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WATER HAMMERS IN SYSTEMS IMPORTANT FOR SAFETY OF NUCLEAR POWER PLANTS

Monograph

LAP LAMBERT Academic Publishing

The monograph presents the original designs of the authors in the field of modeling water hammers in systems important for safety of nuclear power plants. Various types of heat-hydrodynamic instability due to the inertia of the pressure-flow characteristics of pumps, accelerated closing of valves, transonic mode of one- and two-phase flows are considered as the main causes of water hammers.

MAINTENANCE

RELEVANCE

The heat-hydrodynamic instability phenomenon consists in appearance of conditions of self-oscillatory (self-sustaining) and/or aperiodic (pulse) change of heat-hydrodynamic flow parameters (pressure, rate, steam quality, etc.) in systems of the heat engineering equipment (pumps, armature, heat exchangers, etc.) and pipelines of thermal and nuclear power plants. Appearance of heat-hydrodynamic instability (HHI) conditions leads to additional hydrodynamic loads on the heat engineering equipment and pipelines (hydraulic impacts), increases a vibration state and affects reliability in operating, transitional and emergency operation. A failure to fulfil design operation functions and/or destruction of the heat engineering equipment and pipelines (HEP) occurs in extreme conditions of reliability-critical hydraulic impacts (CHI).

Problems of identification of the reasons and conditions for HHI and HHI effect on reliability of HEP systems are investigated for a long time (for example, [1 – 9] and others). In particular, V.A. Gerliga has investigated problems of identification of HHI conditions in steam-generating channels, the boiler volume of steam generators and the heat-exchanging equipment [4]. A.V. Korolev has studied problems of identification of conditions of hydraulic impacts in pipelines with two-phase flows [5]. V.I. Skalozubov has studied problems of identification of conditions of thermo-acoustic instability of the coolant in the nuclear reactor core and its effect on integrity of the fuel claddings [6]. Problems of identification of conditions of aperiodic HHI and hydraulic impacts on working elements of armature of nuclear power facilities, etc. were investigated in [6, 7].

However problems of identification of the reasons and conditions of CHI caused by hydrodynamic instability in pipeline systems with force pumps are insufficiently investigated. So, the newest Korolev's works [8, 9] defines resonant effects of coincidence of the self-resonant pipeline frequency and disturbing frequency of working pump hydrodynamic parameters as a generating mechanism of oscillatory hydrodynamic instability. The main restrictions of practical application of this approach are that resonant effects are extremely special case of HHI conditions and do not cover all possible operating, transitional and emergency modes of force pumps. Besides, there are no sufficient substantiations of CHI on the working pump elements caused by resonant effects in [8, 9].

Specific features of flow (network) pump characteristics, i.e. dependences of a hydrodynamic pressure on a flow rate can be the defining factors for hydrodynamic instability and critical (for pump reliability) hydraulic impacts in the broad range of operating parameters. It defines relevance of this work.

The pressurizer is the safety related system at nuclear power plants with WWERs, and it is designed [1]:

To generate a pressure in a reactor loop when reactor plant is started up,

To hold pressure within the prescribed limits in the normal operational modes and when the reactor is shutdown,

To limit pressure fluctuations in the transitional modes at reactor plant.

In case of malfunction and accidents, the pilot-operated safety valves (POSV) of pressurizer provide:

Pressure release into a reactor loop when limiting values are exceed (16.0 MPa);

Management of accident processes in the "release – feed" mode of a reactor loop.

The pressurizer of WWER-1000 is a vertical cylindrical vessel with a length of 13.5 m and throat diameter of 3.0 m. The pressurizer's body (thickness of 165 mm) consists of four shells and two bottoms. It has branch pipes and unions to connect the pipelines, devices and pulse lines. The pressurizer design provides normal operation in case of the maximum design earthquake of 9 points (MSK-64) and the loadings because of the reactor coolant pipe breaking [1].

The provided depressurization of pressurizer in case of malfunction and accidents defines possibility of the water hammers (WH) on the pressurizer's body and elements. WHs are followed by high-amplitude hydrodynamic loads on elements of pressurizer system that can exceed limiting values and lead to inadmissible damage of pressurizer's elements (especially in junction of branch pipes and unions) under certain conditions. Thus, in case of WHs at intensive filling of pressurizer, the experimental pressurizer analogue stands have registered pressure pulses with amplitudes by 2 times exceeded normal pressure in the equipment [2 – 4]. In full-scale pressurizers of WWER we can expect still greater relative amplitudes of WH pressure pulses as in case of emergency opening of the pressurizer's POSV, pressurizer is filling at differential pressure to environment about 15,0 MPa.

For determination of the maximum amplitudes of WH pressure ΔP_{gm} the known Joukowski formula [3] is traditionally used for different systems / the equipment (see, for example, reviews [5, 6]):

$$\Delta P_{\mathrm{gm}} = \rho \Delta v c, \tag{1}$$

Where ρ is flow density; Δν is change of flow rate before and after WH; c is sonic speed in equipment/ pipeline metal.

The authors of [4] have also applied formula (1) to assess ΔP_{gm} for filling of the WWER-440 pressurizer experimental model. However, such approach is not well-substantiated for modelling of WH conditions and consequences in full-scale WWER pressurizers as the formula (1) does not consider specific features / effects in case of WH generation (for example, an operating filling level of pressurizer, considerable differential pressure in pressurizer and containment in case of WH caused by opening of the pressurizer's POSV, hydraulic characteristics of elements of pressurizer system, the coolant charging in pressurizer, etc.).

Therefore, development of a method for determination of WH conditions and consequences in case of intensive filling of pressurizer is actual problem.

Usually the water hammer (WH) means pulse high-amplitude hydrodynamic impact on equipment/elements of pipeline systems. WHs can significantly affect reliability, operability, a vibration state and wear of the equipment of nuclear power plants (NPP).

A lot of research is devoted to definition of the reasons, conditions and consequences of WH in the single-phase flow modes (for example, [1 – 10], etc.). The accelerated closing of armature, inertia of the head-flow characteristic of pumps, resonant effects, etc. is considered as basic reasons of WHs. Generally, WH in single-phase flows is a result of different kinds of high-amplitude oscillatory or aperiodic hydrodynamic instability (for example, [9, 10]).

Inhomogeneity of a flow structure, intensive processes of an interphasic heat and mass exchange and other factors (for example, [11 – 15], etc.) that are a result of different kinds of oscillatory and aperiodic thermohydrodynamic instability of two-phase flows are basic reasons of WHs in two-phase flows.

The problem of WHs in two-phase transonic flows at the speed close to sonic speed (or the speed of propagation of acoustic disturbances) is the least studied. Feature of this phenomenon is that under certain conditions sonic speed in the steam-liquid medium can be much less than in a steam (for example, [12, 15]). When flow rate exceeds sonic speed there is a sharp stagnation of flow and kinetic energy of a flow transfers into WH pulse energy – "condensation shock". The transonic flow modes can be in the minimum throat areas of the equipment and elements of pipeline systems at NPP. Therefore modelling of WH conditions in transonic flows is an actual problem.

The pressurizer is the safety related system at nuclear power plants with WWERs, and it is designed [1]:

-to generate a pressure in a reactor loop when reactor plant is started up,

-to hold pressure within the prescribed limits in the normal operational modes and when the reactor is shutdown,

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The provided depressurization of pressurizer in case of malfunction and accidents defines possibility of the water hammers (WH) on the pressurizer's body and elements. WHs are followed by high-amplitude hydrodynamic loads on elements of pressurizer system that can exceed limiting values and lead to inadmissible damage of pressurizer's elements (especially in junction of branch pipes and unions) under certain conditions. Thus, in case of WHs at intensive filling of pressurizer, the experimental pressurizer analogue stands have registered pressure pulses with amplitudes by 2 times exceeded normal pressure in the equipment [2 – 4]. In full-scale pressurizers of WWER we can expect still greater relative amplitudes of WH pressure pulses as in case of emergency opening of the pressurizer's POSV, pressurizer is filling at differential pressure to environment about 15.0 MPa.

For determination of the maximum amplitudes of WH pressure ΔP_{gm} the known Joukowski formula [3] is traditionally used for different systems / the equipment (see, for example, reviews [5, 6]):

$$\Delta P_{gm} = \rho \Delta v c, \tag{2}$$

Where ρ is flow density; Δv is change of flow rate before and after WH; c is sonic speed in equipment/ pipeline metal.

The authors of [4] have also applied formula (2) to assess ΔP_{gm} for filling of the WWER-440 pressurizer experimental model. However, such approach is not well-substantiated for modelling of WH conditions and consequences in full-scale WWER pressurizers as the formula (4.1) does not consider specific features / effects in case of WH generation (for example, an operating filling level of pressurizer, considerable differential pressure in pressurizer and containment in case of WH caused by opening of the pressurizer's POSV, hydraulic characteristics of elements of pressurizer system, the coolant charging in pressurizer, etc.).

Therefore, development of a method for determination of WH conditions and consequences in case of intensive filling of pressurizer is actual problem.

Water hammers (WH) - a phenomenon that is accompanied by impulse hydrodynamic impact on equipment and elements of pipeline systems. The amplitude of the pressure pulse at WH can significantly exceed the permissible static loads on equipment structures and elements of pipeline systems (for example, [1-4] and others). The WH is especially dangerous for nuclear power plant (NPP) piping systems (NPP): failure of equipment of systems important for safety (SIS), as a result of WH, can be both an initial emergency event (IEE) and cause of beyond design basis accidents with multiple failures. Multiple failures of the SIS became one of the causes of the catastrophic environmental consequences of the big accident at the Fukushima-Daiichi nuclear power plant in 2011.

The occurrence of WH in the reactor circuit WWER / PWR can lead to the dominant for the safety of nuclear power plants groups of accidents with violation of the cooling conditions of the reactor core and failures to operate the main circulating pump (MCPU); as well as lead to the destruction of internal reactor equipment (reactor, steam generator, MCPU) and increased vibrational state of the pipelines of the reactor circuit.

Numerous studies have been devoted to the computational and experimental modeling of WH in energy equipment systems (for example, [1-7] and others). An analytical review of the latest research in these areas is given, for example, in [3]. Most of the studies have mainly studied the effects of WH. To estimate the maximum pressure amplitude (ΔP_{wm}), the well-known formula N.E. Zhukovsky [1] is traditionally used:

$$\Delta P_{\mathrm{wm}} = \rho \cdot \Delta V \cdot c_{\mathrm{m}} \tag{3}$$

where is ρ - the flux density; ΔV - difference in flow rates before and after WH; c_{m} - speed of sound in the metal of structures.

In particular, in [8], formula (3) was applied to estimate the maximum amplitude of WH when the MCPU of a reactor circuit with WWER "jammed". However, the cause of the occurrence of WH in the reactor circuit may be other effects: the inertia of the pressure-supply characteristic MCPU [3], resonance pressure fluctuations [7], and others. These provisions determine the relevance of the work presented, which is devoted to modeling the conditions and consequences of WH due to aperiodic and vibrational hydrodynamic instability in the reactor circuit.

The urgency of experimental modeling of emergency processes at nuclear power plants (NPP) is determined by the fundamental absence of the possibility of "artificial" creation of accidents on real equipment of systems important for the safety of nuclear power plants; as well as the need for experimental verification of calculated emergency codes (CEC).

After the accident at the NPMI TMI-2 (USA), numerous integrated experimental installations were designed to study accidents involving decompression of the reactor circuit with WWER / PWR: ICB / PCB, PM-5, KMC (Russia); PMR-NVH, PMK-2 (Hungary); SEMISCALE (USA); LOBI-MOD2 (CEC-ISPRA); UMCP (United States); SPES (Italy); PKL (Germany); BETHSY (France); LOFT (USA); ROSA-IV, PWR (Japan); REWET, PACTEL (Finland) and others. The basic design and technical parameters of these experimental installations can be found, for example, in [1].

However, the adequacy of experimental and real conditions and structural and technical parameters have not been adequately studied.

One of the effective approaches to solving these issues is the analysis of the identity of the criteria for the similarity of processes in experimental and real conditions. For example, in the monograph of the authors [2] it was shown that the similarity criteria adopted in the design of the ICB / PCB-WWER experimental installations are not enough to justify the adequacy of the experimental and real WWER / PWR conditions; as well as for a reasonable extrapolation of the results of experimental verification to real conditions of CEC using physically unjustified closing relations of mathematical models [2].

When simulating accidents and safety analysis of nuclear power plants with WWER / PWR, the following initial accident events (IAE) are usually considered: depressurization / leakage of the reactor circuit; leaks / ruptures of steam lines and pipelines of boiler feedwater; complete de-energization of the power unit and others. One of the lessons of the big accident at the Fukushima-Daiichi nuclear power plant is the need for modeling and analysis of beyond-design accidents with multiple failures of systems important for safety (SIS).

Many years of experience in the operation of power equipment (pumps, fittings, heat exchangers, etc.) have shown that the most critical for reliability are water hammers (WH) [8] for equipment / elements of piping systems accompanied by pulsed high-amplitude pressure increase and appropriate braking of the oncoming stream velocity. Under WH conditions, the kinetic energy, when the flow is decelerated, is partially or completely converted into the energy of the WH pulse.

The characteristic statistics of the State Institution for various equipment of the SIS of NPP with the PWR reactors is given, for example, in [3]. According to the statistics given in [3], more than 90% of registered WH are accounted for by safety systems with pumping equipment.

Critical for the reliability / availability of equipment, the WH can be both design-basis failures in the process of accident development, and directly IAS.

Numerous studies have been devoted to the problems of computational modeling of WH (the reviews of these studies are given, for example, in [3, 4, 5]). In most works, the well-known formula of N.E. Zhukovsky [6]:

$$\Delta P_{WH} = \rho \cdot \Delta U \cdot C_m \qquad (4)$$

Where ρ - the density of the flow medium; ΔU - the difference between the average flow velocity before and after the WH; C_m - speed of sound in the metal equipment and pipelines.

However, formula (4) does not determine the conditions for the formation of WH in various pipeline systems and does not take into account the determining effects of the kinetic energy transition when the flow brakes the pulse energy. Thus, in the works of A.V. Korolev in the experimental model of the WWER-440 pressure compensator [7] established significantly underestimated values ΔP_{WH} according to formula (4) with respect to the experimental data.

In the monograph of the authors [5] original methods for determining the conditions and parameters of hydraulic attacks on power equipment of SIS of NPP due to vibrational and aperiodic hydrodynamic instability are presented.

However, the assumptions/ simplifications adopted in the methods [5] determine the need for experimental verification. The key issue of verification is the analysis of the criteria for the similarity of design, technical and operational parameters of experimental and real installations, which determines the relevance of the work.

According to IAEA recommendations and Fukushima-Daiichi lessons programs for qualification of the safety related systems (SRS) at nuclear power plants are developed and implemented (for example, [1, 2]) in world nuclear power engineering (including Ukraine). Qualification of SRS is meant as substantiation of operability and reliability of safety functions of the SRS equipment/ elements under normal operating conditions, failing of normal operating conditions and accidents. Qualification is carried out by experimental, calculation and experiment-calculated methods [1].

The pilot-operated safety valves (POSV) of the pressurizer of a reactor loop are one of such SRSs. POSV is a three-channel protective safety system and is designed for:

Emergency pressure decrease in a reactor loop when it exceeds the maximum permissible value P_m,

Management of accident processes in the "Feed/Bleed" mode of a reactor loop.

