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DYNAMIC LOADS BY VARIOUS WATER HAMMER PHENOMENA

DINAMIČKA OPTEREĆENJA OD RAZLIČITIH OBLIKA VODENOG UDARA

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Abstract

Water hammer phenomena can be induced both in one-phase and two-phase systems during pipelines transients. The basic mechanisms of fluid dynamics that lead to the water hammer are presented. Dynamic loads induced by several water hammer events are presented and analyzed. It is shown that water hammers in two-phase systems can lead to much higher pressure peaks than water hammers in one-phase systems.

Ključne reči
- vodeni čekić
- dvofazno strujanje
- kondenzacija
- numerička simulacija

Izvod

Vodeni udar se može pojaviti kako u jednofaznom tako i u dvofaznom strujanju kroz cevovode. Razmatrani su osnovni mehanizmi koji dovode do pojave vodenog udara. Prikazana je i analizirana pojava dinamičkih opterećenja koja su izazvana različitim vrstama vodenih udara. Pokazano je da vodeni udar u dvofaznom sistemu može prouzrokovati mnogo veće pikove visokog pritiska nego vodeni udar u jednofaznom sistemu.

INTRODUCTION

Transient fluid flows in pipelines can be induced by action of various control and safety valves or malfunction of plant equipment. Typical examples are rapid closure or opening of valves or pumps trip. During these events pressure waves are generated and they propagate along the pipelines, with the possibilities of waves' superposition. In such conditions substantial pressure changes can occur which are known as water hammers. Water hammer in one-phase fluid flows is widely investigated and appropriate engineering tools for its predictions are developed. Simulation and analyses of one-phase fluid flow transients in pipelines are standard procedures during plants design and operational safety analyses. The results of these analyses are input data for the design of control and safety system, defining of operational procedures and analyses of pipelines and pressure vessel dynamic loads, /1, 2/.

The operational experience of heat power plants has shown the occurrence of water hammer accidents in vapour and liquid two-phase systems. These types of water hammers have led to very high pressure peaks, much higher than in cases of one-phase transients, with devastating consequences. The occurrence of some accidents has been highly unexpected, since they have been initiated at low pressure levels. Two-types of water hammers in two-phase systems are considered in this paper, the condensation induced water hammer and the liquid slug impact.

Direct contact between subcooled liquid and steam can lead to condensation induced water hammer, /1, 3-5/. The condensation occurs at the interface between water and steam. During the phase change from vapour to liquid, the specific volume drastically decreases and steam pressure drops, causing water column movement towards the steam. The steam condensation continues towards the propagating water-steam interface. The water column accelerates until it hits into the obstacle (e.g. a closed end of the pipe, a closed valve, or another water column). The water column velocity is decreased to zero when the water column hits the obstacle, which leads to the high pressure peak at the surface of water splashing. In case of intensive condensation, a high velocity of the water column can be developed and a destructive pressure increase can be induced. According to available measured data, maximal pressure peaks induced by intensive condensation and subsequent water hammer splashing could reach values of 10 MPa and even higher, although the initial pipe pressure is equal to the atmospheric pressure. Such a pressure peak can fracture a cast-iron valve or blow out a steam gasket. Amplitudes of the pressure wave, induced by direct condensation of steam onto the head of the water column, may cause damage of equipment which could seriously impact the safety of personnel.

There are several characteristic two-phase flows with a high possibility of water hammer occurrence: the steam–water flow in the horizontal pipe, the steam–subcooled water flow in the vertical pipe, the pressurized water inflow in the vertical pipe filled with steam, and the hot water inflow in the cold low pressure pipeline. In the period between 1969 and 1981 over one hundred reported water hammer incidents occurred in nuclear power plants in USA, /3/. In the district heating system in Fort Wainwright, Alaska, /4/, the same cause led to the hot water and steam explosion, and cracking of valve and steam line burst.

The second type of water hammer in two-phase system is induced by the condensate slug formation, acceleration and splashing onto the closed pipe's end. This event occurs in steam lines at power plant start-up procedures, /5/.

