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Ways of Protection of Pipeline Systems against Hydraulic Hammer

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Abstract

A subject of the present article is the mechanism of hydraulic hammer emergence and main principles of functioning of devices that protect pipeline against its destructive effect. Special attention is paid to reviewing of ability of undissolved air containing in fluids to reduce the power of hydraulic hammer. According to the calculation results the volume concentration of undissolved air necessary for protection of pipelines against destruction from hydraulic hammer is not higher than 5%. In spite of the fact that this way of protection from hydraulic hammer is not versatile its successful usage for wastewater disposal pipeline is possible and optimal way of pipeline system aeration is represented by air inlet at the suction nozzle of the pump.

Keywords: physics of hydraulic hammer, hydraulic hammer alleviators, oscillation damping, air injection

1 Introduction

Among many problems which one have to face during operation of pipeline systems of different purpose one of the main problems is hydraulic hammering. This statement is provided first, by the fact that their occurrence is impossible to avoid by control of pipeline condition and quality of fluid flowing inside it because the hydraulic hammering is based on physical properties of the fluid and peculiarities of its transportation and, second, by their huge destructive power.

As an example of the branch that suffers significantly due to the reason mentioned above there may be housing and utility services in Russian Federation that operates an extensive network of long pipelines of heating and hot water supply. At all housing and utility systems (17 million km of in-house and 1 million km of main pipelines) annually there is more than 1 million of incidents, at that about 170 incidents fall within every 100 km of heat network and about 80 incidents – fall within every 100 km of domestic water supply pipelines and sewage systems. According to the statistics about 60% of these incidents occur due to the hydraulic hammer. At that costs for consequences management of each incident averages from 0.3 to 10 million rub (5.2-175 thousand USD), and combined costs from hot water leakages average 90 billion rub (1580 million USD) annually [12].

2 Mechanism of hydraulic hammer emergence

The mechanism of occurrence of the effect that is able to cause such an extensive damage is definitely worth separate reviewing. Hydraulic hammer is represented by a momentary (shock) increase or decrease of pressure in the pressure pipeline in which the fluid moves due to abrupt change of its speed in time. Processes emerging due to this are quite convenient to consider on the example of closing the valve located at the end of the horizontal straight-line cylindrical pipe with circular cross-section connected to the open tank.

Pipe length is known to equal L, average velocity of the stationary (before hydraulic hammer) fluid motion in the pipe is v_0, fluid density is ρ, excessive hydrodynamic height in the pipe before hydraulic hammer is $h_0 = p_0 / \rho g$ (Figure 1).

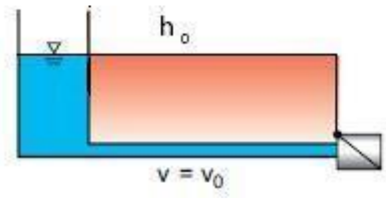
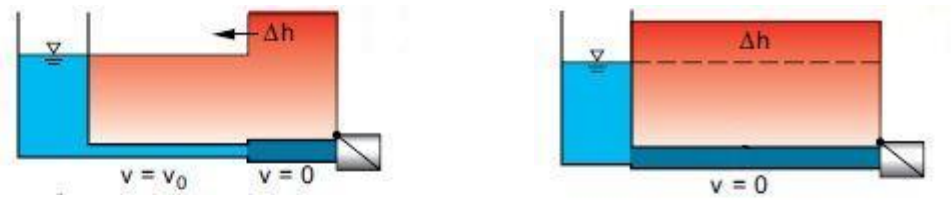


Fig. 1: System condition before hydraulic hammer

At the point of time t_0 the valve immediately closes. If the fluid was an absolutely incompressible ($E_f = \infty$), and pipeline walls were absolutely inelastic ($E = \infty$) then all the fluid mass would stop immediately, all its kinetic energy would be immediately converted into potential energy that would cause very big pressure increase. In real conditions the occurring pressure increases are lower as the fluid compressibility and elasticity of pipeline walls change the nature of the process.