Each POSV channel consists of the main safety valve (MSV) and two pulse valves (PV). PV is response on a setting "Exceeding of pressure into a reactor loop". They open the MSV sliding piston to release pressure and the coolant. The coolant is discharged through MSV to the bubbler tank (BT). A setting "Exceeding of pressure into a reactor loop" actuates at P_m = 18.5 – 19.2 MPa for one of POSV channels, at P_m = 19.0 – 19.6 MPa for two other channels. The settings of MSV closing are $P \leq 17.0$ MPa and $P \leq 17.4$ MPa, respectively.

For experimental qualification of PV and MSV of POSV, operational tests are provided by pressure increase in the pressurizer before operation of PV and by the coolant release to BT under "hot shutdown" of the reactor (for example, [3]). However, results of these tests are insufficient for qualification of POSV due to the following basic reasons:

"Hot shutdown" tests are rare (usually only after system overhaul and/or after emergency operation),

Test conditions on temperature and the coolant level into the pressurizer usually do not meet nominal operating modes of the reactor,

Diagnostics and control of conditions for the water hammers (WH) into the pressurizer, PV and MSV of POSV are insufficient.

WHs are followed by the pulse high-amplitude pressure release ΔP_m and can very affect on operability and reliability of safety functions of the pressurizer POSV. For the WH specified parameters, there are possible:

Damage and destruction of elements of the pressurizer,

Failures to close PV and MSV after pressure release.

Therefore, well-known tests of Prof. Korolev at the WWER pressurizer experimental model [4] have registered the maximum amplitudes of a WH pressure pulse by 2 – 7 times more than static pressure in system. Damage/destruction of elements of the pressurizer (including connecting pipelines of POSV) is possible under such pulse dynamic loads.

Effects of failures to close the safety valves (SV) can be shown on the example of well-known incident at the Rivne-4 in 2009 [1]. During testing of the POSV SV in the "hot shutdown" mode of the reactor there was a jamming to close a PV sliding piston. It led to great release of the radioactive coolant in a reactor building of containment.

WH origins in the POSV SV may be:

The sharp (pulse) stagnation of the coolant flow due to the SV accelerated closing (WH1),

The condensation shocks due to the transonic modes of two-phase flows in the SV flow part (WH2).

The authors' work [5] shows that the WH1 origins may be due to aperiodic hydrodynamic instability when kinetic energy of flow stagnation turns into pressure pulse energy of water hammer. The WH2 origins are because the flow part of armature can be considered conditionally as an element with convergent (entrance) and divergent (output) sites (Fig. 7.1).

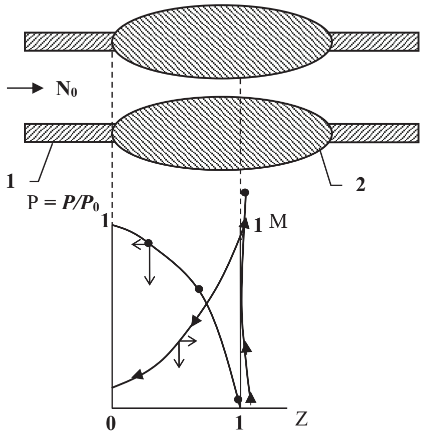


Fig. 1. Water hammer conditions in the transonic modes of a two-phase steam-liquid flow [5]: 1 – pipeline; 2 – local hydrodynamic resistance.

The "hot" coolant coming to a convergent site is accelerated (motion speed v increases), and pressure decreases longwise a divergent site. Pressure decrease P leads to boiling up of the coolant, generating of a two-phase flow, following increase in flow rate and pressure decrease. When the two-phase flow rate reaches sonic speed near to the minimum throat area (the Mach criterion M = 1 – the transonic mode), there are abrupt stagnation of a flow in a divergent site and the pulse local increase of WH2 pressure (condensation shock) (see Fig. 1).

Thus, qualification of the POSV SV for different water hammer conditions during tests and/or accidents at nuclear power plants is an actual problem.

TOPIC 1. MODELLING METHOD OF CONDITIONS FOR RELIABILITY-CRITICAL HYDRAULIC IMPACTS ON PUMPS OF THERMAL AND NUCLEAR POWER PLANTS

1.1 BASIC PROVISIONS OF MODELLING OF CONDITIONS OF CRITICAL HYDRAULIC IMPACTS ON PUMPING EQUIPMENT

1. The typical (for thermal and nuclear power plants) pipeline system of the heat engineering equipment is considered (Fig. 1.1). Pipeline is conditionally separated into the supply and pressure lines with length L_1 and L_2, respectively.

2. It is assumed that flow HHI is a necessary condition for CHI on working elements of pumps. HHI consists in a divergence of hydrodynamic parameters from the steady-state (stable) values.

The inoperative pump failure because of critical hydrodynamic loads on working elements exceeding maximum $N_{кр}$ is sufficient condition for CHI. The maximum (critical) values of a mass flow rate $G_{кр}$ and/or a mean flow rate $v_{кр}$:

$$G \geq G_{кр} = F\sqrt{2\rho N_{кр}};\ v \geq v_{кр} = \sqrt{\frac{2N_{кр}}{\rho}}, \tag{1.1}$$

where G, v – the current values of a mass flow rate and mean flow rate, respectively; F – flow area of the pipeline; ρ – flow density; $N_{кр}$ – maximum loads on the working pump elements determined *by its constructive and technical characteristics.*

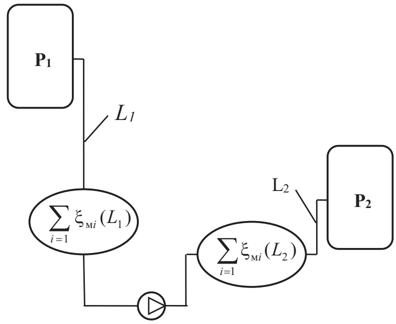


Fig. 1.1. Typical pipeline system of the heat engineering equipment.

3. The key parameters of conditions for HHI and CHI.

The flow (network) characteristic of dependence of hydrodynamic pump head $\Delta P_{\text{н}}$ on a mass flow rate G or mean flow rate in a pipeline system v:

$$\Delta \text{Đ}_{\text{н}} = f(G);\ \Delta \text{Đ}_{\text{н}} = f(v). \tag{1.2}$$

Sensitivity of the flow (network) characteristic to flow rate changes:

$$f' = \frac{\mathrm{d}\Delta P_{\text{н}}}{\mathrm{d}G} \text{ or } f' = \frac{\mathrm{d}\Delta P_{\text{н}}}{\mathrm{d}v}. \tag{1.3}$$

For force pumps design sensitivity of the flow (network) characteristic:

$$f'(G, v) \le 0. \tag{1.4}$$

Decrease in static pressure P below the saturation pressure at a certain flow temperature P_s defines conditions for a flow boiling up in the supply line (at the pump inlet):

$$\text{Đ}(G, v) \le P_s. \tag{1.5}$$

If the condition (1.5) is met at the pump inlet there is an intensive steam generation (boiling up) and cavitation with generation of local steam volume and the "missile" flow mode (with variability of liquid and steam flow phases) [5]. This mode can lead to powerful pulse hydrodynamic impacts on the pump working elements as a result of sharp local drop of hydrodynamic resistance and increase in speed of liquid "missile" in proportion to the relation $\rho/\rho_{\text{п}}$ ($\rho_{\text{п}}$ – density of a steam phase).

It is conservatively (with reliability "margin") assumed that the mode on condition (1.5) corresponds to conditions for CHI on the pump working elements.

4. Oscillatory hydrodynamic instability in the considered pipeline system is defined by inertial delay of corresponding changes of the flow (network) characteristic of the pump and hydrodynamic resistance of the line: according to a formula (3) the increase in an flow G/flow rate v at the moment t leads to reduction of hydrodynamic pump head $\Delta P_{\text{н}}$ and to increase in the general hydrodynamic resistance of the line ΔP_0 after time interval Δt; reduction of a hydrodynamic pressure $\Delta P_{\text{н}}$ and increase in ΔP_0 leads to further reduction of G/v after time interval Δt, etc. Thus, there is an oscillatory process:

$$\begin{aligned}&\uparrow G, v(t) \Rightarrow \uparrow \Delta P_0;\\&\downarrow \Delta P_{\text{н}}(t+\Delta t) \Rightarrow \downarrow G, v(t+2\Delta t) \Rightarrow \downarrow \Delta P_0;\\&\uparrow \Delta P_{\text{н}}(t+2\Delta t) \Rightarrow \uparrow G, v...\end{aligned} \tag{1.6}$$

When current sensitivity of the flow characteristic of the pump f' is insufficient the period and total amplitude of a flow rate increase and can reach critical values (1.1). The system actually transfers to a state of aperiodic instability [4, 6]. Any random (fluctuation) disturbance of hydrodynamic parameters leads to pulse ("intermittent") transition of a pipeline system to a state with CHI on the pump when "choking" of a pressure section of the pipeline (a condition of CHI (1.1)) or sharp local decrease in hydrodynamic resistance and increase in flow rate (a condition of CHI (1.5)).

Assuming incompressibility of liquid and an isothermality of processes, a flow equation in the considered pipeline system and the current change of hydrodynamic pump head:

$$\rho L \frac{\mathrm{d}v}{\mathrm{d}t} = \Delta P_{\text{н}}(v) + P_1 - P_2 - \Delta P_1(v) - \Delta P_2(v), \tag{1.7}$$

$$\Delta P_{\text{н}} = \Delta P_{\text{нm}} + \int_0^t f'(v) \frac{\mathrm{d}v}{\mathrm{d}\tau}\mathrm{d}\tau \tag{1.8}$$

at initial conditions

$$v(t=0)=0 \tag{1.9}$$

$$\Delta P_{\text{н}}(t=0) = \Delta P_{\text{нm}}, \tag{1.10}$$

where ρ – flow density; L – length of the pipeline; $\Delta P_{нт}$ – the maximum hydrodynamic pump head determined by its technical characteristics; t – current time; v – mean flow rate; f' – the current sensitivity of the flow characteristic of the pump; P_1, P_2 – static pressure in object of a source and consumption, respectively (Fig. 1.1).

Hydrodynamic resistance in the supply L_1 and pressure L_2 line:

$$\Delta Đ_1 = \left[\xi_{\text{тр}}\frac{L_1}{D} + \sum_{i=1} \xi_{\text{м}i}(L_1)\right]\rho v^2 - \rho g \sum_{j=1} h_j \operatorname{sign}\left[v_j(L_1)\right], \tag{1.11}$$

$$\Delta Đ_2 = \left[\xi_{\text{тр}}\frac{L_2}{D} + \sum_{i=1} \xi_{\text{м}i}(L_2)\right]\rho v^2 - \rho g \sum_{j=1} h_j \operatorname{sign}\left[v_j(L_2)\right], \tag{1.12}$$

where $\xi_{тр}$, $\xi_{мi}$ – coefficient of transport and local hydrodynamic loss in pipelines, respectively; D – diameter of pipeline flow area; g – acceleration due to gravity; h_j – height of vertical sections of a pipeline system;

$$\operatorname{sign}(v) = \begin{cases} 1, & \text{for downflow;} \\ -1, & \text{for upflow.} \end{cases}$$

1.2 THE ANALYSIS OF CONDITIONS FOR CRITICAL HYDRAULIC IMPACTS AT HEATHYDRODYNAMIC INSTABILITY

To bring the equations to a criteria form we will enter dimensionless variables of hydrodynamic parameters and their corresponding scales (m):

$$\boldsymbol{v} = \frac{v}{v_{\text{кр}}};\ \boldsymbol{t} = \frac{t}{t_{\text{м}}};\ \boldsymbol{\Delta Đ}_i = \frac{\Delta Đ_{\text{н}}}{\Delta Đ_{\text{нm}}};\ \boldsymbol{Đ} = \frac{Đ}{\Delta Đ_{\text{нm}}}.$$

Then the equations (1.7) and (1.8) in a criteria form:

$$\rho \frac{L v_{\text{кр}}}{\Delta Đ_{\text{нm}} t_{\text{м}}} \frac{\mathrm{d}\boldsymbol{v}}{\mathrm{d}\boldsymbol{t}} = \boldsymbol{\Delta P}_{\text{i}} + \boldsymbol{P}_1 - \boldsymbol{P}_2 - \boldsymbol{\Delta P}_1 - \boldsymbol{\Delta P}_2, \tag{1.13}$$

$$\boldsymbol{\Delta P}_{\mathrm{i}} = 1 + \int_0^{t} \frac{\mathrm{d}\boldsymbol{\Delta P}_{\mathrm{i}}}{\mathrm{d}\boldsymbol{v}} \frac{\mathrm{d}\boldsymbol{v}}{\mathrm{d}\boldsymbol{\tau}} \mathrm{d}\boldsymbol{\tau} \,. \tag{1.14}$$

The time scale of process t_m follows from (1.13):

$$\frac{\rho L v_{\text{кр}}}{\Delta Đ_{\text{нм}} t_{\text{м}}} \equiv 1 \Rightarrow t_{\text{м}} = \frac{\rho L v_{\text{кр}}}{\Delta Đ_{\text{нм}}} \tag{1.15}$$

Flow rate, critical for CHI, follows from conditions (1.1) and (1.5):

$$v_{\text{кр}} = \min \begin{cases} \sqrt{\dfrac{2N_{\text{кр}}}{\rho}}, \\ \sqrt{\dfrac{2\left[P_1(L_1) - P_s\right]}{\rho}}. \end{cases} \tag{1.16}$$

Thus, flow equation in a criteria form:

$$\frac{\mathrm{d}\boldsymbol{v}}{\mathrm{d}\boldsymbol{t}} = 1 + \int_0^{t} \frac{\mathrm{d}\boldsymbol{\Delta P}_{\mathrm{i}}}{\mathrm{d}\boldsymbol{v}} \frac{\mathrm{d}\boldsymbol{v}}{\mathrm{d}\boldsymbol{\tau}} \mathrm{d}\boldsymbol{\tau} + \boldsymbol{P}_1 - \boldsymbol{P}_2 - \boldsymbol{\Delta P}_1 - \boldsymbol{\Delta P}_2 \tag{1.17}$$

$$\boldsymbol{v}(\boldsymbol{t} = 0) = 0 \,, \tag{1.18}$$

The equation (1.17) is the nonlinear differential equation and does not have generally analytical solutions. In this case solutions can be obtained by numerical methods of integration.

Assuming linear approximation of sensitivity of the flow (network) characteristic of the pump

$$f'(v) = -\kappa \ (\kappa > 0) \,, \tag{1.19}$$

the equation (17) has the following form:

$$\frac{\mathrm{d}\boldsymbol{v}}{\mathrm{d}\boldsymbol{t}} = \grave{A} + \hat{A}\boldsymbol{v} - \tilde{N}\boldsymbol{v}^2 \,, \tag{1.20}$$

where

$$\grave{A} = 1 + \frac{\rho g \sum\limits_{j=1} h_j \operatorname{sign}\left[v_j(L)\right]}{\Delta Đ_{\text{нм}}} + \frac{P_1 - P_2}{\Delta Đ_{\text{нм}}} \,,$$

$$\hat{A} = -\kappa \frac{v_{\text{кр}}}{\Delta Đ_{\text{нм}}} \,,$$

$$\tilde{N} = \left[\xi \frac{L}{D} + \sum_{i=1} \xi_{\text{м}i}(L)\right] \frac{\rho v_{\text{кр}}^2}{\Delta Đ_{\text{нм}}} \,.$$

The analytical solution of the equation (1.20):

$$\frac{\sqrt{\hat{A}^2 + 4\grave{A}\tilde{N}} - \hat{A} + 2\tilde{N}\boldsymbol{v}}{\sqrt{\hat{A}^2 + 4\grave{A}\tilde{N}} + \hat{A} - 2\tilde{N}\boldsymbol{v}} = \exp\left(\mathrm{const} + \sqrt{\hat{A}^2 + 4\grave{A}\tilde{N}}\,\boldsymbol{t}\right), \tag{1.21}$$

$$\boldsymbol{v}(\boldsymbol{t} = 0) = 0 \,.$$

Condition for CHI:

$$\boldsymbol{v} \geq 1 \,. \tag{1.22}$$

The typical example of the solution of (1.21) in a criteria form is shown in Fig.1.2.