Water hammer phenomena in complex pipeline networks are successfully predicted with numerical models based on transient compressible gas flow and appropriate mass, momentum and energy balance equations in the form of hyperbolic partial differential equations. These equations can be effectively solved numerically with the application of the method of characteristics, as it was presented in /2, 6/. The predictions of water hammers in two-phase systems are much more complex due to the presence of irregular gas-liquid interfaces, intensive condensation of vapour onto the subcooled liquid surfaces, and rapid changes of fluid's thermophysical properties at the interfaces. A numerical procedure for the gas-liquid interface tracking is developed for gas-liquid interface tracking and condensation, and it was successfully applied to both condensation induced and slug impact water hammer simulations, /7, 8/.

In this paper several examples of water hammer events and accidents are presented that happened in components of heat power plants. Also, the analytical tools for their prediction are presented. The dynamic loads on the pipeline structure induced during the water hammer phenomena are analysed.

WATER HAMMER IN ONE-PHASE SYSTEM

Modelling approach

Fluid transient in one-phase, one-dimensional, compressible fluid flow in the flow channel with rigid walls is described with the mass, momentum and energy balance equations:

– mass balance

$$\frac{D\rho}{Dt}+\rho\frac{\partial u}{\partial x}=0 \tag{1}$$

– momentum balance

$$\frac{Du}{Dt}+\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{fu|u|}{2D_H}+g\sin\theta=0 \tag{2}$$

– energy balance

$$\frac{Dh}{Dt}-\frac{1}{\rho}\frac{Dp}{Dt}-\frac{fu^2|u|}{2D_H}+\frac{\dot{q}}{\rho}=0 \tag{3}$$

where ρ denotes density, u is fluid velocity, p is pressure, h is specific enthalpy, t is time, x denotes spatial coordinate, f is friction coefficient, q denotes exchanged heat flux per unit of mass, D_H is the flow channel hydraulic diameter, and Θ denotes flow channel inclination. The set of balance equations (1) to (3) is closed by the equation of state in the form of the appropriate polynomial.

This system of equations is solved for the prescribed initial and boundary conditions. Initial conditions determine the pressure, velocity and temperature (enthalpies) in the gas network in the initial time instant before the distribution action. Boundary conditions describe the action of safety, control and technical components, such as various kinds of valves, pipeline junctions, compressor units. Equations (2) to (4) are solved numerically by the method of characteristics which is based on the physics of mechanical disturbance propagation in the form of pressure waves by the speed of sound and enthalpy propagation by fluid particle flow. The method of characteristics transforms the partial differential equations in the form of ordinary differential equations that holds along the characteristic paths determined by pressure waves propagations and fluid particle movement in the time-space coordinate system. Obtained ordinary differential equations are approximated with finite difference equations that are solved explicitly for every new time step of integration. The time step of integration is determined by the Courant criterion

$$\Delta t\le\min\left(\frac{\Delta x}{c_j+|u_j|}\right) \tag{4}$$

where j indicates a pipe within the network and Δx is the distance between two adjacent numerical nodes.

In order to simulate complex pipe networks and various transient scenarios, several models of boundary conditions (i.e. flow channel discontinuities) are developed that can be linked in a modular way. Calculation of flow parameters at the ends of a pipe must be done using additional hydraulic models. These additional equations describe the mass, momentum and energy balance at a point of discontinuity, and they replace the equations of the characteristics which do not belong to the physical domain of a pipe. The following boundary conditions are included in the code: a subcritical or critical leakage from a pipe, a closed end of a pipe, a pipe joining a reservoir, a junction of two or more pipes, a valve in a pipe, and a pump.

The estimation of pressure changes in transient fluid flow can be done with the simple relation attributed to Zhukovsky, /9/. By neglecting wall friction and gravity, and observing isothermal flow conditions, the momentum equation (2) can be reduced to the following simple algebraic relation along the path of pressure propagation with sonic velocity $c = dx/dt$

$$\Delta p=-\rho c\Delta u \tag{5}$$

According to /5/, an instantaneous decrease of cold water flow velocity from 2 m/s to zero will result in a significant pressure increase of 2 MPa

$$\Delta p=-1000(\text{kg/m}^3)\times1000(\text{m/s})\times(0-2)(\text{m/s})=2\,\text{MPa} \tag{6}$$

where assumed sonic velocity of 1000 m/s is for technical water and density of 1000 kg/m^3. In case of gas flow, the pressure changes due to velocity changes are lower. For instance, in case of superheated steam flow at 50 m/s in a steam pipeline at 5 MPa and 540°C, the pressure increase due to rapid isolation valve closure will result in

$$\Delta p=-14(\text{kg/m}^3)\cdot500(\text{m/s})\cdot(0-50)(\text{m/s})=0.35\,\text{MPa} \tag{7}$$