For an infinitesimal time period Δt after immediate closing the valve the layer of fluid directly adjacent to the valve will stop while the upstream fluid layers continue moving in the direction of the valve with the original velocity v_0. Affected by these layers the stopped fluid mass compresses, pipeline walls expand, the pressure increases by the value Δp (hydrodynamic height – by Δh) (Figure 2a). The emerged wave of pressure increase moves along the pipeline in the reverse direction with the velocity equal to sound speed. The disturbance will reach the tank at the point of time $t = t_0 + L / v$ (Figure 2b). At this point the fluid in the pipeline is in the instantaneous quiescent state ($v = 0$) at the whole pipeline length L, the pressure increased and became equal $p = p_0 + \Delta p$.



a b

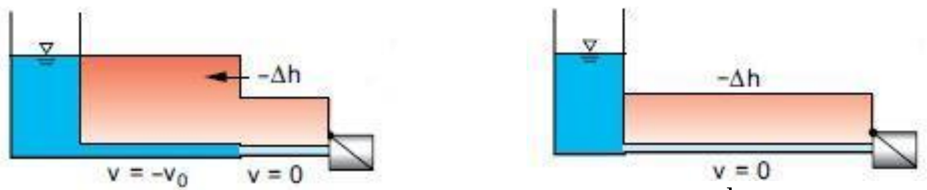
Fig. 2: System condition: a – at the time point $t_0 < t < L / v$;
b – at the time point $t = t_0 + L / v$

Let's assume that the tank volume is so big that level and, correspondingly, pressure of the fluid in the pipeline does not depend on processes in the tank and preserves the initial value equal to p_0. Due to this at the time point $t = t_0 + L / v$ there appear fluid motion from the pipe to the tank with the velocity in absolute value equal to the initial but in reverse direction, i.e. the fluid layer adjacent to the tank starts flowing out to the tank. At that as the disturbance spreads the pressure decreases to p_0, the pipeline converges (Figure 3a). The wave of pressure decrease spreads with the velocity v form the tank to the valve and it will reach it in the time point $t = t_0 + 2 L / v$. At that the pressure along the whole pipe length will drop to p_0 hydrodynamic height – to h_0, and its walls will reshape (Figure 3b).

a b

Fig. 3: System condition: a – at the time point $t_0 + L / v < t < t_0 + 2 L / v$;
b – at the time point $t = t_0 + 2 L / v$

In spite of the fact that at the time point $t = t_0 + 2 L / v$ the pressure in the pipeline equals to the pressure in the tank, the fluid is still in the motion state towards the tank. The moving mass inertia will lead to the fact that at the same time point the tank pressure will start decreasing below p_0. We assume that at this the pressure in the pipeline will remain higher than the saturation pressure at the given temperature, whereupon there is no aperture of discontinuity in the fluid and it cannot detach from the valve. Pressure decrease is followed by fluid stop and pipe walls deformation (Figure 4a), spreads from the valve to the tank with the velocity v, and reaches the tank in the last point of time $t = t_0 + 3 L / v$ (Figure 4b).



a b

Fig. 4: System condition: a – at the time point $t_0 + 2 L / v < t < t_0 + 3 L / v$;
b – at the time point $t = t_0 + 3 L / v$

As the pressure in the tank is constant and equals to p_0, now it exceeds the pressure in the pipeline, the quiescence occurred at the time point $t = t_0 + 3 L / c$ is unstable. Consequently, at the same time point the fluid starts moving with the velocity v_0 in the direction from the tank to the valve, i.e. there emerges the “reflected” from the tank wave of pressure increase spreading with the velocity v (Figure 5a). The time point $t = t_0 + 4 L / v$ it reaches the valve and fluid in the pipeline returns its initial condition characterized by the flow velocity v_0, directed to the valve and pressure p_0 (hydrodynamic height h_0) (Figure 5b). Due to this a new hydraulic hammer will occur, the pressure at the valve will increase by Δp and events will repeat in the consequence described above [1, 20].

a b

Fig. 5: System condition: a – at the time point $t_0 + 3\ L / v < t < t_0 + 4\ L / v$;
b – at the time point $t = t_0 + 4\ L / v$

Thus the hydraulic hammer is essentially a transient process occurring during system condition change and it has oscillating nature peculiar for such effects.

As a consequence the hydraulic hammer may have the following causes except for the above described:

1. Startup or shutdown of one or several pumps. At the moment of pump shutdown (e.g. due to power supply shortage) the fluid that the pipeline is filled with continues moving the initial direction because of momentum, and as its supply was stopped the pressure at the pump station drops. When the momentum effect is compensated then the fluid moved by the occurred pressure drop directs to the pump and meets the check valve that only can pass the flow in one direction – from the pump. In the result of abrupt deceleration of the fluid there occurs the hydraulic hammer starting with the pressure reduce wave.