This leads to the following main conclusions.

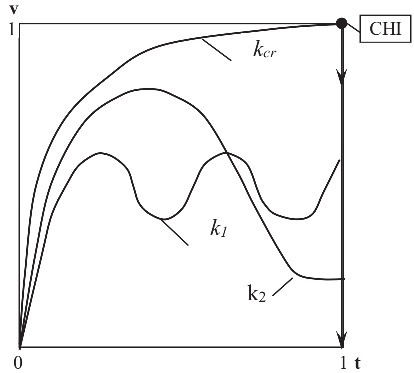


Fig. 1.2. A mean flow rate for HHI with different sensitivity of the flow (network) characteristic of the pump and coefficients of linear approximation $k_1 > k_2 > k_{кр}$.

TOPIC 2. METHOD FOR DETERMINATION OF WATER HAMMER CONDITIONS & CONSEQUENCES IN WWER PRESSURIZER

2.1. BASIC PROVISIONS OF METHOD FOR DETERMINATION OF WATER HAMMER CONDITIONS & CONSEQUENCES IN CASE OF EMERGENCY FILLING OF THE PRESSURIZER

The design model of WH conditions in WWER pressurizer is given in Fig. 2.1.

Basic provisions and assumptions of a design model are following.

1. Filling of pressurizer can result from overcharging of a reactor loop by the relevant systems G_b and/or emergency opening of the pressurizer's POSV and/or realizing of the "release – feed" mode to manage accident processes.

2. Non-isothermality of processes is conservatively negligible.

3. When pressurizer is filling, the WH hydrodynamic parameters are determined from a condition of transfer of kinetic energy of a flow when stagnation to WH pulse energy transition of kinetic energy when stagnation to WH pressure pulse energy.

4. It is assumed the pressurizer is a regular cylinder with a length H and throat area $\Pi_к$ (see Fig. 2.1).

5. Constructional and technical and hydraulic characteristics of full-scale pressurizers of WWER-1000 were accepted according to [1, 7].

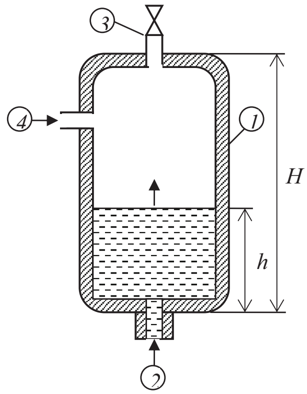


Fig. 2.1. Design model of WH conditions in case of filling of WWER pressurizer:
1 – Pressurizer; 2 – Reactor loop;
3 – Pressurizer's POSV;
4 – Coolant-charging system of pressurizer.

All generating process of WH conditions is separated into two non-uniformly time scaled stages:

Stage of filling of pressurizer with the coolant $t_{\text{к}}$,

Stage of interaction between a WH pulse and a pressurizer's inner surface t_{c}.

Taking into account the accepted assumptions, at the 1st stage ($0 \le t \le t_{\text{к}}$) the balance equation of mass of pressurizer's steam volume and the motion equation of coolant level in pressurizer:

$$\Pi_{\text{к}} \frac{\mathrm{d}}{\mathrm{d}t}\left[\rho_{\text{v}}(H-h)\right] = -G_{\text{к}} - G_{\text{f}} - \Delta G_{\text{b}}, \tag{2.2}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}(hG_{\text{к}})^2 = (P_{\text{т}} - P_0)\Pi_{\text{к}} - \frac{\xi_{\text{к}}}{2}\frac{G_{\text{к}}^2}{\rho\Pi_{\text{к}}} - \rho\Pi_{\text{к}} g h, \tag{2.3}$$

$$G_{\text{f}} = \Pi_{\text{f}}\sqrt{\frac{2(P_{\text{т}} - P_0)\rho}{\xi_{\text{f}}}}, \tag{2.4}$$

$$\Delta G_{\text{b}} = G_{\text{b}}\frac{i_{\text{v}} - i_{\text{b}}}{i_{\text{b}}} \tag{2.5}$$

Under initial conditions

$$G_{\text{к}}(t=0) = 0; \qquad h(t=0) = h_0. \tag{2.6}$$

Necessary condition for WH when pressurizer is filling:

$$h(t = t_{\text{к}}) = Í\ . \tag{2.7}$$

Taking into account the accepted assumptions, at the 2nd stage ($0 \le t \le t_{\text{c}}$) the equation of conservation laws under a necessary condition (7):

$$H\Pi_{\text{к}}\frac{\mathrm{d}\rho}{\mathrm{dP}}\frac{\mathrm{d}\Delta Đ_{\text{g}}}{\mathrm{d}t} = G_{\text{к}} - G_{\text{f}}, \tag{2.8}$$

$$H\frac{\mathrm{d}G_{\text{к}}}{\mathrm{d}t} = -\Delta Đ_{\text{g}}(\Pi_{\text{к}} - \Pi_{\text{f}}), \tag{2.9}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{G_{\text{к}}^2}{2\rho\Pi_{\text{к}}} + \rho\Delta i_{\text{g}}\right) = 0, \tag{2.10}$$

Where t is process time; ρ_v, ρ is the steam and coolant density, respectively; h is coolant level height *in pressurizer; G_k is a coolant mass flow in pressurizer; G_f is a mass flow through the pressurizer's POSV; P_m, P_0 is pressure in the coolant and containment, respectively; $\xi_к$ is total coefficient of hydraulic resistance in pressurizer; ξ_f is coefficient of hydraulic resistance of the pressurizer's POSV; Π_f is throat area of the pressurizer's POSV; i is specific enthalpy (per mass unit); G_a is a rated flow of the coolant through the reactor core; ΔP_g, Δi_g is a pressure and enthalpy pulse in WH zone, respectively.*

The equations (2.2) – (2.10) are in a criteria (dimensionless) format:

$$\frac{d}{d\mathbf{t_1}}\left[\rho_v(1-\mathbf{h})\right] = -\mathbf{K_1} - \mathbf{K_2} - \mathbf{G_к}, \quad (2.11)$$

$$\frac{d}{d\mathbf{t_1}}(\mathbf{hG_к}) = \mathbf{K_3} - \mathbf{K_4}\mathbf{G_к^2} - \mathbf{K_5}\mathbf{h}, \quad (2.12)$$

$$\mathbf{K_7}\frac{d\boldsymbol{\rho}}{d\mathbf{P}}\frac{d\mathbf{\Delta P_g}}{d\mathbf{t_2}} = \mathbf{G_к}(\mathbf{h}=1) - \mathbf{K_1}, \quad (2.13)$$

$$\mathbf{K_8}\frac{d\mathbf{G_к}(\mathbf{h}=1)}{d\mathbf{t_2}} = -\mathbf{K_3}(1-\mathbf{\Pi_f}), \quad (2.14)$$

$$\frac{d}{d\mathbf{t_2}}\left[\frac{\mathbf{G_к^2}(\mathbf{h}=1)}{\boldsymbol{\rho}}\right] = -\frac{d}{d\mathbf{t_2}}\left(\boldsymbol{\rho}\frac{d\mathbf{i}}{d\mathbf{P}}\mathbf{\Delta P_g}\right) \quad (2.15)$$

$$\mathbf{G_к}(\mathbf{t_1}=0) = 0; \quad \mathbf{h}(\mathbf{t_1}=0) = \mathbf{K_6}, \quad (2.16)$$

Where:

$$\mathbf{K_1} = \frac{\Pi_к}{G_{à}}\sqrt{\frac{2(P_т - P_0)\rho_v}{\xi_f}}; \quad \mathbf{K_2} = \frac{G_b}{G_{à}}\frac{i_v - i_b}{i_b};$$

$$\mathbf{K_3} = \frac{\Pi_к^2(P_т - P_0)\rho_т}{G_{à}^2}; \quad \mathbf{K_4} = \frac{\xi_к}{2}; \quad \mathbf{K_5} = \frac{\rho_т^2\Pi_к^2 Hg}{G_{à}^2};$$

$$\mathbf{K_6} = \frac{h_0}{H}; \quad \mathbf{K_7} = \frac{\rho ñ\Pi_к}{G_{à}}; \quad \mathbf{K_8} = \frac{G_{à}ñ}{Đ_т\Pi_к};$$

$$\boldsymbol{\rho} = \frac{\rho}{\rho_т}; \quad \mathbf{t_1} = \frac{tG_a}{\rho_т H\Pi_к}; \quad \mathbf{h} = \frac{h}{H}; \quad \mathbf{G_{ê}} = \frac{G_к}{G_{à}}; \quad \mathbf{\Pi_f} = \frac{\Pi_f}{\Pi_к};$$

$$\mathbf{i} = \frac{i\rho^2\Pi_к^2}{G_{à}^2}; \quad \mathbf{P} = \frac{P}{P_т}; \quad \mathbf{t_2} = \frac{tc}{H}; \quad \mathbf{\Delta P_g} = \frac{\Delta P_g}{P_т};$$

c is the speed of disturbance propagation (sonic speed) in metal of a pressurizer's inner surface.

Maximum amplitude of a WH pressure pulse:

$$\mathbf{\Delta P_{gm}} = \int_0^1 \frac{d\mathbf{\Delta P_g}}{d\mathbf{t_2}}(\mathbf{h}=1; \mathbf{K_1}; ...; \mathbf{K_8})d\mathbf{t_2}. \quad (2.17)$$

Generally, combined equations (11) – (17) can be solved by numerical methods.

Unlike traditional Joukowski formula (2.1), found solution (2.17) considers background of generating of WH conditions in pressurizer (**K1**, …, **K6**), and also effects of direct generating and consequences of a WH pressure pulse in case of a spontaneous (sharp) flow stagnation against an inner surface of the pressurizer body (2.7, 2.8) (**K7**, **K8**).

2.2. ANALYSIS OF RESULTS OF COMPUTED MODELLING

The known experimental data of Prof. Korolev [4] on the WWER-440 pressurizer model were used to verify the presented method for determination of the WH conditions and parameters in pressurizer. Fig. 2 presents experimental data on the maximum WH amplitude ΔP_{gm} [4] for different lock diameters of the WWER-440 pressurizer model (pressurizer's POSV simulator), and the relevant calculations for the equations (2.1) and (2.17). The presented results say that calculations for Joukowski formula (2.1) have the understated values of ΔP_{gm} versus experimental data, and solutions for the equations (2.11) – (2.17) have satisfactory conservative estimates.

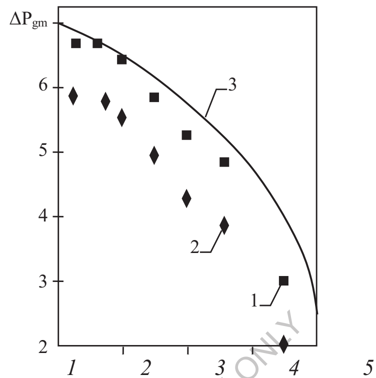
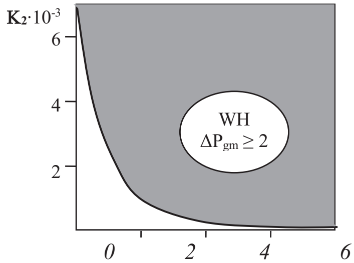


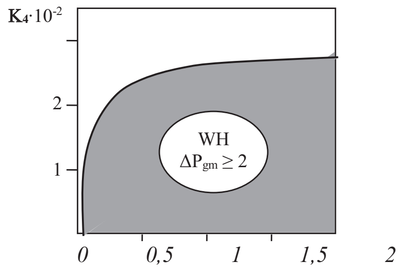
Fig. 2.2. Maximum WH amplitudes when the WWER-440 pressurizer experimental model is filling depending
on a lock diameter d (pressurizer's POSV simulator):
1 – Experiment; 2 – Calculation for a formula (1);
3 – Calculation for formulas (11) – (17).

For computed modelling of WHs in full-scale WWER pressurizer, estimate disarrangement of ΔP_{gm} for (2.1) and (2.17) can be even more considerable as differential pressures in pressurizer and environment are two orders more under natural conditions than in experimental model.

The main results of numerical integration of combined equations (2.11) – (2.17) by Runge-Kutta method for full-scale WWER-1000 pressurizer are presented in Fig. 2.3. As a result of variation calculations with a possible range of criteria $\mathbf{K_1}$, …, $\mathbf{K_8}$ it is found that $\mathbf{K_1}$, $\mathbf{K_2}$, $\mathbf{K_4}$, $\mathbf{K_6}$ are the key criteria of change of WH conditions and amplitudes of WH pressure pulses ΔP_{gm} at the fixed differential pressure in pressurizer and containment.



a) WH area boundaries in a format of $\mathbf{K_2}$ *and* $\mathbf{K_6}$



b) WH area boundaries in a format of **K1** *and* **K4**

Fig. 2.3. WH area boundaries in a format of the key criteria.

Fig. 3, *a* presents computed values of the WH area boundaries for $\Delta P_{gm} \geq 2$ in a format of the key criteria $\mathbf{K_2}$ and $\mathbf{K_6}$ reflecting an initial filling level of pressurizer with the coolant of a reactor loop and charging systems. The results in Fig. 2.3, *a* say that an initial level of the coolant in pressurizer h_0 has the greatest influence on WH conditions (with other things being equal): when $\mathbf{K_6} = h_0/H \geq 0{,}7$, high-amplitude WHs in pressurizer are generated regardless of rate of opening of the pressurizer's POSV. It should be noted that the filling level of pressurizer $h_0/H \approx 0.7$ in a rated operating mode of the reactor [1]. At pressure fluctuations in a reactor loop, this filling level can be even more. Thus, when the pressurizer's POSV is opened emergently there are prerequisites for critical (for reliability) WHs on the pressurizer elements. WH generation can also prevent from necessary closing of POSV at pressure release and loss of the coolant in a reactor loop. Such emergency effect of not closing of the POSV and its consequences took place during "hot shutdown" tests of the pressurizer's POSV at Rivne-4 in 2009.

"Hot shutdown" tests for opening/closing of the pressurizer's POSV are usually carried out at initial filling levels of pressurizer $h_0/H = 0.5 - 0.6$ after overhaul of system. Perhaps, therefore, WHs could not be observed during tests for opening/closing of the pressurizer's POSV as there was no full filling of pressurizer (see Fig. 3, *a*).

Fig. 2.3, *b* presents computed values of the WH area boundaries for $\Delta P_{gm} \geq 2$ in a format of the key criteria $\mathbf{K_1}$ и $\mathbf{K_4}$ reflecting hydraulic characteristics of pressurizer system. The results in Fig. 2.3, *b* say that the total hydraulic resistance of pressurizer $\xi_к$ effects on generating of high-amplitude WHs in pressurizer. When great values of $\xi_к$ are reached "artificially" (see Fig. 2.3, *b*), high-amplitude WHs can be prevented at different rate of opening of the pressurizer's POSV and without great charging of pressurizer.

Thus, the presented results of computed modelling show that the increase in total hydraulic resistance of flow path of pressurizer is the most effective action to prevent high-amplitude WHs in pressurizer.