The propagation of pressure waves in a pipeline leads to the induction of transient fluid dynamic forces in pipes which are bounded by other flow components, such as elbows or closed ends, /10/. These transient wave forces act along the straight pipe axis. Their positive direction is assumed to be opposite to the fluid flow direction. It is calculated by the following expression

$$F = \int_L \frac{d\dot{m}}{dt} dx \qquad (8)$$

where $\dot{m}$ denotes fluid mass flow rate.

The fluid force exerted on a pipe with an open end (expulsion from a pipe) includes both a wave force and a blowdown component associated with momentum expulsion and the difference between discharge and ambient pressure. It acts along the pipe axis and the force intensity is

$$F = \int_L \frac{d\dot{m}}{dt} dx + \left[(p_l - p_{atm}) + \rho_l u_l^2 \right] A \qquad (9)$$

where index l denotes parameters at the location of leakage, p_{atm} is atmospheric pressure and A is the area of leakage.

Case study-Dynamic loads in the steam pipeline

The simulation of the real fluid flow transient is performed for the reheated steam pipeline at the Thermal Power Plant "Nikola Tesla B" in Obrenovac (Fig. 1).

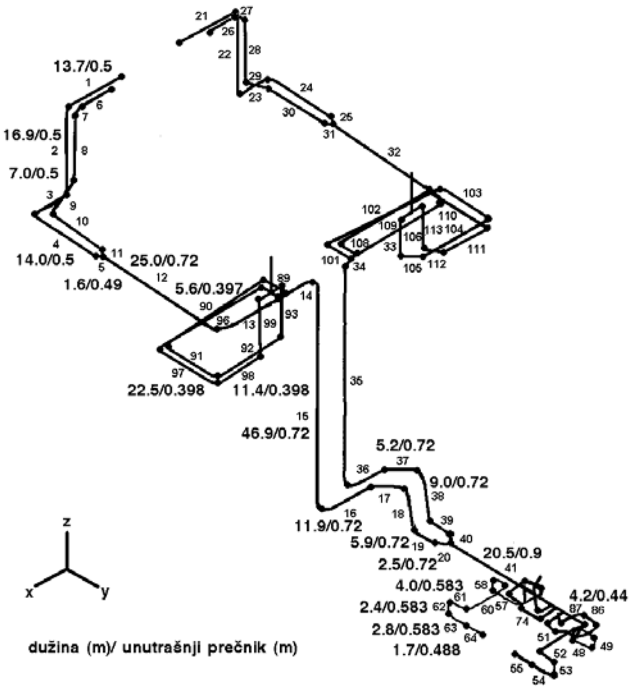

Figure 1. Isometric drawing of the reheated steam pipeline at the thermal power plant "Nikola Tesla B".
Slika 1. Izometrijski crtež cevovoda međupregrejane pare u Termoelektrani "Nikola Tesla B"

Positions of the isolation valves in front of the middle pressure steam turbine are shown in Fig. 2. During the turbine trip the isolation valves are closed very fast within 0.2 s which results in the rapid steam flow decrease and the induction of pressure waves propagation. The velocity and pressure changes during this transient in front of the isolation valve are shown in Figs. 3 and 4, respectively. An example of transient fluid dynamic forces is shown in Fig. 5. These data are input for the design of pipeline supports and pipeline structure stress analyses. Obviously the proper treatment of these dynamic forces in the design state are crucial for system integrity and safety.

