device, that is, corresponding to the critical simulation conditions;

– Speed of sound at the input of the simulation area was supposed to be 5 m/s to avoid the occurrence of flow separation (vacuum) zones and local cavities ($\Delta p_{UD \geq} p_0$) in the non-stationary simulation environment;

– Apart from taking the medium compressibility factor into account, the variation of density with pressure was to have been determined to correctly describe the hydraulic shock occurrence in the simulation area (caused here by the abrupt closure of the shutoff device). Hence, the Tait's equation of state [8] establishing a relationship between the fluid density and the pressure was used to describe the fluid's behavior

$$\rho = \rho_0 \cdot \sqrt[n]{1 + \frac{n(p - p_0)}{\rho_0 \cdot c_0^2}}. \qquad (4)$$

where n is the fluid property factor equal to 7.15 (for water);
p_0 is the initial (absolute) fluid pressure;
$(p - P_0)$ is the fluid pressure variation;
c_0 is the speed of sound in the medium with a density of ρ_0;
ρ_0 is the fluid density at a pressure of p_0; and
ρ is the fluid density at a pressure of p.

Settings selected for finite element discretization are listed below:

- Characteristic form of the elements - hexahedron;
- Characteristic dimension of an element – 0.5 mm;
- Number of prismatic layers - 1;
- Prismatic layer thickness – 0.1 mm.

Thus, a finite element grid that had been generated included 410600 elements and 464600 nodes.

Hydrodynamic analysis includes two steps:

- Steady-state analysis, from which a steady-state flow is obtained for the piping system with a self-stabilization device installed; and

- Transient analysis step that includes the occurrence and propagation of a hydraulic shock wave in the medium.

The purpose of the steady-state hydrodynamic analysis is to determine the parameters of the steady-state flow when the convergence of results and the curves' approach to an asymptote is attained.

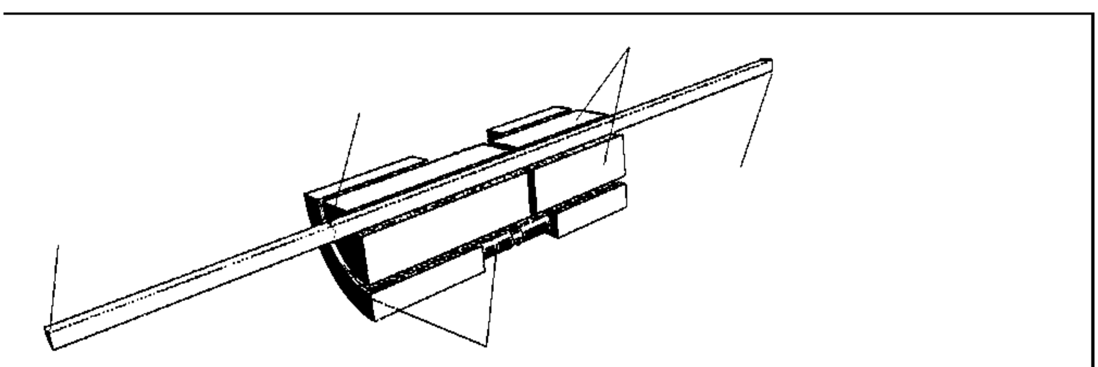

Fig. 3. Boundary conditions for the steady-state simulation

Скорость на входе – Input velocity
Сечение 2 – Cross-section 2
Плоскость симметрии – Plane of symmetry
Сечение 1 – Cross-section 1