TOPIC 3. WATER HAMMERS IN TRANSONIC MODES OF STEAM-LIQUID FLOWS IN THE EQUIPMENT OF NUCLEAR POWER PLANTS

3.1. BASIC PROVISIONS AND RESULTS OF MODELLING OF WATER HAMMERS IN TRANSONIC FLOWS

The pipeline system with the local hydrodynamic resistance (armature, throttle devices, pumps/

compressors, etc.) with the variable throat area of a two-phase steam-liquid flow is modelled (Fig. 1). The adiabatic flashing flow in setting of the local hydrodynamic resistance (LHR) is modelled in one-dimensional and quasi-stationary approximation. Frictional pressure drops of a flow into LHR are negligible. The two-phase flow into LHR has homogeneous equilibrium structure. The single-phase medium (steam) is a special case of the considered model.

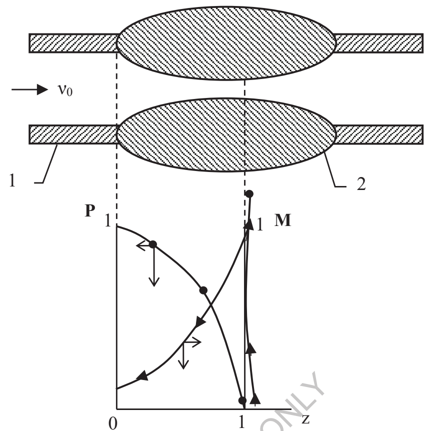


Fig. 3.1. Conditions for water hammers in the transonic modes of a two-phase steam-liquid flow: 1 – Pipeline, 2 – Local hydrodynamic resistance

The following systems of NPP can be analogues of such design model:
Armature of passive part of an emergency core cooling system,
Pipeline system "steam-generator – main steam isolation valve" in emergency operation,
Pilot-operated relief valve of the pressurizer of a nuclear reactor in emergency operation,
Steam dump devices of the turbine plant in emergency operation,
Drainage systems of saturated condensate of the turbine plant, etc.
Under the accepted assumptions, the fundamental equations of conservation laws of a two-phase flow are [12]:

$$\frac{\mathrm{d}\rho\Pi v}{\mathrm{d}z} = 0 , \tag{3.1}$$

$$\frac{\mathrm{d}\rho\Pi v^2}{\mathrm{d}z} = -\Pi\frac{\mathrm{d}P}{\mathrm{d}z} , \tag{3.2}$$

$$\frac{\mathrm{d}}{\mathrm{d}z}\left(\rho i + \frac{\rho v^2}{2}\right) = 0 , \tag{3.3}$$

$$v(z=0) = v_0 , \tag{3.4}$$

$$P(z=0) = P_0 , \tag{3.5}$$

$$i(z=0) = i_0 , \tag{3.6}$$

Where v, P are speed and pressure of a two-phase homogeneous flow, respectively, ρ (R, x) is density of a two-phase flow, i(R, x) is specific enthalpy of a two-phase flow, and x is mass steam content of a two-phase flow.

After transformations, the decisions (3.1) – (3.6) in a criteria format:

$$\frac{\mathrm{d}\mathbf{P}}{\mathrm{d}\mathbf{z}} = \frac{\mathbf{K}_{\mathrm{v}}\boldsymbol{\rho}\mathbf{v}^2}{(1-\mathbf{M}^2)}\mathbf{K}_{\Pi} = \mathrm{grad}(\mathbf{P})\,, \tag{3.7}$$

$$\frac{\mathrm{d}\mathbf{v}}{\mathrm{d}\mathbf{z}} = -\mathrm{grad}(\mathbf{P})\frac{\mathbf{v}}{\boldsymbol{\rho}}\left(\frac{\partial\boldsymbol{\rho}}{\partial\mathbf{P}} + \frac{1}{2r}\frac{\partial\boldsymbol{\rho}}{\partial x}\right) - \mathbf{v}\mathbf{K}_{\Pi}\,, \tag{3.8}$$

$$\frac{\mathrm{d}x}{\mathrm{d}\mathbf{z}} = -\frac{i_0}{2r}\mathrm{grad}(\mathbf{P})\,, \tag{3.9}$$

$$\mathbf{P}(\mathbf{z}=0) = 1; \quad \mathbf{v}(\mathbf{z}=0) = 1; \quad x_0(\mathbf{z}=0) = (i_0 - i_1)/r\,, \tag{3.10}$$

Where $\boldsymbol{P} = P/P_0$, $\boldsymbol{z} = z/L$ (L is length of convergent section of LHR), $\boldsymbol{\rho} = \rho/\rho_0$, $\boldsymbol{v} = v/v_0$, $\boldsymbol{i} = i/i_0$, $\boldsymbol{\Pi} = \Pi_0/\Pi$.

Criteria of the transonic modes of a current:

$$\mathbf{K}_{\mathrm{v}} = \frac{\rho_0 v_0^2}{P_0}; \quad \mathbf{K}_{\Pi} = \frac{1}{\mathbf{\Pi}}\frac{\mathrm{d}\mathbf{\Pi}}{\mathrm{d}\mathbf{z}},$$

$$\mathbf{M} = \frac{v}{a} \text{ – Mach criterion,} \tag{3.11}$$

Where $a = \sqrt{\frac{\partial\rho}{\partial P} + \frac{1}{2r}\frac{\partial\rho}{\partial x}\left(1 - x\frac{\partial i}{\partial P}\right)}$ *is sonic speed in a two-phase homogeneous equilibrium flow, and $r = i_{\mathrm{v}} - i_1$ is latent heat of steam generation.*

The found decisions (3.7) – (3.10) are resulted in followings: pressure and steam content of a flow decrease longwise the channel, and flow rate increases in convergent section of LHR ($\mathbf{K}_{\Pi} < 0$). When flow rate reaches transonic values ($\mathbf{M} \to 1$), modules of flow rate gradients increase sharply; and when flow rate reaches transonic mode ($\mathbf{M} = 1$), there is the subsequent sharp stagnation of a flow ($v \to 0$) and the local pulse growth of pressure and condensation of a steam-liquid flow ("condensation shock" as a result of aperiodic thermo-hydrodynamic instability of the transonic modes of steam-liquid flows). Thus, a necessary condition for WH on the equipment of pipeline systems in the transonic modes of two-phase flows is:

$$\mathbf{M} = 1\,. \tag{3.12}$$

For modelling the WH parameters (a hydrodynamic load ΔP_{g}, rates of change of P and v), it is reputed that when stagnation kinetic energy of a two-phase flow transfers into energy of "condensation shock". Then the balance equations of mass, pressure pulse and energy into "condensation shock":

$$L\frac{\mathrm{d}\rho}{\mathrm{d}t} = \rho v - \rho v_{\mathrm{c}}\,, \tag{3.13}$$

$$L\frac{\mathrm{d}\rho v}{\mathrm{d}t} = -\Delta P_{\mathrm{g}}(t)\,, \tag{3.14}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\rho v^2}{2} + \rho i\right) = 0\,, \tag{3.15}$$

Where t is time, v_c is LHR output flow rate, L is channel length before the section of the transonic mode, and $\Delta P_g(t)$ is amplitude of a hydrodynamic load for WH.

Initial conditions for WH ($\mathbf{M} = 1$):

$$v(t=0) = a\,. \tag{3.16}$$

Assumed conservatively (i.e. reliability margin) that WH processes are isothermal, steam condensation is "instantaneous", and a flow rate after "condensation shock" against a LHR input flow rate is negligible, after transformation (3.13) – (3.16) in a criteria form are:

$$\frac{d\mathbf{P}}{d\mathbf{t}} = \mathbf{v}\,, \tag{3.17}$$

$$\frac{d\mathbf{v}}{d\mathbf{t}} + \mathbf{K}_1 \mathbf{v} \frac{d\mathbf{P}}{d\mathbf{t}} = -\Delta \mathbf{P}_g(\mathbf{t})\,, \tag{3.18}$$

$$\mathbf{v}\frac{d\mathbf{v}}{d\mathbf{t}} + \frac{\mathbf{K}_1}{2}\mathbf{v}^2 \frac{d\mathbf{P}}{d\mathbf{t}} + \mathbf{K}_2 \frac{d\mathbf{P}}{d\mathbf{t}} = 0\,, \tag{3.19}$$

For initial conditions:

$$\mathbf{v}(\mathbf{t}=0) = \frac{a}{v_m} = \mathbf{v}_{g0} \;\; \mathbf{P}(\mathbf{t}=0) = \mathbf{P}_{g0} \;\; \mathbf{i}(\mathbf{t}=0) = \mathbf{i}_{g0}\,, \tag{3.20}$$

Where:

$$\mathbf{P} = \frac{P}{P_m}, \quad \mathbf{v} = \frac{v}{v_m}, \quad \mathbf{t} = \frac{t}{t_m}, \quad \mathbf{i} = \frac{i_l}{i_m},$$

$$\mathbf{K}_1 = \frac{P_m}{\rho_l a_l^2} \;\; \mathbf{K}_2 = \frac{d\mathbf{i}}{d\mathbf{P}}\,, \tag{3.21}$$

ρ_l, a_l are density and sonic speed of a fluid phase, respectively.

Scales of WH parameters follow from transformations of the equations (3.16) – (3.18) to a criteria form:

$$P_m = P_0; \;\; v_m = \frac{P_0}{\rho_l a_l}; \;\; t_m = \frac{L}{a_l}; \;\; i_m = \frac{P_0^2}{\rho_l a_l^2}.$$

The current and maximum relative amplitudes of a hydrodynamic load in the transonic modes of a steam-liquid flow follow from the solution of combined equations (3.17) – (3.21):

$$\Delta \mathbf{P}_g(\mathbf{t}) = \int_0^{\mathbf{t}} \mathbf{v}(\tau)d\tau\,, \tag{3.22}$$

$$\frac{d\mathbf{v}}{d\mathbf{t}} + \frac{\mathbf{K}_1}{2}\mathbf{v} + \mathbf{K}_2 = 0\,, \quad \mathbf{v}(\mathbf{t}=0) = \frac{\rho_l a_l a}{P_0}. \tag{3.23}$$

Sufficient condition for WH in the transonic modes of a two-phase steam-liquid flow is:

$$\Delta \mathbf{P}_g(\mathbf{t}=1) \geq 1\,. \tag{3.24}$$

Against traditional approach for definition of hydrodynamic WH loads due to increase in local hydrodynamic resistances (for example, Joukowski formula [1]), the found decisions consider:

Necessary and sufficient conditions for WH (3.12), (3.24) and the corresponding amplitudes of hydrodynamic loads on the equipment and elements of pipeline systems in case of two-phase steam-liquid flows,

Transfer of kinetic energy of a flow stagnation to WH pulse energy.

The latter provision determines much greater estimated hydrodynamic WH loads against Joukowski formula [1].

Generally, numerical methods of modelling are useful to decide (3.7) – (3.10), (3.22), (3.23) for definition of conditions and consequences of WH in the two-phase steam-liquid modes. Results of numerical integration of combined equations (3.7) – (3.10), (3.22), (3.23) by Runge-Kutta method are given as an example in Fig. 1. The figure shows results for the transonic mode of a two-phase steam-liquid flow at $\mathbf{K}_v = 1$, a convergent section of LHR with $\Pi_{min}/\Pi_0 = 0.5$ and $L = 0.25$ m ($\mathbf{K}_\Pi = 4$). Disclosure of uncertainty of integration of the equations (3.7) – (3.10) was carried out at $\mathbf{M} = 1$ based on L'Hospital rule.

Consequently, we found that flow rate and steam content increase longwise a convergent section, and pressure decreases. When flow rate reaches transonic values ($\mathbf{M} \to 1$), modules of gradients of flow rate and pressure increase significantly; and when transonic mode is reached in the minimum section of LHR Π_{min} (**M** = 1), there is "condensation shock" followed by the local pulse growth of pressure and a flow stagnation. Thus, a necessary condition for WH is generated because of aperiodic instability in transonic two-phase (steam-liquid) flows.

Fig. 3.2 presents the rated range of the initial key modelling criteria $\mathbf{K}_v$ and $\mathbf{K}_\Pi$ met necessary and sufficient conditions for WH (3.12), (3.24) because of aperiodic instability in transonic adiabatic flashing steam-liquid flows.

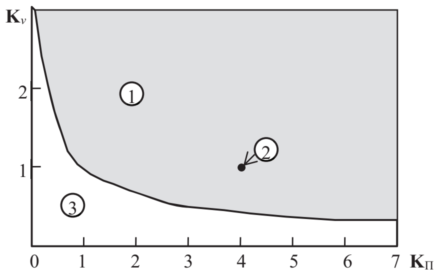


Fig. 3.2. Range of criteria for water hammers because of aperiodic instability in the transonic two-phase steam-liquid flows:
1 – Area of water hammers,
2 – Rated mode at $K_v = 1$, $K_\Pi = 4$, $\Pi_{min}/\Pi_0 = 0{,}5$, $L = 0{,}25$ m (see Fig.3.1),
3 – Area of no water hammers

TOPIC 4. METHOD FOR DETERMINATION OF WATER HAMMER CONDITIONS & CONSEQUENCES IN WWER PRESSURIZER

4.1. BASIC PROVISIONS OF METHOD FOR DETERMINATION OF WATER HAMMER CONDITIONS & CONSEQUENCES IN CASE OF EMERGENCY FILLING OF THE PRESSURIZER

The design model of WH conditions in WWER pressurizer is given in Figure 4.1.

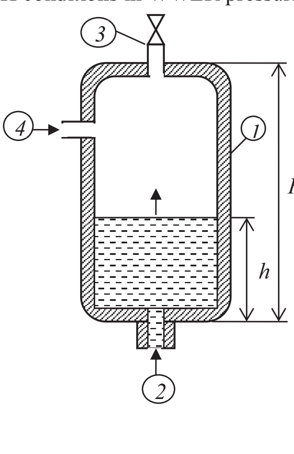


Fig. 4.1. Design model of WH conditions in case of filling of WWER pressurizer: 1 – pressurizer; 2 – reactor loop; 3 – pressurizer's POSV; 4 – coolant-charging system of pressurizer.

Basic provisions and assumptions of a design model are following:

1. Filling of pressurizer can result from overcharging of a reactor loop by the relevant systems G_b and/or emergency opening of the pressurizer's POSV and/or realizing of the "release – feed" mode to manage accident processes.
2. Nonisothermality of processes is conservatively negligible.
3. When pressurizer is filling, the WH hydrodynamic parameters are determined from a condition of transfer of kinetic energy of a flow when stagnation to WH pulse energy transition of kinetic energy when stagnation to WH pressure pulse energy.
4. It is assumed the pressurizer is a regular cylinder with a length H and throat area $\Pi_к$ (as shown in Figure 4.1).
5. Constructional and technical and hydraulic characteristics of full-scale pressurizers of WWER-1000 were accepted according to [1, 7].

All generating process of WH conditions is separated into two non-uniformly time scaled stages:

Stage of filling of pressurizer with the coolant $t_к$,

Stage of interaction between a WH pulse and a pressurizer's inner surface t_c.