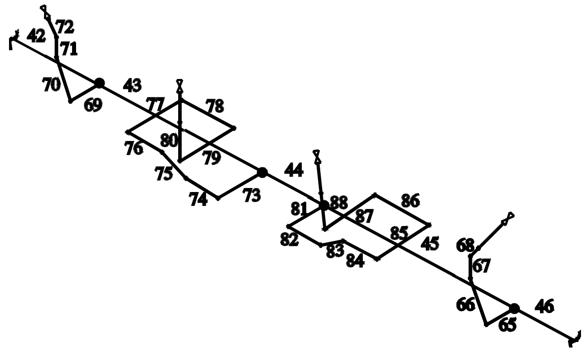

Figure 2. Steam pipeline section in front of isolation valves.
Slika 2. Shema sekcije cevovoda ispred pregradnog ventila

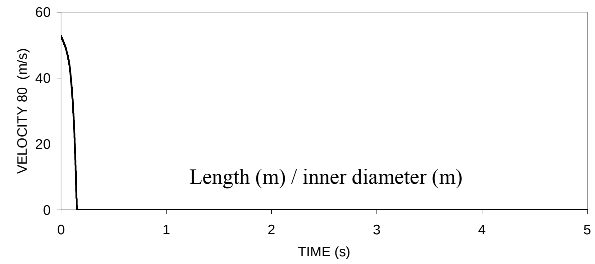

Figure 3. Velocity change in front of isolation valve.
Slika 3. Promena brzine ispred pregradnog ventila

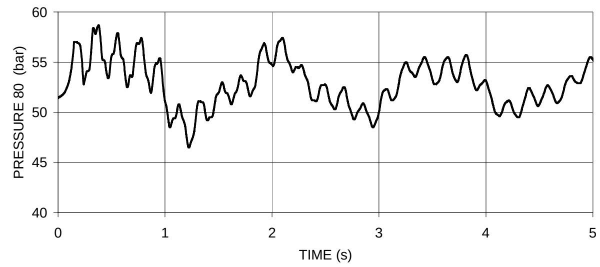

Figure 3. Pressure change in front of isolation valve.
Slika 3. Promena pritiska ispred izolacionog ventila

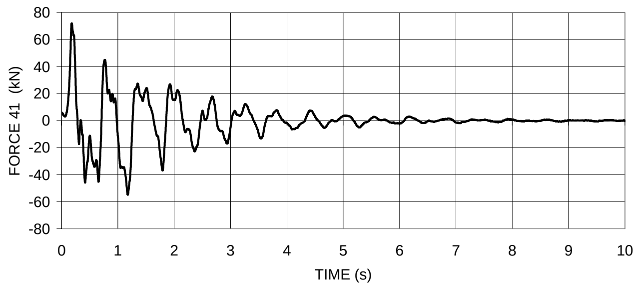

Figure 5. Fluid dynamic force in steam pipeline segment 41 (depicted in Fig. 1 as the common header upstream to steam input lines to the turbine).

Slika 5. Dinamička sila fluida u segmentu 41 parovoda (označena na sl. 1 kao sabirnik sveže pare na ulazu u turbinu)

WATER HAMMER IN TWO-PHASE SYSTEM

Modelling approach

Two-phase transient flows in pipes are described with the homogeneous model, where both velocity and thermal equilibrium are assumed in case of two-phase flow, /6/. In order to describe the important effects of transient friction on the pipe's wall and the vapour condensation at the interface with the cold liquid, the balance equation (1, 2, 3) are stated in the following form

– mass conservation

$$\frac{\mathrm{D}\rho}{\mathrm{D}t}+\rho\frac{\partial u}{\partial x}=0 \tag{10}$$

– momentum conservation

$$\frac{\mathrm{D}u}{\mathrm{D}t}+\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{fu|u|}{2D_{\mathrm{H}}}+f_t\,\mathrm{sign}(u)\left|\frac{\partial u}{\partial t}\right|+g\sin\theta=0 \tag{11}$$

– energy conservation

$$\frac{\mathrm{D}h}{\mathrm{D}t}-\frac{1}{\rho}\frac{\mathrm{D}p}{\mathrm{D}t}-\frac{fu^2|u|}{2D_{\mathrm{H}}}-f_t u\,\mathrm{sign}(u)\left|\frac{\partial u}{\partial t}\right|+ \\ +\int_{x_i-\varepsilon}^{x_i+\varepsilon}\frac{\dot{q}_A a_i}{\rho}\delta\left(x-x_i\right)dx=0 \tag{12}$$