Давление на выходе – Output pressure
Стенка – Wall

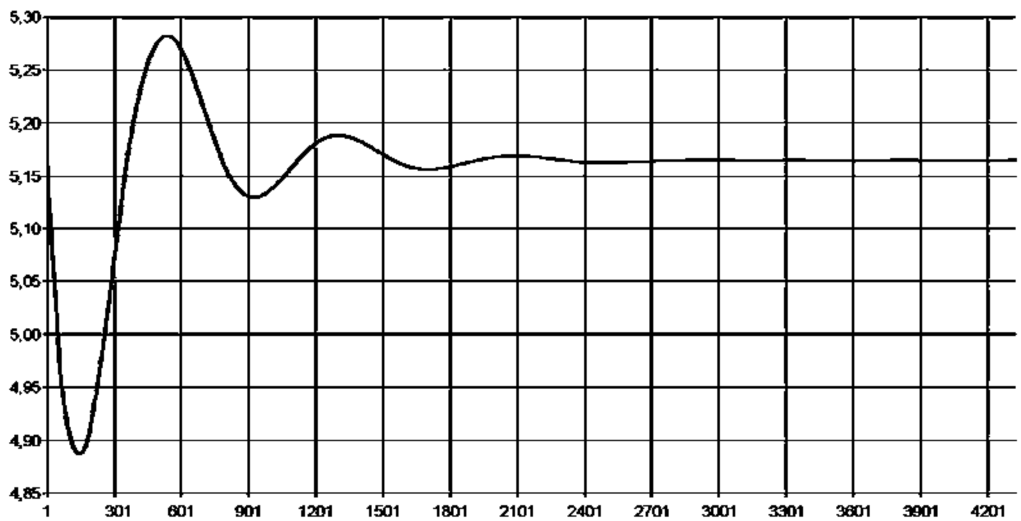

Fig. 4. Relationship between the flowrate fluctuation at the reference cross-section 1 and the number of iterations

Скорость – Velocity
Итерация – Iteration

The boundary conditions are shown in Figure 3. Flowrate at the input of the simulation area was 5 m/s. Output pressure was represented by the initial absolute pressure of the working medium, p_0 = 8 MPa.

Convergence of the solution was estimated using the objective function. Here, it was the velocity at the reference cross-section 1 and pressure at the reference cross-section 2.

Figures 4 and 5 show the flowrate and pressure fluctuation relative to the number of iterations, respectively.

It is obvious from the diagrams that the solution remains essentially unchanged after a certain number of iterations is reached that corresponds to the convergence criterion with pressure and velocity assuming steady-state values:

$$\upsilon = 5.16 \text{ m/s},$$
$$p = 8.002 \text{ MPa}.$$

All stationary flow parameters required for transient analysis have been obtained.

The purpose of the transient hydrodynamic analysis is to estimate the efficiency of the pressure self-stabilization device by determining the absolute pressure amplitude fluctuation at the reference cross-sections before and after the self-stabilizer.

Reference cross-sections of the pipeline are those where the duration of the fluid's dynamic impact on the walls is longest (the location of direct occurrence of the hydraulic shock and the pipeline cross-section located downstream the pressure self-stabilization device, that is, at the opposite side to the location of hydraulic shock occurrence).

To simulate the hydraulic shock occurrence, the boundary condition "output pressure" used in the steady-state simulation was substituted by a boundary condition called "wall". Changing the boundary condition in this way enables the reconstruction (simulation) of the fluid behavior after the shutoff device abruptly closes and reproduction of the hydraulic shock effect. Boundary conditions for the non-stationary simulation are shown in Figure 6.

According to the Courant-Friedrichs-Lewy criterion (CFL) [10], the minumum time increment is determined as

$$\Delta t_{\min} \leq C_{CFL} \cdot \left[\frac{h}{c_0}\right]_{\min},$$

where C_{CFL} is the Courant number;
h is the characteristic dimension of a finite element of the grid; and
c_0 is the speed of sound in the medium.