Taking into account the accepted assumptions, at the 1st stage ($0 \le t \le t_к$) the balance equation of mass of pressurizer's steam volume and the motion equation of coolant level in pressurizer:

$$\Pi_к \frac{\mathrm{d}}{\mathrm{d}t}\left[\rho_v (H-h)\right] = -G_к - G_f - \Delta G_b, \tag{4.2}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}(hG_к)^2 = (P_т - P_0)\Pi_к - \frac{\xi_к}{2}\frac{G_к^2}{\rho\Pi_к} - \rho\Pi_к g h, \tag{4.3}$$

$$G_f = \Pi_f \sqrt{\frac{2(P_т - P_0)\rho}{\xi_f}}, \tag{4.4}$$

$$\Delta G_b = G_b \frac{i_v - i_b}{i_b} \tag{4.5}$$

Under initial conditions:

$$G_к(t=0) = 0; \qquad h(t=0) = h_0. \tag{4.6}$$

Necessary condition for WH when pressurizer is filling:

$$h(t = t_к) = H. \tag{4.7}$$

Taking into account the accepted assumptions, at the 2nd stage ($0 \le t \le t_c$) the equation of conservation laws under a necessary condition (7):

$$H\Pi_к \frac{\mathrm{d}\rho}{\mathrm{dP}} \frac{\mathrm{d}\Delta P_g}{\mathrm{d}t} = G_к - G_f, \tag{4.8}$$

$$H\frac{\mathrm{d}G_к}{\mathrm{d}t} = -\Delta P_g(\Pi_к - \Pi_f), \tag{4.9}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{G_{\kappa}^{2}}{2\rho\Pi_{\kappa}}+\rho\Delta i_{g}\right)=0, \quad (4.10)$$

Where t is process time; ρ_v, ρ is the steam and coolant density, respectively; h is coolant level height in pressurizer; G_k is a coolant mass flow in pressurizer; G_f is a mass flow through the pressurizer's POSV; P_m, P_0 is pressure in the coolant and containment, respectively; ξ_κ is total coefficient of hydraulic resistance in pressurizer; ξ_f is coefficient of hydraulic resistance of the pressurizer's POSV; Π_f is throat area of the pressurizer's POSV; i is specific enthalpy (per mass unit); G_a is a rated flow of the coolant through the reactor core; ΔP_g, Δi_g is a pressure and enthalpy pulse in WH zone, respectively.

The equations (4.2) – (4.10) are in a criteria (dimensionless) format:

$$\frac{\mathrm{d}}{\mathbf{dt_1}}\left[\rho_{\mathrm{v}}(1-\mathbf{h})\right]=-\mathbf{K_1}-\mathbf{K_2}-\mathbf{G_{\kappa}}, \quad (4.11)$$

$$\frac{\mathrm{d}}{\mathbf{dt_1}}(\mathbf{hG_{\kappa}})=\mathbf{K_3}-\mathbf{K_4}\mathbf{G_{\kappa}^2}-\mathbf{K_5}\mathbf{h}, \quad (4.12)$$

$$\mathbf{K_7}\frac{\mathbf{d\rho}}{\mathbf{dP}}\frac{\mathbf{d\Delta P_g}}{\mathbf{dt_2}}=\mathbf{G_{\kappa}}(\mathbf{h}=1)-\mathbf{K_1}, \quad (4.13)$$

$$\mathbf{K_8}\frac{\mathbf{dG_{\kappa}}(\mathbf{h}=1)}{\mathbf{dt_2}}=-\mathbf{K_3}(1-\mathbf{\Pi_f}), \quad (4.14)$$

$$\frac{\mathrm{d}}{\mathbf{dt_2}}\left[\frac{\mathbf{G_{\kappa}^2}(\mathbf{h}=1)}{\boldsymbol{\rho}}\right]=-\frac{\mathrm{d}}{\mathbf{dt_2}}\left(\boldsymbol{\rho}\frac{\mathbf{di}}{\mathbf{dP}}\mathbf{\Delta P_g}\right), \quad (4.15)$$

$$\mathbf{G_{\kappa}}(\mathbf{t_1}=0)=0; \quad \mathbf{h}(\mathbf{t_1}=0)=\mathbf{K_6}, \quad (4.16)$$

Where:

$$\mathbf{K_1}=\frac{\Pi_{\kappa}}{G_a}\sqrt{\frac{2(P_{\mathrm{T}}-P_0)\rho_{\mathrm{v}}}{\xi_{\mathrm{f}}}}; \quad \mathbf{K_2}=\frac{G_{\mathrm{b}}}{G_a}\frac{i_{\mathrm{v}}-i_{\mathrm{b}}}{i_{\mathrm{b}}};$$

$$\mathbf{K_3}=\frac{\Pi_{\kappa}^2(P_{\mathrm{T}}-P_0)\rho_{\mathrm{T}}}{G_a^2}; \quad \mathbf{K_4}=\frac{\xi_{\kappa}}{2}; \quad \mathbf{K_5}=\frac{\rho_{\mathrm{T}}^2\Pi_{\kappa}^2 Hg}{G_a^2};$$

$$\mathbf{K_6}=\frac{h_0}{H}; \quad \mathbf{K_7}=\frac{\rho c\Pi_{\kappa}}{G_a}; \quad \mathbf{K_8}=\frac{G_a c}{P_{\mathrm{T}}\Pi_{\kappa}};$$

$$\boldsymbol{\rho}=\frac{\rho}{\rho_{\mathrm{T}}}; \quad \mathbf{t_1}=\frac{tG_a}{\rho_{\mathrm{T}}H\Pi_{\kappa}}; \quad \mathbf{h}=\frac{h}{H}; \quad \mathbf{G_{\kappa}}=\frac{G_{\kappa}}{G_a}; \quad \mathbf{\Pi_f}=\frac{\Pi_{\mathrm{f}}}{\Pi_{\kappa}};$$

$$\mathbf{i}=\frac{i\rho^2\Pi_{\kappa}^2}{G_a^2}; \quad \mathbf{P}=\frac{P}{P_{\mathrm{T}}}; \quad \mathbf{t_2}=\frac{tc}{H}; \quad \mathbf{\Delta P_g}=\frac{\Delta P_g}{P_{\mathrm{T}}};$$

c is the speed of disturbance propagation (sonic speed) in metal of a pressurizer's inner surface.

Maximum amplitude of a WH pressure pulse:

$$\mathbf{\Delta P_{gm}}=\int_0^1\frac{\mathbf{d\Delta P_g}}{\mathbf{dt_2}}(\mathbf{h}=1;\mathbf{K_1};...;\mathbf{K_8})\mathbf{dt_2}. \quad (4.17)$$

Generally, combined equations (4.11) – (4.17) can be solved by numerical methods.

Unlike traditional Joukowski formula (4.1), found solution (4.17) considers background of generating of WH conditions in pressurizer (**K**$_1$, …, **K**$_6$), and also effects of direct generating and consequences of a WH pressure pulse in case of a spontaneous (sharp) flow stagnation against an inner surface of the pressurizer body (4.7, 4.8) (**K**$_7$, **K**$_8$).

4.2. ANALYSIS OF RESULTS OF COMPUTED MODELLING

The known experimental data of prof. Korolev [4] on the wwer-440 pressurizer model were used to verify the presented method for determination of the WH conditions and parameters in pressurizer**.**

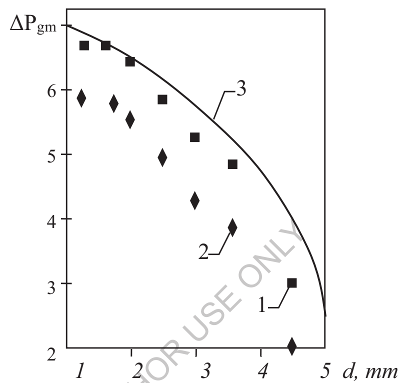
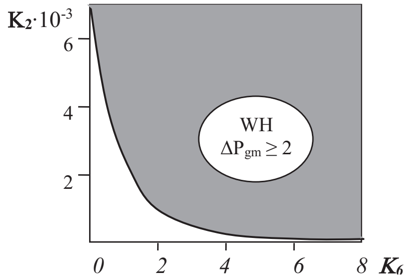


Fig 4.2. Maximum WH amplitudes when the WWER-440 pressurizer experimental model is filling depending on a lock diameter d (pressurizer's POSV simulator):1 – experiment; 2 – calculation for a formula (4.1); 3 – calculation for formulas (11) – (17).

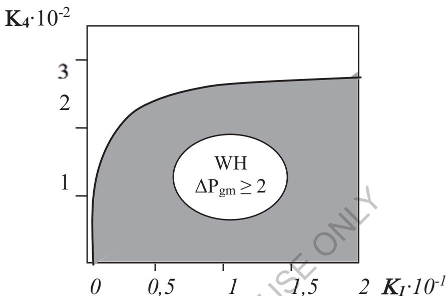
Figure 2 presents experimental data on the maximum WH amplitude ΔP_{gm} [4] for different lock diameters of the WWER-440 pressurizer model (pressurizer's POSV simulator), and the relevant calculations for the equations (4.1) and (4.17). The presented results say that calculations for Joukowski formula (4.1) have the understated values of ΔP_{gm} versus experimental data, and solutions for the equations (4.11) – (4.17) have satisfactory conservative estimates.

For computed modelling of WHs in full-scale WWER pressurizer, estimate disarrangement of ΔP_{gm} for (4.1) and (4.17) can be even more considerable as differential pressures in pressurizer and environment are two orders more under natural conditions than in experimental model.

The main results of numerical integration of combined equations (4.11) – (4.17) by Runge-Kutta method for full-scale WWER-1000 pressurizer are presented in Figure. 4.3.



a) WH area boundaries in a format of $\mathbf{K_2}$ and $\mathbf{K_6}$



b) WH area boundaries in a format of $\mathbf{K_1}$ and $\mathbf{K_4}$

Fig. 4.3. WH area boundaries in a format of the key criteria.

As a result of variation calculations with a possible range of criteria $\mathbf{K_1}$, …, $\mathbf{K_8}$ it is found that $\mathbf{K_1}$, $\mathbf{K_2}$, $\mathbf{K_4}$, $\mathbf{K_6}$ are the key criteria of change of WH conditions and amplitudes of WH pressure pulses ΔP_{gm} at the fixed differential pressure in pressurizer and containment.

Figure 4.3, *a* presents computed values of the WH area boundaries for $\Delta P_{gm} \geq 2$ in a format of the key criteria $\mathbf{K_2}$ and $\mathbf{K_6}$ reflecting an initial filling level of pressurizer with the coolant of a reactor loop and charging systems. The results in Figure 3, *a* say that an initial level of the coolant in pressurizer h_0 has the greatest influence on WH conditions (with other things being equal): when $\mathbf{K_6} = h_0/H \geq 0.7$, high-amplitude WHs in pressurizer are generated regardless of rate of opening of the pressurizer's POSV. It should be noted that the filling level of pressurizer $h_0/H \approx 0.7$ in a rated operating mode of the reactor [1]. At pressure fluctuations in a reactor loop, this filling level can be even more. Thus, when the pressurizer's POSV is opened emergently there are prerequisites for critical (for reliability) WHs on the pressurizer elements. WH generation can also prevent from necessary closing of POSV at pressure release and loss of the coolant in a reactor loop. Such emergency effect of not closing of the POSV and its consequences took place during "hot shutdown" tests of the pressurizer's POSV at Rivne-4 in 2009.

"Hot shutdown" tests for opening/closing of the pressurizer's POSV are usually carried out at initial filling levels of pressurizer $h_0/H = 0.5 - 0.6$ after overhaul of system. Perhaps, therefore, WHs could not be observed during tests for opening/closing of the pressurizer's POSV as there was no full filling of pressurizer (as shown in Figure 4.3, *a*).

Figure 3, *b* presents computed values of the WH area boundaries for $\Delta P_{gm} \geq 2$ in a format of the key criteria **K_1 и K_4** reflecting hydraulic characteristics of pressurizer system. The results in Figure 3, *b* say that the total hydraulic resistance of pressurizer $\xi_к$ effects on generating of high-amplitude WHs in pressurizer. When great values of $\xi_к$ are reached "artificially" (as shown in Figure 4.3, *b*), high-amplitude WHs can be prevented at different rate of opening of the pressurizer's POSV and without great charging of pressurizer.

Thus, the presented results of computed modelling show that the increase in total hydraulic resistance of flow path of pressurizer is the most effective action to prevent high-amplitude WHs in pressurizer.

TOPIC 5. WATER HAMMERS IN THE REACTOR CIRCUIT OF NUCLEAR POWER STATIONS WITH WWER AS A RESULT OF HYDRODYNAMIC INSTABILITY

5.1. BASIC PROVISIONS AND RESULTS OF MODELING OF WATER HAMMER IN THE RECTOR CIRCUIT

The main provisions of the method for determining the water hammer conditions (WH) in the reactor circuit:

1. The calculation scheme of the conditions for the formation of a hydrostatic shock is shown in Fig. 1. The main elements of the reactor circuit are the reactor (RR); steam generator (SG); the main circulation pump (MCPU) and the main circulation pipeline (MCPI).
2. The occurrence of WH can lead to a violation of the cooling conditions of the core RR, failure of the MCPI to work, leaks of the collectors and heat exchanger tubes SG. Therefore, the occurrence of WH is assumed to be either an initial alarm event (IAE), or failures of MCPI elements in the course of beyond design basis accidents.
3. Two initial states are simulated:

- transition mode (MCPU start);
- operating mode (RR operates at rated power).

In transient regimes, the conditions for the emergence of WH are determined by aperiodic hydrodynamic instability (AHI), which is characterized by a pulsed (jump like) change in the hydrodynamic parameters of the coolant. In operating conditions, the occurrence of WH can be caused by oscillatory hydrodynamic instability (OHI), which is accompanied by fluctuations in pressure and velocity of the coolant.

4. The defining effect of WH in the reactor circuit is the inertia of the pressure-supply characteristic (PSC) MCPU [2]:

$$\Delta P_{pu} = f(G) \tag{5.2}$$

where is the pressure head developed by the MCPU; mass flow in the MCC.

The PSC is determined by the design and technical characteristics of the MCPU. By the inertia of PSC, the delay in the PSC response to the change in flow in the MCPI is understood as the maximum delay time of the initial PSC reaction MCPU (Δt) is determined by the total length of the reactor circuit L (including MCPI, height RR and length of the heat exchange tubes SG) velocity of the coolant in the reactor circuit:

$$\Delta t = \rho \Pi_{max} L / G(t) \tag{5.3}$$

where is ρ - the density of the coolant; Π_{max} - the maximum area of the reactor section.