The last term in the energy equation (11) describes heat flux per unit of mass due to phase change, where

$$\dot{q}_{\mathrm{A}}=h_{\mathrm{cond}}(T_g-T_1) \tag{13}$$

is the heat flux per unit of interface area, a_i is the specific water-steam interface area, $\delta(x-x_i)$ is Delta function that has characteristic $\int_{x_i-\varepsilon}^{x_i+\varepsilon}\delta(x-x_i)dx=1$, where ε is the short thickness at the interface surrounding. The interface is located at x_i. The fourth terms on the left hand sides of the momentum and energy equations take into account the influence of transient friction, /11/. This system of equations is solved with the procedure based on the method of characteristics, while the higher order numerical scheme is applied for solving of the energy equation in order to achieve accurate prediction of vapour-liquid interface tracking, /8, 12/. The time step of integration is also determined with (4), where the sonic velocity in homogeneous equilibrium flow is calculated as a function of pressure and enthalpy

$$c=\sqrt{\frac{1}{\left(\frac{\partial\rho}{\partial p}\right)_h+\frac{1}{\rho}\left(\frac{\partial\rho}{\partial h}\right)_p}} \tag{14}$$

Case studies – Condensation induced water hammer

The condensation induced water hammer is experimentally investigated in a test facility presented in Fig. 6. A horizontal pipe of 5.5 m length and 0.0921 m diameter is connected to the bottom of the vertical tank. Both the tank and horizontal pipe are initially filled with subcooled water. A fast-acting ball valve is installed between horizontal and vertical pipe. The vertical pipe is initially filled with steam. Its length is 5 m, diameter 0.0921 m, and it is closed at the top end. Pressure transducers were mounted in the vertical pipe to record the pressure transient during the experiment. The transient is initiated by the ball valve fast opening. Subcooled water comes into contact with steam and the intensive direct condensation of steam onto the head of the water column occurs. Initially, the steam pressure decreases due to its condensation and the water column is accelerated toward the closed end of the vertical pipe. Upon steam condensation, the water column splashes onto the closed end and impulse pressure rise occurs, with an amplitude above hundred bars, although the initial pressure is only 6.5 bar, as shown in Fig. 7. The generated pressure wave propagates from the closed end of the vertical pipe to the water vessel, reflects at the vessel water mass and enters the vertical pipe again. This process repeats periodically. The pressure amplitudes are attenuated due to the friction on the pipes' walls and their elastic deformations. Obtained results are in agreement with the measured values.

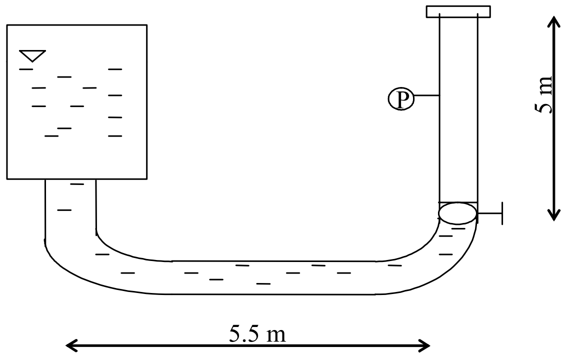

Figure 6. Schematic of the water hammer test facility.
Slika 6. Shema uređaja za ispitivanje vodenog udara

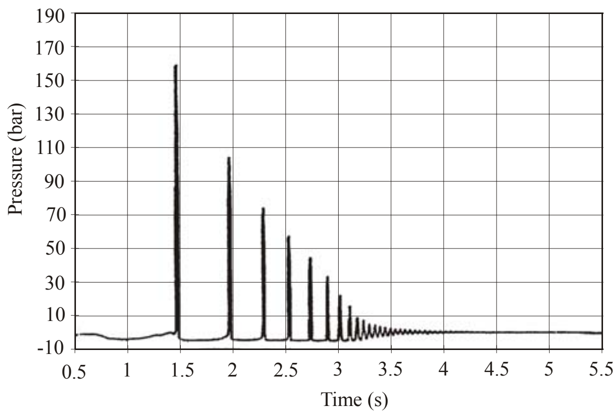

Figure 7. Measured pressure peaks during the condensation induced water hammer.
Slika 7. Pikovi pritiska izmereni pri vodenom udaru izazvanog kondenzacijom pare

The calculated pressure peaks with here outlined model are presented in Fig. 8 and good agreement is achieved with the measured results presented in Fig. 7.