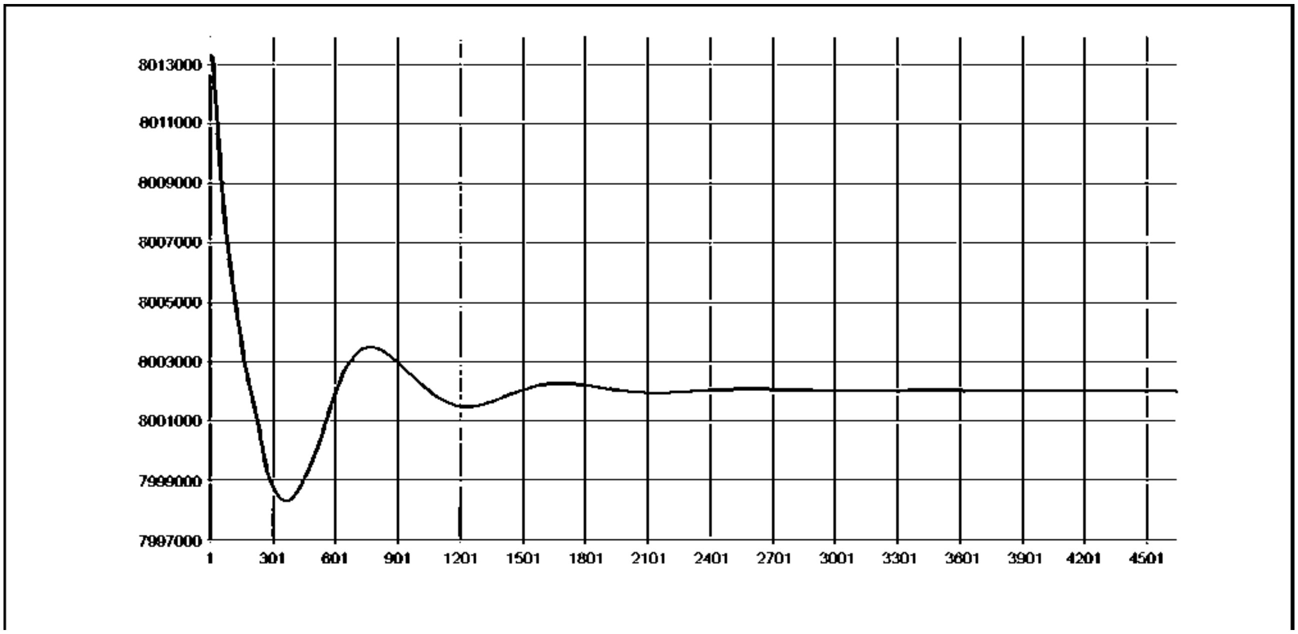

Fig. 5. Relationship between the pressure fluctuation at the reference cross-section 2 and the number of iterations

Давление, Па – Pressure, Pa
Итерация – Iteration

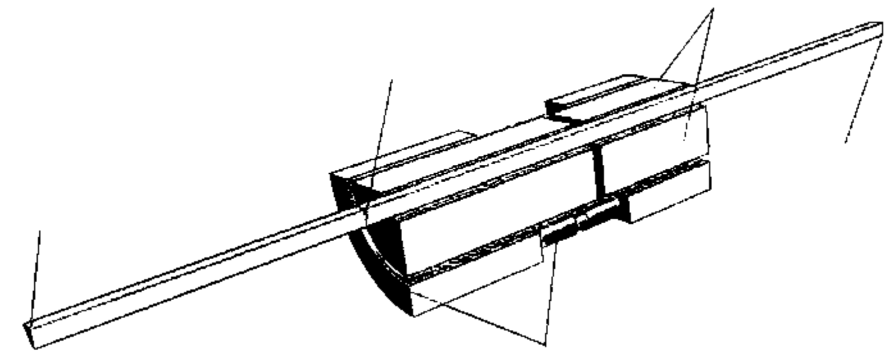

Fig. 6. Boundary conditions for the non-stationary simulation environment

Давление на выходе – Output pressure
Сечение 2 – Cross-section 2
Плоскость симметрии – Plane of symmetry
Сечение 1 – Cross-section 1
Стенка – Wall

Thus, the time increment, $\Delta t_{\min}$, was $3\cdot10^{-7}$ s.