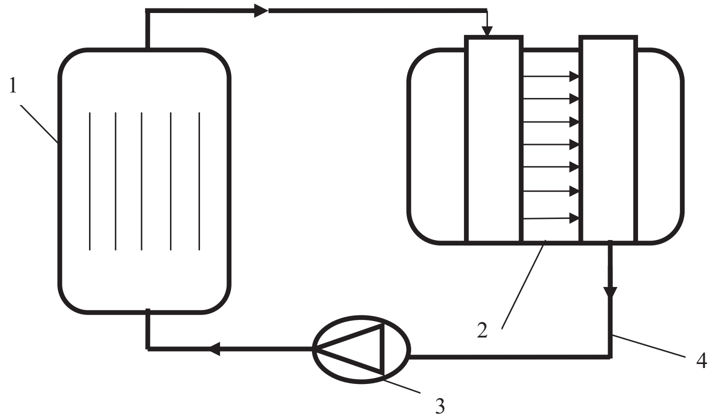


Fig.5.1. The design scheme of the reactor circuit: 1 - reactor (RR); 2 - steam generator (SG); 3 - main circulation pump (MCPU); 4 - main circulation pipeline (MCPI)

In the transient start-up mode MCPU at the initial time (t = 0), the head is the maximum (ΔP_{pum}), and the flow rate in MCPI G (t = 0) = 0

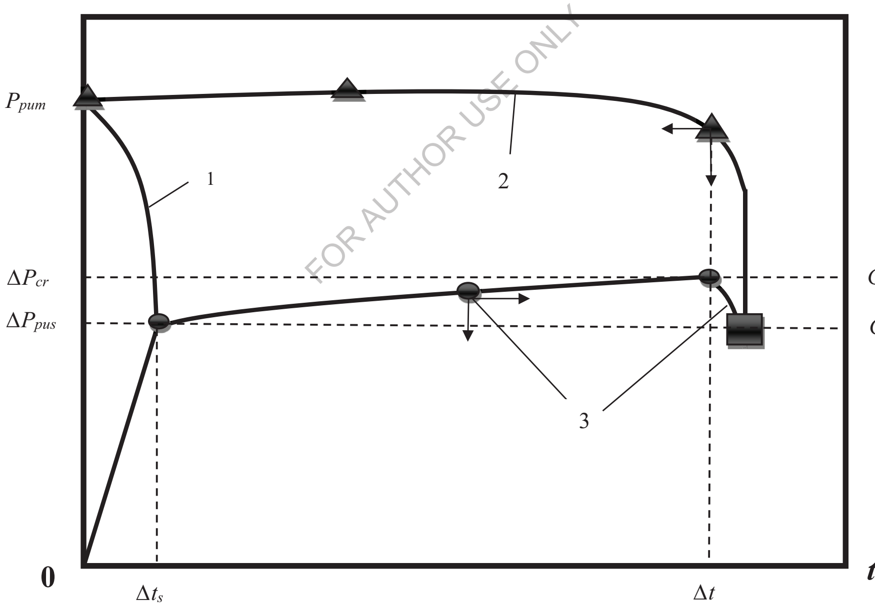


Fig.5.2. The conditions for occurrence of WH in the transient trigger mode of MCPU:
1 - pressure MCPU (ΔP_{pu}) at $\Delta t_s > \Delta t$;
2 - ΔP_{pu} MCPU at $\Delta t \geq \Delta t_s$
3 - coolant flow rate (G).

If the time to reach the steady state (Δt_s) is greater Δt, further increase G leads to a decrease ΔP_{pu} in accordance with the design PSC pump (Fig. 5.2). In the steady state:

$$G(t \geq \Delta t_s) = G_s;\ \Delta P_{pus}(G_s) = \Delta P_g(G_s^2, h); \tag{5.3.a}$$

where is ΔP_q the pressure drop at the hydraulic resistances and in sections of the reactor circuit with a leveling height h.

If $\Delta t \geq \Delta t_s$, then the pump head retains its initial value (ΔP_{pum}) until the time $t = \Delta t$. In this case, the flow continues to increase in the time interval $\Delta t_s < t \leq \Delta t$ (Fig. 5.2). In this time interval, the hydrodynamic system of the reactor circuit is in an unstable state, since $G > G_s$ and $\Delta P_{pu} = \Delta P_{pum} > \Delta P_{pus}$. When $t > \Delta t$ hydrodynamic parameters ($G, \Delta P_{pu}$) go into a stable operating state (3.a), which leads to impulsive braking of the coolant flow and the occurrence of WH as a result of AHI. In this case, the kinetic energy of the deceleration of the coolant flow passes into the energy of the pressure pulse WH.

With a conservative assumption that the processes are isothermal in MCPI and the level drops of pressure are neglected, the equations of motion and conservation of energy in the transient MCPU startup mode can be represented as:

$$\frac{L}{\Pi}\frac{dG}{dt} = \Delta P_{pum} + \int_0^t f'(\tau - \Delta t)\cdot\frac{dG}{d\tau}\cdot d\tau - \frac{\xi_R}{\rho\Pi^2}\cdot G^2 \tag{5.4}$$

$$\frac{d}{dt}\left[\frac{G^2}{2\rho\Pi^2} + \rho i\right] = 0 \tag{5.5}$$

$$G(t=0) = 0;\ i(P, t=0) = i_0(P_0) \tag{5.6}$$

where ξ_R is the total coefficient of hydraulic resistance of the reactor circuit; i - specific (per unit mass) enthalpy of the coolant; P - pressure at the entrance to the MCPU.

Gradient PSC (2): $$f'(\tau - \Delta t) = \begin{cases} 0, \tau < \Delta t \\ d\Delta P_{pu}/dG, \tau \geq \Delta t \end{cases}: \tag{5.7}$$

The criterion of the conditions for the occurrence of WH as a result of AHI:

$$K_{WHA} = \frac{\Delta t}{\Delta t_s} > 1 \tag{5.8}$$

The maximum amplitude of the pressure WH after the AHI follows from equations (5.5):

$$\Delta P_{WMA} = \int_0^{t_c}\frac{dP}{dt}dt = -\frac{\int_0^{t_c} G\cdot\frac{dG}{dt}\cdot dt}{\rho\Pi^2\left(\rho\frac{di}{dP} + \frac{i}{c_2^2}\right)} \tag{5.9}$$

where $t_c \approx L/c_m$ is the time of action of the impulse WH, c_2 is the speed of sound in the liquid coolant.

Unlike the well-known formula (5.1), equation (5.9) takes into account the structural and technical parameters of the reactor circuit, the inertia of the PSC MCPU, and the conditions for the transition of the kinetic energy of the braking of the flow to the energy of impulsive WHs.

In the general case, the solution of the system of equations (5.3) - (5.9) is possible by numerical methods.

A simplified approach to determining the criterion K_{WH} for the occurrence of WH after the AHI is based on the following conservative assumptions (those with a "margin" for WH conditions):

$$\Delta t \approx \frac{\rho\Pi_{max} L}{G_s};\ G\frac{dG}{dt} \approx G_{cr}\frac{G_{cr}-G_s}{\Delta t_s} \tag{5.10}$$

Where $G_{cr}(\Delta t)$ is the maximum (critical) flow at the time $t=\Delta t$ (Fig. 5.2), provided $dG/dt<0$. Under such assumptions, it follows from equations (5.4), (5.5):

$$\Delta t_s = \frac{1}{2\sqrt{K_{pu}\cdot K_\xi}}\cdot \ln\frac{\sqrt{K_{pu}}+G_s\sqrt{K_\xi}}{\sqrt{K_{pu}}-G_s\sqrt{K_\xi}} \tag{5.11}$$

$$K_{WHA} \approx \frac{2\rho\Pi_{max} L\cdot\sqrt{K_{pu}\cdot K_\xi}}{G_s\cdot \ln\left[\left(\sqrt{K_{pu}}+G_s\sqrt{K_\xi}\right)/\left(\sqrt{K_{pu}}-G_s\sqrt{K_\xi}\right)\right]} \tag{5.12}$$

$$\Delta P_{WMA} \approx \frac{G_{cr}(G_{cr}-G_s)}{\rho\Pi^2\left(\rho\frac{di}{dP}+\frac{i}{c_l^2}\right)} \tag{5.13}$$

where $K_{pu} = \Delta P_{pum}\cdot\Pi/L$; $K_\xi = \xi_R/(\rho\Pi\cdot L)$ (5.14)

$$G_{cr} = \rho\Pi\cdot L\sqrt{\frac{K_{pu}}{K_\xi}\cdot\frac{\exp(K_{cr})-1}{\exp(K_{cr})+1}};\ K_{cr} = \frac{2\sqrt{K_{pu}\cdot K_\xi}\,\rho\Pi L}{G_s} \tag{5.15}$$

Under conditions:
$K_{WHA} < 1 - WH\ after\ AHI\ is\ absent$ (5.15a)
$K_{WHA} \geq 1 -\ there\ is\ a\ WH\ as\ a\ result\ of\ AHI$ (5.15b)

5.2. OPERATING MODE OF THE REACTOR

In the steady-state operating mode ($G=G_s$, $P=P_s$), the occurrence of WH can be a consequence of the oscillatory hydrodynamic instability (OHI). The mechanism of the process of the appearance of OHI in the working steady state is presented below. Let a random (fluctuation) increase in flow $\delta G_0 << G_s$ occur at the initial time point ($t=0$) (MCPI) (for example, because of flow turbulence or unstable operation (MCPU)). Fluctational increase in flow rate should lead to a corresponding decrease in pump head $\delta\Delta P_{pu}(\delta G_0)$ and increase in hydrodynamic resistance $\delta\Delta P_g(\delta G_0)$ (Fig. 3). However, the drop in pressure is not instantaneous, but with a certain "lag" Δt (3).

Therefore, for the time interval $0<t\leq\Delta t$, in fact, the PSC gradient $f'(\delta G_0)>0$ is relatively stable (steady) state. In the event $\delta\Delta P_{pu} < \delta\Delta P_g$ that the resulting fluctuation perturbation rapidly "decays" (Fig. 3).

If on the time interval $0<t\leq\Delta t$ at $f'>0$:

$$\delta\Delta P_{pu} > \delta\Delta P_g, \tag{5.16}$$

then the amplitude of the initial disturbance δG_0 will continue to increase to the maximum value δG_A. After $t>\Delta t$ the PSC gradient $f'<0$, the flow perturbation begins to decrease in time. When the pump head reaches this unstable state, PSC determines a further reduction in flow rate ($\delta G < 0$) to maximum amplitudes δG_A (Fig. 5.3); and the pump head is increased to an unstable value $\delta \Delta P_{pu}(\delta G_A)$. Such an unstable state determines a further increase in flow to the maximum amplitude δG_A, taking into account Δt.

According to the energy conservation equation (5.5), the pressure fluctuations are in antiphase with the flow oscillations (Fig. 5.3).

Thus, an oscillatory process of the change in hydrodynamic parameters and oscillatory hydrodynamic instability (OHI) arise.

If the time of the change in pressure $\Delta t_p << \Delta t$, then the fundamental frequency of the oscillations of the hydrodynamic parameters:

$$\omega \approx 1/(4 \cdot \Delta t) \tag{5.17}$$

The amplitude of the pressure oscillations determines the intensity of WH at OHI.

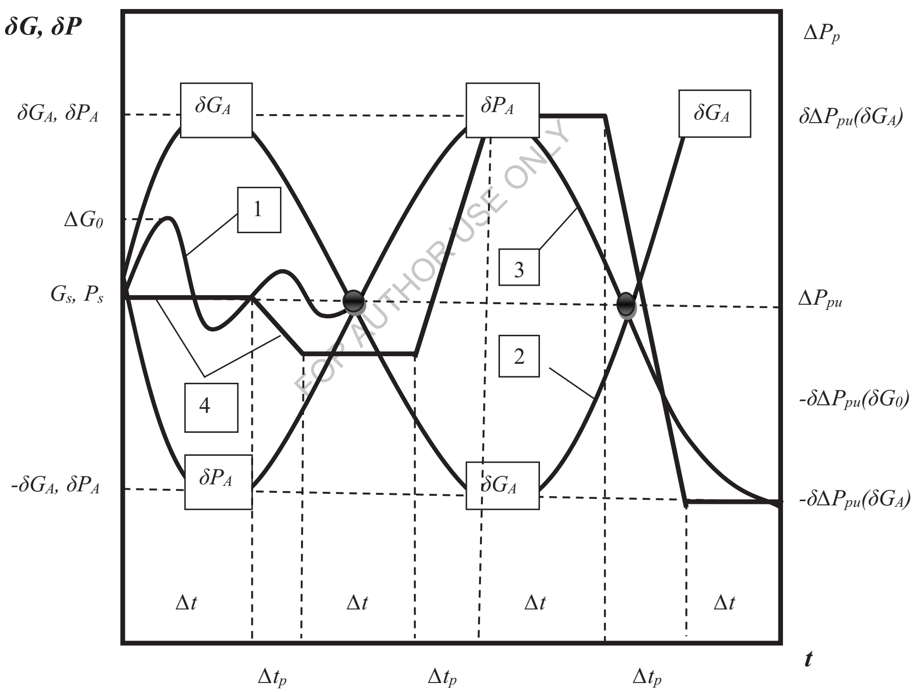


Fig. 5.3. The conditions for the formulation of WH in the steady-state operating mode: 1 - flow oscillations under steady hydrodynamic conditions; 2 - flow fluctuations with OHI; 3 - pressure fluctuations at OHI; 4 - changes in the PSC pump.

$$\frac{L}{\Pi} \cdot \frac{d\delta G}{dt} = \int_0^t f'(\tau - \Delta t) \cdot \frac{V \cdot d\delta G}{d\tau} \cdot d\tau - \frac{\xi_R \cdot V}{\rho \Pi^2} \cdot \left(\delta^2 G + 2G_s \delta G\right) \tag{5.18}$$

$$\frac{d\delta \mathrm{P}}{dt} = -\left(\rho \frac{di}{d\mathrm{P}} + \frac{i}{a_l^2}\right)^{-1} \cdot \frac{G_s}{\rho \Pi^2} \cdot \frac{d\delta G}{dt} \tag{5.19}$$

$$\delta G(t=0) = \delta G_0;\ \delta \mathrm{P}(t=0) = 0 \tag{5.20}$$

In the general case, the solution of the system of equations (5.18) - (5.20) can be obtained by numerical methods.

In the particular case, with the linear approximation of the PSC MCPU ($f' = \mathrm{K_f} = const$) and the assumption $\delta^2 G << 2G_s \delta G$ of the solution (5.18) - (5.20) on the time interval Δt .

$$\delta G = \delta G_0 \exp\left(\mathrm{K_{WHC}} \cdot \Delta t\right) \tag{5.21}$$

$$\Delta \mathrm{P_{WHC}} = \delta \mathrm{P_A} = \frac{G_s \cdot \exp\left(\mathrm{K_{WHC}} \cdot \Delta t\right)}{\rho \cdot \Pi^2 \cdot \left(\rho di / d\mathrm{P} + i / c_1^2\right)} \cdot \delta G_0 \tag{5.22}$$

where the criterion of WH conditions due to OHI:

$$\mathrm{K_{WHC}} = \left[\left|\mathrm{K_f}\right| \cdot \Pi - 2\xi_R \cdot G_s / (\rho \Pi)\right] / L \tag{5.23}$$

At $\mathrm{K_{WHC}} < 0$ - operating mode is stable;

At $\mathrm{K_{WHC}} > 0$ - OHI. (5.24)

5.3. ANALYSIS OF THE RESULTS OF THE WH SIMULATION IN THE REACTOR WWER-1000 CIRCUIT

Based on the results of computational modeling on the presented conservative methods for determining the conditions of occurrence and consequences of WH in the reactor circuit, as well as structural and technical data of nuclear power plants with WWER-1000/320 serial reactors, it was established:

1. The maximum amplitude of the WH pressure pulse in the transitional trigger mode of the MCPU-195M in the AHI investigation is 2.3 MPa.

2. The maximum amplitude of the WH pressure pulse in the operating mode of the nuclear reactor as a result of the OHI is 1.1 MPa.

The calculated values of the maximum amplitude of the WH pressure pulse do not exceed the design maximum permissible dynamic loads on the equipment hulls and pipelines of the reactor circuit with seismic actions up to 9 points on the MSK scale. However, the consequences of these WHs can be the following:

1) initial nuclear accidents for nuclear safety with violation of the cooling conditions of the core RR;

2) decrease in reliability and residual life of the internal devices RR, MCPU and SG.

3) destruction of the elements of the core of RR and heat exchanger tubes SG;

4) increased vibration of the reactor circuit and others.

Additional organizational and technical measures are needed to prevent WH in the reactor circuit.

TOPIC 6. ANALYSIS OF CRITERIA OF SIMILARITY OF EXPERIMENTAL MODELS AND EQUIPMENT OF NUCLEAR PLANT SAFETY SYSTEMS

6.1. CRITERIA FOR THE SIMILARITY OF HYDRODYNAMIC PROCESSES IN REAL AND EXPERIMENTAL CONDITIONS

An analysis of the similarity criteria for hydrodynamic processes is presented on an example of a typical scheme of NPP safety systems. A typical scheme of the channel of active safety systems (ASS) with NPP pumps with WWER / PWR is shown in Fig. 6.1.