Case study – Condensation induced water hammer

The other mechanism type of water hammer in the two-phase system is presented in Fig. 9. The condensate is not properly drained from the pipeline that delivers steam from the boiler towards the consumer.

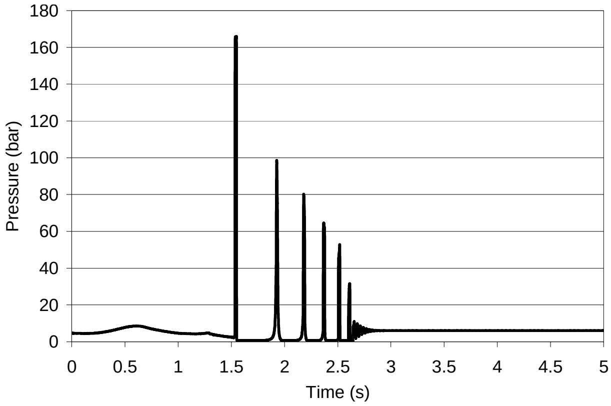

Figure 8. Calculated pressure peaks during the condensation induced water hammer.
Slika 8. Sračunati pikovi pritiska pri vodenom udaru zbog kondenzacije

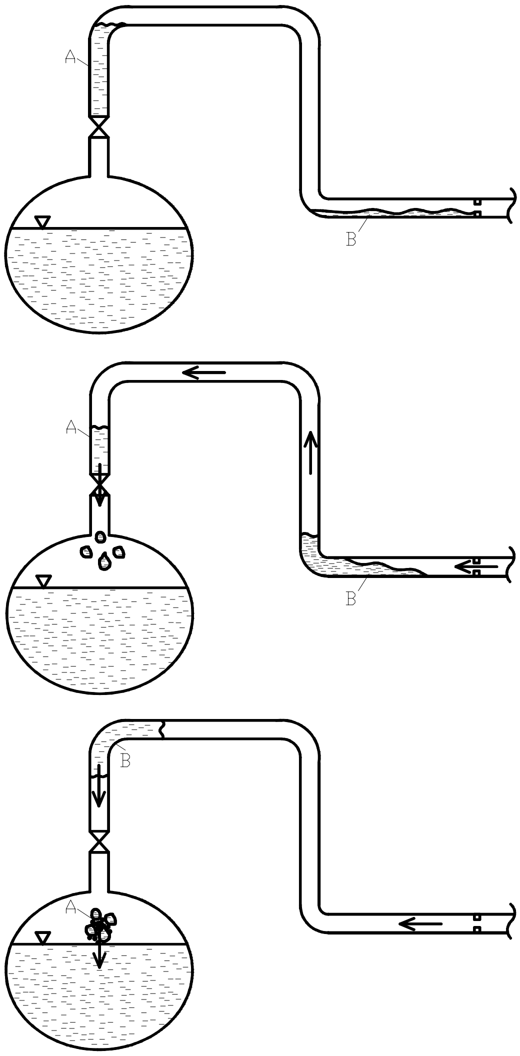

Figure 9. Formation of the slug of condensate and its impact onto the isolation valve at the steam boiler exit.
Slika 9. Obrazovanje čepa kondenzata i njegov uticaj na pregradni ventil na izlazu pare iz kotla

During the out-of-service period, the isolation valve above the steam dome at the boiler exit is closed. The condensate is accumulated above the isolation valve and in the horizontal segment near the elbow, as shown in Fig. 9. The condensate in the horizontal segment could not be drained due to the presence of the orifice that closed the substantial part of the pipe cross section. During the start-up period, opening of the isolation valve is initiated, the vibration of the steam pipeline was noticed for a period of ten of seconds and the isolation valve had fractured. The broken valve is shown in Fig. 10, /13/.