The computation is finished if the following condition holds

$$\Delta p^{i}_{UD} \leq 0{,}1 \cdot \Delta p^{max}_{UD},$$

where i is the pressure buildup cycle number;
Δp^{i}_{UD} is the maximum pressure buildup corresponding to the i-th cycle; and
Δp^{max}_{UD} is the maximum pressure buildup corresponding to the 1st cycle.

This condition was selected to comply with the requirement of at least 10-fold pressure reduction to bring the pressure to the safe level stated in Technical Specifications for pressure self-stabilization device [7]. Such degree of pressure reduction downstream the self-stabilizer is sufficient to ensure the durability and safety of the pipeline and valves/fittings throughout the entire service life specified in the datasheet. In other words, the non-stationary simulation had been carried out until the absolute pressure fluctuation amplitude in the pipeline section upstream the self-stabilizer was 10 times lower than the fluctuation amplitude at the time of the hydraulic shock occurrence (Fig. 7).

According to curve *1* in Fig. 7, the values of Δp^{i}_{UD} and Δp^{max}_{UD} do not exceed 0.48 MPa (at i = 11) and 6.67 MPa, respectively, hence, $0.1 \cdot \Delta p^{max}_{UD}$ = 0.667 MPa, i.e. the simulation completion condition is met.

For assessment of the problem statement correctness, the value of pressure buildup, Δp_{UD}, determined using (1), was compared with the value of Δp^{max}_{UD}, obtained from the simulation.

Pressure buildup determined using (1):

Δp_{UD} = 6647994.4 Pa.

Pressure buildup obtained from the simulation:

Δp^{max}_{UD} = 6668025.1 Pa.

Inaccuracy was 0.3 %.

For additional verification of the results, the values of pressure fluctuation half-period determined using (6) were compared with those obtained from the simulation analysis (Fig. 8).

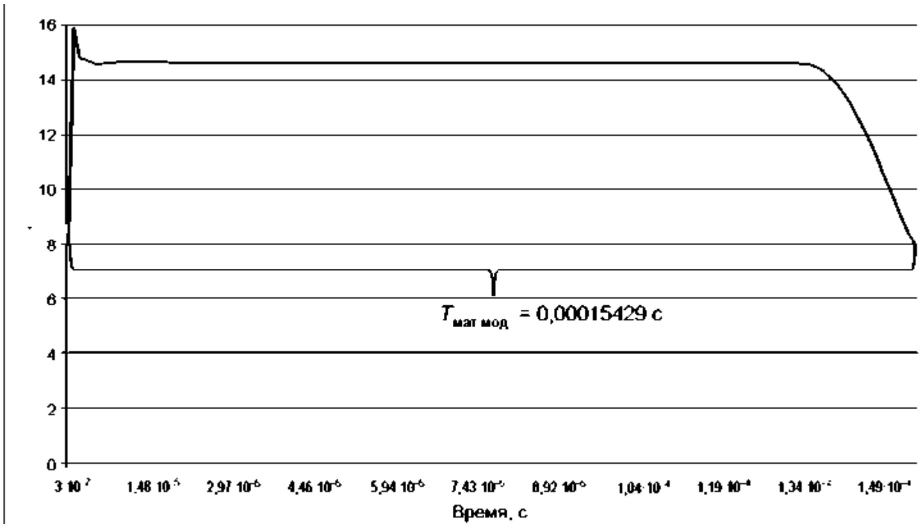

Fig. 7. Absolute pressure fluctuation diagram:
1 – for the cross-section 1 (hydraulic shock occurrence location);
2 – for the cross-section 2 (at the output of the pressure self-stabilization device)