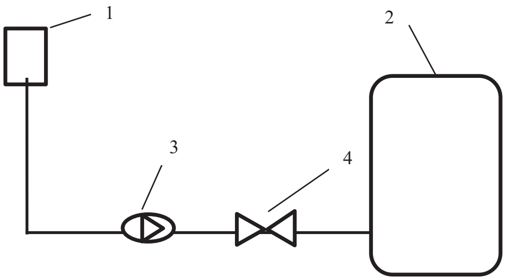


Fig. 6.1. Typical scheme of the ASS channel: 1- water capacity of the water reserve; 2- reactor / steam generator; 3- the pump; 4 - the valve.

ASS WWER - emergency cooling systems of the reactor with high and low pressure pumps; a system for feeding and injecting a solution of boric acid into the reactor; systems of emergency and auxiliary make-up of the steam generator and others.

Two characteristic modes of operation of ASS are analyzed:
- transient start-up / stopping of the pump or opening / closing the valve;
- stationary steady-state operation modes.

With conservative assumptions of fluid incompressibility and isothermal processes, the nonstationary equations of motion and energy conservation in the criterion (dimensionless) format for transient regimes have the form [5] after the transformations:

$$\frac{d\mathbf{U}}{d\mathbf{t}} = \Delta\mathbf{P}_{PO}(\mathbf{t}=0) + \int_0^{\mathbf{t}} f'(\mathbf{U})\cdot\frac{d\mathbf{U}}{d\tau}\cdot d\tau - (1-\mathbf{K}_p) - \mathbf{K}_\xi\cdot\mathbf{U}^2 \quad \mathbf{U}(\mathbf{t}=0)=0 \tag{6.2}$$

$$\frac{d}{d\mathbf{t}}\left(\frac{\mathbf{U}^2}{2} + \mathbf{K}_i\cdot\mathbf{P}\right) = 0 \tag{6.3}$$

где $\mathbf{U} = U/U_{\mathbf{s}}$; $\mathbf{t} = t\cdot \mathrm{P}_{\mathbf{out}}/(\xi L U_{\mathbf{s}})$; $\Delta\mathbf{P}_{\mathrm{po}} = \Delta\mathrm{P}_{po}(t=0)\cdot\mathrm{P}_{out}$; $f'(\mathbf{U}) = d\Delta\mathbf{P}_p/d\mathbf{U} \le 0$ - flow-rate characteristic of the pump, corresponding to the type of pump and its design and technical parameters;, $\Delta\mathrm{P}_{\mathbf{pm}}$ $\Delta\mathrm{P}_p$ - the maximum and current head of the pump, respectively; $\mathrm{P} = \mathrm{P}/\mathrm{P}_{out}$, U - average (in terms of cross-sectional area of the pipeline) flow rate; $U_{\mathbf{s}}$ - average speed in pipelines in steady-state steady mode; P, P_{ent}, P_{out} , - respectively, the pressure in the pipelines, in the hydraulic reservoirs of the water reserve and in the reactor / steam generator; t - time of the process; ρ - the density of the liquid; L - total length of pipelines; i - the specific enthalpy of the flow.

Equations (6.2), (6.3) follow the criteria for the conditions for the appearance of the WH:

$$\frac{d\mathbf{U}}{dt} < 0\,; \frac{d\mathbf{P}}{d\mathbf{t}} > 0 \tag{6.4}$$

The conditions for the similarity of real ASS and experimental installations in transient regimes are determined by the identity of the criteria:

$$\mathbf{K}_p = \frac{\mathrm{P}_{ent}}{\mathrm{P}_{out}} \equiv iden\,;$$

$$\mathbf{K}_\xi = \left[\xi_p + \xi_k(t)\right]\frac{\rho U_0^2}{2\cdot \mathrm{P}_{out}} \equiv iden\,;$$

$$\mathbf{K}_i \frac{1}{U_{\mathbf{s}}}\cdot\frac{di}{dp} \equiv iden\,;$$

$$\Delta\mathbf{P}_{po} \equiv iden\,;$$

$$f'(\mathbf{U}) \equiv iden\,;$$

$$\mathbf{t} = \frac{\mathrm{P_{out}}}{\rho \mathrm{LU_s}}\mathrm{t} \equiv iden\,; \tag{6.5}$$

where, ξ_p, $\xi_k(t)$ - the coefficients of the hydraulic resistance of the pump and valve, respectively.

The conditions for the appearance of WH (4) are a consequence of aperiodic hydrodynamic instability in the pipeline system with pumps (Figure 1) and in the general case can be determined by solving the systems of equations (6.2), (6.3) by numerical methods [5].

In steady-state stationary regimes with speed U_s and pressure $\Delta \mathrm{P}_{po}(U_s)$, the cause of the WH appearance can be oscillatory hydrodynamic instability: a random (fluctuation) change in the flow velocity δU (caused, for example, by the pump) under certain conditions can lead to pressure and flow velocity fluctuations [5].

For the system under consideration (Fig. 6.1), the inertial "delay" of the reaction of the pressure-supply characteristic (PSC) of the pump during the time Δt for fluctuational changes in the hydrodynamic parameters is the determining factor for the onset of the vibrational instability. The parameter of inertia Δt depends on the design and technical data of the pump and affects the amplitude and frequency of velocity and pressure fluctuations (in antiphase velocity) of the flow.

Under the conditions when the hydro-volumes of objects 1 and 2 (Figure 6.1) are much larger than the volume of liquid in the pipeline system by fluctuational disturbances δU $\delta \mathrm{P}$, objects 1 and 2 can be conservatively neglected. Then, the equations of flow and conservation of energy in the pipeline system in the perturbation parameters δU, $\delta \mathrm{P}$, in this case have the form:

$$\frac{d\delta\mathbf{U}}{d\mathbf{t}} = \int_0^{\mathbf{t}} f'(\Delta\mathbf{t})\cdot\frac{d\delta\mathbf{U}}{d\tau}\cdot d\tau - 2\cdot\mathbf{K}_\xi\cdot\delta\mathbf{U} + \Delta\mathbf{P}_{po}(U_s) \tag{6.6}$$

$$\mathbf{K}_i\frac{d\delta\mathbf{P}}{d\mathbf{t}} = -\frac{d\delta\mathbf{U}}{d\mathbf{t}};\ \delta U(\mathbf{t}=0) = \delta\mathbf{U}_0 \tag{6.7}$$

In the general case, solutions (6.6), (6.7) can be obtained by numerical methods, and the amplitudes and frequencies of oscillations of the hydrodynamic parameters depend on the criteria; [5].

Thus, the conditions for the similarity of real WH and experimental installations in steady-state regimes are determined by the identity of the criteria: (6.8)

$$f' = \begin{cases} 0,\ \text{при}\ \mathrm{t} \le \Delta\mathrm{t} \\ f'(\delta\mathrm{U})\,\text{при}\ \mathrm{t} > \Delta\mathrm{t} \equiv \mathrm{iden} \end{cases} \tag{6.8}$$

Practical application of the obtained similarity criteria can be demonstrated on the results of experimental studies [9]. The experimental setup [9] represents a closed circulation circuit with piston pumps and valves. As a result of the experiments carried out in [9], the amplitude of the pressure oscillations over 30% of the mean value was recorded at operating conditions. To reduce the amplitude of pressure and WH fluctuations, damping devices (DD) were installed at the pump outlet. Effective to reduce the amplitude of WH, the design and technical characteristics of remote control were determined by experimental methods.

The above analysis of the similarity criteria in real and experimental conditions [9] showed that the conditions for the similarity of the hydrodynamic processes (6.8) in the experimental setup [9] and in real ASS of nuclear power plants with WWER are not fulfilled. Therefore, the extrapolation of the experimental results to the solution of the SIS of NPP condition with WWER is unreasonable.

TOPIC 7. METHODS AND CRITERIA FOR QUALIFICATION OF PILOT-OPERATED SAFETY VALVE OF REACTOR PRESSURIZER FOR WATER HAMMER CONDITIONS

7.1. QUALIFICATION CRITERIA OF POSV SV FOR WATER HAMMER CONDITIONS OF WH1 TYPE

Basic provisions of a calculation method for qualification for WH1 conditions are the following.

1. When flows stagnation in front of the closing SV, kinetic energy of an incident flow with a rate v turns into the internal energy of a flow WH1 with a specific enthalpy i and amplitude of a WH1 pressure pulse ΔP_g.
2. Flow rate in the supply pipeline to SV (v) and discharge pipeline (v_o) is assumed as average one in the pipeline throat area Π.
3. For simplicity, we assume that all processes are isothermal.

Taking into account the accepted assumptions, the masses and energy balance equations for SV and flow equation in a supply pipeline of length L are:

$$L\frac{\delta\rho}{dt} = \rho(v - v_o), \tag{7.1}$$

$$\frac{d}{dt}\left(\rho\frac{v^2}{2} + \rho i\right) = 0, \tag{7.2}$$

$$\frac{d(\rho v)}{dt} = \left[\Delta P_0 - \xi\frac{\rho}{2}v^2\right]\frac{1}{L}, \tag{7.3}$$

Where ρ is flow medium density, t is time, ξ(t) is local hydraulic resistance coefficient, and ΔP_0 is differential pressure in the pressurizer and BT.

After transformation of combined equations (7.1) – (7.3) taking into account that

$$\frac{d(\rho i)}{dt} = \rho\frac{di}{dP}\frac{dP}{dt} + i\frac{d\rho}{dP}\frac{dP}{dt}; \ \frac{d(\rho v)}{dt} = \rho\frac{dv}{dt} + v\frac{d\rho}{dP}\frac{dP}{dt},$$

Decisions are:

$$\frac{dĐ}{dt} = \frac{\xi(t)\frac{\rho}{2}v^2 - \Delta P_0}{\rho\frac{di}{dP} + \frac{i}{c^2} - \frac{v^2}{2c^2}}\frac{1}{L}v \equiv v_p, \tag{7.4}$$

$$\frac{\mathrm{d}\nu}{\mathrm{d}t} = \frac{\left[\Delta P_0 - \xi(t)\frac{\rho}{2}\nu^2\right]\frac{1}{L} - \frac{\nu\nu_{\mathrm{p}}}{c^2}}{\rho}, \tag{7.5}$$

$$v_0 = v - \frac{1}{L}v_{\mathrm{p}}, \tag{7.6}$$

Where $v_{\mathrm{p}} = \mathrm{d}P/\mathrm{d}t$ *is local rate of pressure increase characterizing intensity of WH1 on closing SV, and* $c = \sqrt{\mathrm{d}P/\mathrm{d}\rho}$ *is speed of disturbance propagation (sound speed) in a flow.*

Maximum load on operating SV elements under WH1 is:

$$\Delta \text{Đ}_{\mathrm{g}} = \int_0^{t_{\mathrm{g}}} v_{\mathrm{p}}(\tau)\mathrm{d}\tau, \tag{7.7}$$

Where t_g *is a SV closing time.*

By tradition, the known Joukowski formula is applied to determine a maximum WH load on closing armature [6]:

$$\Delta \text{Đ}_{\mathrm{g}} = \rho(v - v_0)\tilde{n}_{\text{т}}. \tag{7.8}$$

In particular, the authors of [4] have used formula (7.8) to calculate ΔP_{g} of water hammer when the pressurizer is filling. Thus, calculated values of the maximum amplitudes ΔP_{g} are much underestimated versus experimental data.

As appears from the obtained solution (7.4), (7.7), traditional approach to an estimated maximum load ΔP_{g} using a formula (7.8) does not consider real conditions for WH1 generating, structural and technical characteristics of SV and a pipeline system as a whole, thermal and physical properties of a flow and other factors. Besides, it should be noted that the obtained solution (7.4), (7.7) determines the greater dependence of a maximum WH1 load on SV on the incident flow rate $\Delta P_{\mathrm{g}} \sim \nu^3$ than a formula (7.8): $\Delta P_{\mathrm{g}} \sim \nu$.

WH1 condition follows from the equations (7.4), (7.5):

$$\xi(t) > \frac{2\Delta P_0}{\rho\nu^2}. \tag{7.9}$$

Change in hydraulic resistance ξ when SV is closing directly depends on the closing speed of the sliding piston: $\nu_{\mathrm{h}} = \mathrm{d}h/\mathrm{d}t$. For average (by time) speed v_{h}:

$$\xi(t) = \xi_0 + \frac{1}{h_0}v_{\mathrm{h}}t, \tag{7.10}$$

Where ξ_0, h_0 *are initial value of* ξ *and a sliding piston position h when SV is fully open, respectively.*

Taking into account the equations (7.9), (7.10) necessary condition for WH1:

$$v_{\mathrm{h}} \geq v_{\mathrm{hm}} = \left(\frac{2\Delta P}{\rho v^2} - \xi_0\right) + \frac{h_0}{t_{\mathrm{g}}} \qquad (7.11)$$

Thus, qualification criterion of POSV system for water hammers condition when SV is closing (WH1):

$$v_{\mathrm{h}} < v_{\mathrm{hm}}. \qquad (7.12)$$

As follows from the found criterion (7.12), operational test programs for POSV SV must include diagnostics of speed and time of valve closing.

7.2. QUALIFICATION CRITERIA OF POSV SV FOR WATER HAMMER CONDITIONS OF WH2 TYPE

Basic provisions and assumptions of a method for the POSV SV qualification for WH2water hammer conditions are the following.

1. Profiles of throat area of the SV convergent and divergent sections are modelled by linear approximations according to design and operational documentation. The SV minimum throat area corresponds to the sliding piston location in state "open".

2. The two-phase flow in the SV throat areas is modelled by one-dimensional homogeneous model [5].

3. When the average two-phase flow rate reaches sonic speed (the Mach criterion M = 1 – the transonic mode), condensation pressure jump (CPJ) generates in the minimum SV throat area. It corresponds to WH2 conditions (see Fig. 1) Under WH2 conditions kinetic energy of a flow stagnation turns into a WH2 pressure pulse energy (with the maximum amplitude ΔP_{g}).

WH2 generation can lead to a considerable delay of the SV closing time because of pulse reduction of the coolant flow in BT and additional loads on the SV sliding piston.

Under the accepted assumptions, the fundamental equations of conservation laws of a two-phase flow [5]:

$$\frac{\mathrm{d}\rho\Pi v}{\mathrm{d}z} = 0, \qquad (7.13)$$

$$\frac{\mathrm{d}\rho\Pi v^2}{\mathrm{d}z} = -\Pi\frac{\mathrm{d}P}{\mathrm{d}z}, \qquad (7.14)$$

$$\frac{\mathrm{d}}{\mathrm{d}z}\left(\rho i + \frac{\rho v^2}{2}\right) = 0, \qquad (7.15)$$

$$v(z=0) = v_0, \qquad (7.16)$$

$$P(z=0) = P_0, \qquad (7.17)$$

$$i(z=0) = i_0, \qquad (7.18)$$

Where v, P are speed and pressure of a two-phase homogeneous flow, respectively, ρ(R, x) is density of a two-phase flow, i(R, x) is specific enthalpy of a two-phase flow, and x is mass steam content of a two-phase flow, Π is current SV throat area.