Figure 10. Isolation valve ruptured by the condensate slug impact.
Slika 10. Pregradni ventil slomljen zbog udara čepa kondenzata

The accident investigation revealed that prior to the main isolation valve opening, the steam pipeline was under the pressure slightly above the pressure in the boiler steam dome. Hence, the opening of the valve initiated the reverse steam flow towards the boiler, as depicted in Fig. 9. The condensate was moved by the steam flow towards the boiler, the slug was formed and it impacted onto the valve. Calculated flow parameters are shown in Figs. 11–13. The initial pressure in the steam line was 1.15 MPa, while the pressure in the boiler steam dome was lower for approximately 0.01 to 0.05 MPa. Calculated results have shown that the pressure peak at the moment of condensate slug impact onto the valve could reach an extremely high value above 20 MPa, as shown in Fig. 13. Obviously, this is the devastating event for the valve integrity. The design valve pressure was 1.6 MPa.

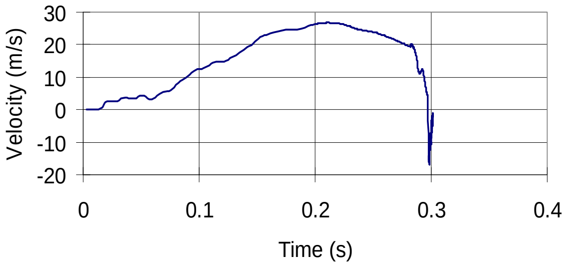

Figure 11. Steam velocity in reverse flow towards the steam boiler.
Slika 11. Brzina pare u povratnoj struji prema parnom kotlu

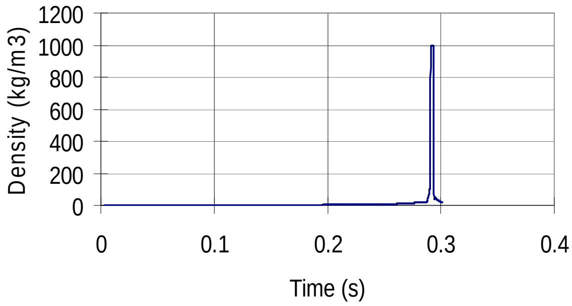

Figure 12. Fluid density at the main steam isolation valve.
Slika 12. Gustina fluida na pregradnom ventilu sveže pare

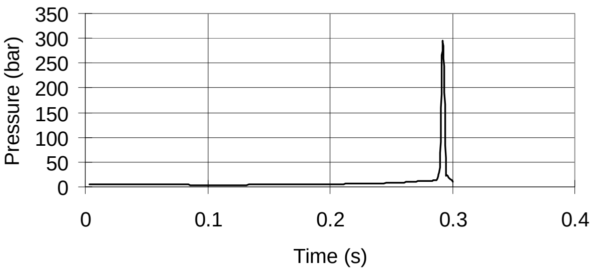

Figure 13. Pressure at the main steam isolation valve.
Slika 13. Pritisak na pregradnom ventilu sveže pare

CONCLUSION

Various mechanisms of one-phase and two-phase flows producing the water hammer effect and steam hammer phenomena are presented, such as steam hammer during the rapid closure of isolation valve in front of the steam turbine, condensation induced water hammer and impact of condensate slug. These transients and accidents are numerically simulated and analysed with the non-stationary compressible fluid flow model. Results show dynamic pressure and transient fluid forces generation during these events. Water hammer events in two-phase systems can cause much higher pressure peaks than in one-phase systems.

Laboratory experiments and plant accidents have confirmed that pressure peaks above hundred bars can occur due to condensation induced water hammer or water slug impact in a piping system that is initially at low pressure, close to atmospheric pressure.

Direct numerical simulation of these phenomena requires water-steam interface tracking and accurate prediction of interface transport processes. Numerical prediction of these phenomena is difficult due to numerical diffusion and consequent smearing of the liquid-vapour interface within the applied numerical mesh. In order to solve this problem, a higher order numerical scheme based on particle tracking and pressure wave propagation is developed for the solution of transport equations and interface tracking.

Obtained results show the importance of the investigation and prevention of the condensation induced water hammer and slug impact phenomena in the plants design phase and during plants operation, such as analyses of operational procedures and action of control and safety systems and proper drainage.

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