Давление, Мпа – Pressure, MPa
Время, с – Time, s

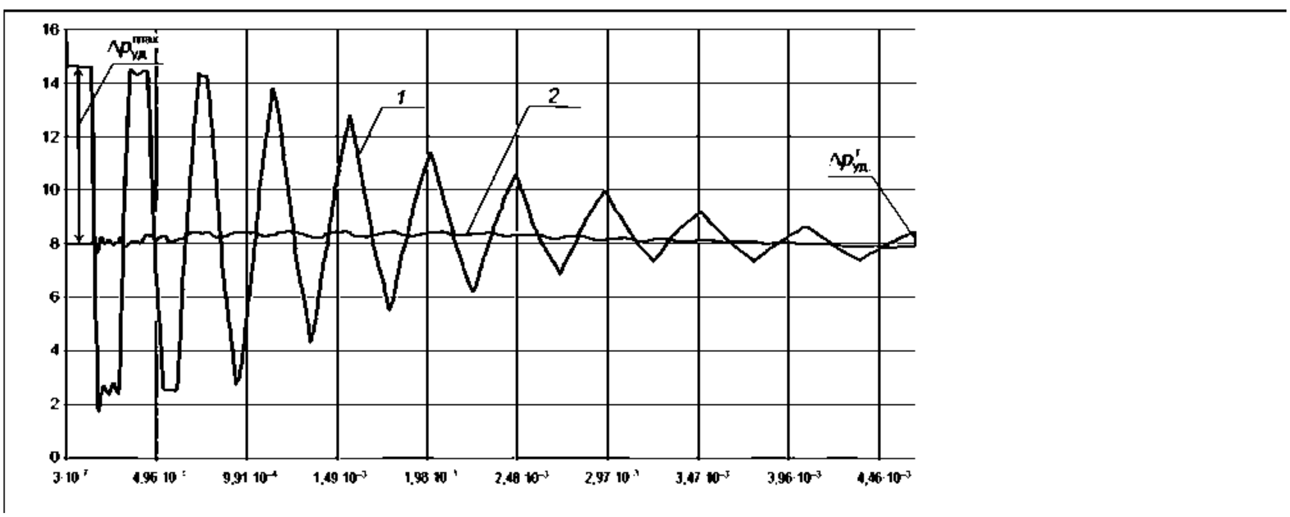

Fig. 8. Diagram of the first pressure buildup cycle for the reference cross-section 1 (hydraulic shock occurrence location)

Давление, Мпа – Pressure, MPa
Время, с – Time, s

Fluctuation half-period is the time required for the sonic wave to cover the distance equal to the double pipeline length to the location of nearest expansion.

It is obvious that if the pipeline length, *l*pipe. (100 mm), and sound speed in the medium, c_0 (1330 m/s), are known, the time required for the wave to cover the distance of 200 mm can be determined:

$$T_{\text{теор}} = \frac{l_{\text{тр.}}}{c_0}. \qquad (6)$$

where $T_{\text{теор}}$.= 0.0001504 s.

According to the diagram (see Fig. 8), the fluctuation half-period obtained from the simulation was

$T_{\text{sim.}}$ = 0.00015429 s,

Inaccuracy was 2.5 %.

Thus, it was proven by matching the values as described above that the problem statement in the simulation analysis and the computation results are consistent with the known physics and similar solutions obtained elsewhere [4, 6, 9]. Absolute pressure fluctuation diagrams shown in Figure 7 distinctively evidence the efficiency of the self-stabilization device. For the whole period of simulation, the maximum pressure buildup at the reference cross-section 1 was Δp^{max}_{UD} = 6.67 MPa (before the self-stabilizer), and the maximum pressure buildup at the reference cross-section 2 for the same period was, respectively, Δp_{CCD} = 0.48 MPa (after the self-stabilizer).

Hence, the efficiency of the self-stabilization device is determined by a relation $\Delta p^{max}_{UD} / \Delta p_{CCD}$ and equal to 13.89. Otherwise stated, the dynamic impact on the piping system piping system with a pressure self-stabilization device installed was reduced more than 13 times.

The findings of this study have been experimentally proven on the equipment of Power Plant Research and Test Center in Kashira and on a pilot plant of LUKOIL Co. Data obtained from the research were used for the development and industrial application of a series of pressure self-stabilization devices.

REFERENCES