After transformations, the decision (7.13) – (7.18) in a dimensionless format:

$$\frac{\mathrm{d}\mathbf{P}}{\mathrm{d}\mathbf{z}} = \frac{\mathbf{K}_{\mathrm{v}}\boldsymbol{\rho}\mathbf{v}^2}{(1-\mathbf{M}^2)}\mathbf{K}_{\Pi} = \mathrm{grad}(\mathbf{P}), \tag{7.19}$$

$$\frac{\mathrm{d}\mathbf{v}}{\mathrm{d}\mathbf{z}} = -\mathrm{grad}(\mathbf{P})\frac{\mathbf{v}}{\boldsymbol{\rho}}\left(\frac{\partial\boldsymbol{\rho}}{\partial\mathbf{P}} + \frac{1}{2r}\frac{\partial\boldsymbol{\rho}}{\partial x}\right) - \mathbf{v}\mathbf{K}_{\Pi}, \tag{7.20}$$

$$\frac{\mathrm{d}x}{\mathrm{d}\mathbf{z}} = -\frac{i_0}{2r}\mathrm{grad}(\mathbf{P}), \tag{7.21}$$

$$\mathbf{P}(\mathbf{z}=0)=1;\quad \mathbf{v}(\mathbf{z}=0)=1;\quad x_0(\mathbf{z}=0)=(i_0-i_1)/r \tag{7.22}$$

Where $\boldsymbol{P} = P/P_0$, $\boldsymbol{z} = z/L$ *(L is length of convergent section),* $\boldsymbol{\rho} = \rho/\rho_0$, $\boldsymbol{v} = v/v_0$, $\boldsymbol{i} = i/i_0$, $\boldsymbol{\Pi} = \Pi_0/\Pi$.

From (7.19) – (7.21), the key criteria of the transonic flow modes:

$$\mathbf{K}_{\mathrm{v}} = \frac{\rho_0 v_0^2}{P_0};\quad \mathbf{K}_{\Pi} = \frac{1}{\boldsymbol{\Pi}}\frac{\mathrm{d}\boldsymbol{\Pi}}{\mathrm{d}\mathbf{z}};\quad \mathbf{M} = \frac{v}{a}\ \text{– Mach criterion,} \tag{7.23}$$

Where $a = \sqrt{\dfrac{\partial\rho}{\partial P} + \dfrac{1}{2r}\dfrac{\partial\rho}{\partial x}\left(1 - x\dfrac{\partial i}{\partial P}\right)}$ *is sonic speed in a two-phase homogeneous equilibrium flow, and* $r = i_{\mathrm{v}} - i_1$ *is latent heat of steam generation.*

The found decisions (7.19) – (7.22) are resulted in followings: pressure and steam content of a flow decrease longwise the channel, and flow rate increases in convergent SV section ($\mathbf{K}_{\Pi} < 0$). When flow rate reaches transonic values ($\mathbf{M} \to 1$), modules of flow rate gradients increase sharply; and when flow rate reaches transonic mode ($\mathbf{M} = 1$), there is the subsequent sharp stagnation of a flow ($v \to 0$) and the local pulse growth of pressure and condensation of a steam-liquid flow (CPJ as a result of aperiodic thermo-hydrodynamic instability of the transonic modes of steam-liquid flows). Thus, a necessary condition for WH2 on the equipment of pipeline systems in the transonic modes of two-phase flows is:

$$\mathbf{M} = 1. \tag{7.24}$$

For modelling the WH2 parameters (a hydrodynamic load ΔP_{g}, rates of change of P and v), it is reputed that when stagnation kinetic energy of a two-phase flow transfers into CPJ energy. Then the balance equations of mass, pressure pulse and energy into CPJ:

$$L\frac{\mathrm{d}\rho}{\mathrm{d}t} = \rho v - \rho v_{\mathrm{c}}, \tag{7.25}$$

$$L\frac{\mathrm{d}\rho v}{\mathrm{d}t} = -\Delta P_{\mathrm{g}}(t), \tag{7.26}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\rho v^2}{2} + \rho i\right) = 0, \tag{7.27}$$

Where t is time, v_c *is SV output flow rate, L is channel length before the section of the transonic mode, and* $\Delta P_g(t)$ *is amplitude of a hydrodynamic WH2 load.*

Assumed conservatively that CPJ processes are isothermal, steam condensation is "instantaneous", and a flow rate after CPJ against a SV input flow rate is negligible, after transformation (7.25) – (7.27) in a criteria form:

$$\frac{d\mathbf{P}}{d\mathbf{t}} = \mathbf{v}, \tag{7.28}$$

$$\frac{d\mathbf{v}}{d\mathbf{t}} + \mathbf{K}_1 \mathbf{v} \frac{d\mathbf{P}}{d\mathbf{t}} = -\Delta \mathbf{P}_g(\mathbf{t}), \tag{7.29}$$

$$\mathbf{v}\frac{d\mathbf{v}}{d\mathbf{t}} + \frac{\mathbf{K}_1}{2}\mathbf{v}^2 \frac{d\mathbf{P}}{d\mathbf{t}} + \mathbf{K}_2 \frac{d\mathbf{P}}{d\mathbf{t}} = 0, \tag{7.30}$$

Under initial conditions

$$\mathbf{v}(\mathbf{t}=0) = \frac{a}{v_m} = \mathbf{v}_{g0}, \ \mathbf{P}(\mathbf{t}=0) = \mathbf{P}_{g0}, \ \mathbf{i}(\mathbf{t}=0) = \mathbf{i}_{g0}, \tag{7.31}$$

Where

$$\mathbf{P} = \frac{P}{P_m}, \quad \mathbf{v} = \frac{v}{v_m}, \quad \mathbf{t} = \frac{t}{t_m}, \quad \mathbf{i} = \frac{i_l}{i_m},$$

$$\mathbf{K}_1 = \frac{P_m}{\rho_l a_l^2}, \quad \mathbf{K}_2 = \frac{d\mathbf{i}}{d\mathbf{P}}, \tag{7.32}$$

ρ_l, a_l are density and sonic speed of a fluid phase, respectively.

Scales of WH2 parameters follow from transformations of the equations (7.28), (7.29) to a criteria form:

$$P_m = P_0; \ v_m = \frac{P_0}{\rho_l a_l}; \ t_m = \frac{L}{a_l}; \ i_m = \frac{P_0^2}{\rho_l a_l^2}.$$

The maximum amplitude of a hydrodynamic WH2 load:

$$\Delta \mathbf{P}_g(\mathbf{t}) = \int_0^{\mathbf{t}} \mathbf{v}(\tau) d\tau, \tag{7.33}$$

$$\frac{d\mathbf{v}}{d\mathbf{t}} + \frac{\mathbf{K}_1}{2}\mathbf{v} + \mathbf{K}_2 = 0, \quad \mathbf{v}(\mathbf{t}=0) = \frac{\rho_l a_l a}{P_0}. \tag{7.34}$$

Sufficient condition for WH2 in the transonic modes of a two-phase steam-liquid flow:

$$\Delta \mathbf{P}_g(\mathbf{t}=1) \geq 1. \tag{7.35}$$

Generally, numerical methods of modelling are useful to decide the equations for definition of conditions and consequences of WH2 [5].

The calculated area of WH2 conditions in a format of the key criteria $\mathbf{K}_v$ and $\mathbf{K}_\Pi$ is presented in Fig. 7.2. There is presented well-known experimental data on CPJ (see, for example, [1, 7, 8], etc.) obtained in relatively "short" convergent-divergent channels.

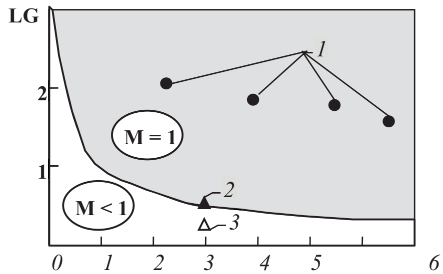


Fig. 7.2. Area of conditions for water hammers as consequences of aperiodic instability into transonic two-phase steam-liquid flows: 1 – experiment; 2 – MSV VS-99 at a nominal reactor power; 3 – MSV VS-99 in "hot" reactor shutdown tests.

Values of criteria $\mathbf{K}_v$ and $\mathbf{K}_\Pi$ for Sempell MSV VS-99 are on WH2 area boundary in a nominal operating mode of WWER (*2* in Fig. 2).

Values of criteria $\mathbf{K}_v$ and $\mathbf{K}_\Pi$ for MSV VS-99 are out of the WH2 area in "hot" shutdown tests (*3* in Fig. 2). This disagreement is because coolant temperature in tests is 40 – 60 ° less coolant temperature in a nominal operating mode of the reactor. Results of known experiments in convergent-divergent channels have found that decrease in coolant temperature T_0 at an entrance to a convergent part very affects CPJ conditions. Decrease in T_0 at an entrance to MSV leads to boiling up of the coolant in sections of a convergent part more remote from an entrance, and to change of CPJ conditions. At sufficiently low temperature T_0, a two-phase flow in MSV convergent part and necessary conditions for WH2 cannot be generated at all.

Thus, operational test programs for POSV SV must ensure that the coolant temperature meets a nominal operating mode of the reactor.

CONCLUSIONS

1. Inertia of sensitivity of the flow (network) characteristic of force pumps of pipeline systems is the dominating factor for heat-hydrodynamic instability conditions and critical hydraulic impacts. When sensitivity of the flow characteristic is minimum (critical) total amplitudes of the rate flow reach critical values and an inoperative failure of pump elements occurs.

Appearance of critical hydraulic impacts corresponds to transfer of oscillatory heat-hydrodynamic instability to the aperiodic one.

2. The hydrodynamic resistance of a pipeline system is the stabilizing factor to prevent conditions of heat-hydrodynamic instability with the increased amplitudes of hydrodynamic parameters.

However the main restrictions of technical solutions for "artificial" increase of hydrodynamic resistance in the supply part (to the pump) of a pipeline system are related to risk of cavitation effects, and in a pressure part – with risk of "choking" of a flow in object of consumption.

3. Using of pumps with the most sensitive flow (network) characteristic (with other equal technical capabilities) is perspective approach to prevent critical hydraulic impacts.

4. Results of known experimental and computed studies of conditions for water hammers in the reactor pressurizer are analysed. It is found that when emergency opening of pilot-operated safety valves of the WWER pressurizer, water hammers on pressurizer elements can be generated, and

amplitudes of hydrodynamic load pulses override significantly limiting values.

5. The original method for determination of water hammer conditions and consequences in case of emergency filling of the WWER pressurizer is proposed. Unlike traditional Joukowski formula, the presented method considers the key features/effects of forming of conditions and consequences of water hammers in the WWER pressurizer.

6. Results of computer modelling of the maximum amplitudes of a pressure pulse when water hammers using the presented method are in satisfactory agreement with the Prof. Korolev's experiments on the WWER-440 pressurizer model.

7. Variation calculations using the presented method have found that the considerable increase in total hydraulic resistance of internal elements can be effective action to prevent water hammers in the WWER pressurizer.

8. The analysis of well-known studies in modelling of conditions for water hammers in equipment and elements of pipeline systems has found that definition of conditions and parameters of water hammers in the transonic modes of single- and two-phase flows (at a speed of propagation of acoustic disturbances) is the least studied problem.

9 The original method is proposed for determining the conditions and parameters of water hammers in transonic flow modes subject to the transition of the kinetic energy of the flow stagnation into the energy of the water hammer pulse.

10. It is found the simulated hydrodynamic loads in transonic modes can significantly exceed the corresponding known recommendations of N. Joukowski.

11. The computational modelling of the equations of the proposed method has determined the range of the criteria for water hammers due to aperiodic thermo-hydrodynamic instability in transonic flow modes.

12. Results of known experimental and computed studies of conditions for water hammers in the reactor pressurizer are analysed. It is found that when emergency opening of pilot-operated safety valves of the WWER pressurizer, water hammers on pressurizer elements can be generated, and amplitudes of hydrodynamic load pulses override significantly limiting values.

13. The original method for determination of water hammer conditions and consequences in case of emergency filling of the WWER pressurizer is proposed. Unlike traditional Joukowski formula, the presented method considers the key features/effects of forming of conditions and consequences of water hammers in the WWER pressurizer.

14. Results of computer modelling of the maximum amplitudes of a pressure pulse when water hammers using the presented method are in satisfactory agreement with the Prof. Korolev's experiments on the WWER-440 pressurizer model.

15. Variation calculations using the presented method have found that the considerable increase in total hydraulic resistance of internal elements can be effective action to prevent water hammers in the WWER pressurizer.

16. Methods are developed for determining the conditions for the occurrence and consequences of hydraulic shocks in the reactor circuit of nuclear power installations as a result of aperiodic and oscillatory hydrodynamic instability caused by the inertia of the pressure-supply characteristic of the main circulation pump. In contrast to the known approaches to determining the consequences of hydraulic shocks, the proposed methods take into account the design and technical parameters of the equipment of the reactor circuit and the conditions for the transfer of the kinetic energy of the flow of coolant flow to the energy of the hydrostatic pressure pulse.

17. Based on the proposed methods, the hydraulic shock amplitudes in the reactor circuit are determined when the pump is started due to aperiodic hydrodynamic instability - 2.3 MPa; and in the operating mode due to oscillatory hydrodynamic instability - 1,1 MPa. The calculated values of the maximum pressure hammer pressure amplitudes do not exceed the dynamic loads with seismic actions up to 9 points on the MSK scale. However, the consequences of these water hammers can lead to a decrease in the reliability of the integrity and residual life of equipment and pipelines of the reactor circuit; to the initial for the nuclear safety initial emergency events with violation of the cooling conditions of the core of the reactor; to increased vibration and other negative effects.

18. An effective measure to prevent water hammer in the reactor circuit with WWER is to reduce the inertia of the main circulation pump.

19. A criterial method is proposed for analyzing the adequacy of real pipeline systems with pumps of nuclear power plants and experimental installations. The method is based on an analysis of the identity of the determining criteria for the similarity of hydrodynamic processes in real and experimental conditions.

20. The criteria for similarity of real and experimental conditions and conditions of hydrodynamic impacts for pipeline systems with pumps of nuclear power plants in transient and operating modes are determined. Hydrodynamic shocks in transient regimes are a consequence of aperiodic hydrodynamic instability of the flow; and in operating conditions - a consequence of oscillatory hydrodynamic instability. The determining factor of hydrodynamic oscillatory instability is the inertia of the pressure-supply characteristic of pumps.

21. On the basis of the proposed method, an example of the practical application of the similarity criteria obtained for real active safety systems and an experimental plant is presented A.V. Korolev. It is shown that the necessary conditions for identically of similarity criteria are not met and extrapolation of the results of known experiments to real conditions of active safety systems of nuclear installations with WWER reactors is not justified.

22. An original method for qualification of the pilot-operated safety valves of the reactor's pressurizer is proposed for the water hammer conditions when valves are closing. It is found that an aperiodic hydrodynamic instability in case of accelerated valve closing is the cause for water hammers. A transfer of kinetic energy to water hammers pulse energy accompanies this instability when flow stagnation.

The maximum closing speed of the valves, which depends on the differential pressure in the pressurizer and the bubbler tank, the velocity of the coolant in the supply pipelines, the local coefficient of hydraulic resistance of the valves and other parameters, is qualification criterion.

23. An original method for qualification of safety valves is proposed for the water hammer conditions in for two-phase flows. It is found that the transonic modes of a two-phase flow in case of aperiodic thermo-hydrodynamic instability can be the cause for water hammer in such conditions. A pulse transfer of kinetic energy to water hammers energy accompanies this instability when flow stagnation.

In the format of key qualification criteria, it is determined that under the nominal operating conditions of the reactor, the main safety valve is at the boundary of the water hammer area in two-phase flows.

24. The results of the qualification by the presented methods have determined that valve test programmes must include the additional diagnostics of speed and time of valve closing, and ensure that the coolant temperature meets the nominal operating mode and tests of the reactor.

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