2. Fast opening of the shutoff valve. During opening the valve the pressure drops abruptly so the hydraulic hammer occurrence is similar to the one caused by pump shutdown.

3. Air not discharged from the hydraulic system before filling it. If air located in the pipeline causes flow interruption then the pressure before the air bubble continues increasing, and consequently it will cause a collision of two parts of the flow – with higher and lower pressure – and occurrence of pressure increase wave.

4. Ebullition or condensing of the fluid in the pipeline. Specific volume of steam is higher than specific volume of fluid so occurrence of steam bubble causes a wave of pressure increase, its condensing causes pressure decrease wave. In the process of phase transition both effect may be observed simultaneously.

5. Unstable flow characteristics of the pump. Deviation from the safe operating area during pump operation may be the cause of occurrence of continuous self-oscillating of hydrodynamic nature. Particularly off-nominal flows occurring in some sections of pump wet end lead to hydraulic hammers [4, 5, 9, 10].

Obviously it does not seem possible to eliminate occurrence of such situations during operating pipelines, so instead of preventing the hydraulic hammer it is worth focusing on reducing its consequences. In connection with it for a long history of struggling with this problem there was developed a wide range of devices by some means reducing oscillating amplitude and thus protecting the pipeline against destruction.

3 Devices protecting against hydraulic hammer

The function of all existing nowadays hydraulic hammering protection devices (pressure stabilizers) is based on four main principles.

1. Drain of the fluid displaced by the pressure increase during hydraulic hammer. Essentially the device is represented by discharge valves that open at some increase of pressure in the pipeline. Hydraulic hammer alleviators of this type are usually quite widespread at modern pipeline systems due to their comparative simplicity and low cost, and also they don’t need additional maintenance. However their operation complicates design and operation of pipelines as it requires construction of additional vessels for drained fluid and losses makeup.

2. Pressure oscillations energy dissipation. The function of this type of devices is based on energy dissipation of the pressure increase wave due to friction. As a rule such stabilizers contain multiple perforation holes or hard-streamlined bodies. As a consequence they introduce big additional hydraulic resistance, that results in the fact that this principle is quite rarely used and thus it has a supplementary function.

3. Air injection and cramping. Devices of this type of function may be used only for relief of hydraulic hammers that start from pressure decrease wave. Such pressure relief valves are opened in case if the pressure in the pipeline is lower than the atmospheric one and they admit air that make up the created vacuum preventing the pipeline walls deformation.

4. Damping of pressure oscillations. Devices of this type contain a damping element which can be presented by some elastically deforming part (rubber insert, spring, bellows), or air-filled chamber. During occurrence of hydraulic hammer the pressure oscillation energy transfers to an energy of elastic element deformation and (or) air compression power [7, 13, 14].

Now such devices are considered to be most advanced. Figure 6a shows pressure stabilizer which utilizes compressed air and elastic inserts as a damping element. This pressure stabilizer that includes perforated section of a pipeline 4, external casing 1 and chamber 8. Elastic element 2, located between pipeline section and casing separates the chamber into two sealed pockets: gas and fluid. Fluid pocket is connected to the internal part of the pipeline through the perforation 5, and gas pocket 3 is connected to the gas source. Pressure oscillation damping occurs as a consequence of dissipation of energy at the perforation and energy absorption by the separating element and gas in the pocket 3.

To damp pressure oscillation in big diameter main pipelines there used pressure stabilizers with automobile tires as elastic elements (Figure 6b). The device contains central pipeline 1, containing perforated sections 2, 7 and blind part 4. Outside of the central pipeline 1 the sealed cylindrical casing 5 is fixed. Inside the casing 5 there are installed two perforated baffles 3 and 6, that separate it into three pockets: two edge pockets 8 and 12 and central one 9. In the central pocket 9 there is located elastic element made in the shape of set of used automobile tires 10 installed at the blind end 4 of the central pipeline 1 with radial clearance with respect to the side wall of the cylindrical casing 5. Automobile tires 10 are filled with foamed rubber. As a filler depending on physical properties of the fluid media in pipeline 1 – temperature and operational pressure – the following materials can also be used: cutting wastes of rubber or polyethylene hose, empty polyethylene bottles, laminated molded plastic, metal fiber, aluminum wool. Wide list of automobile tires allows using this type of stabilizers for different dimensions of pipelines [19].

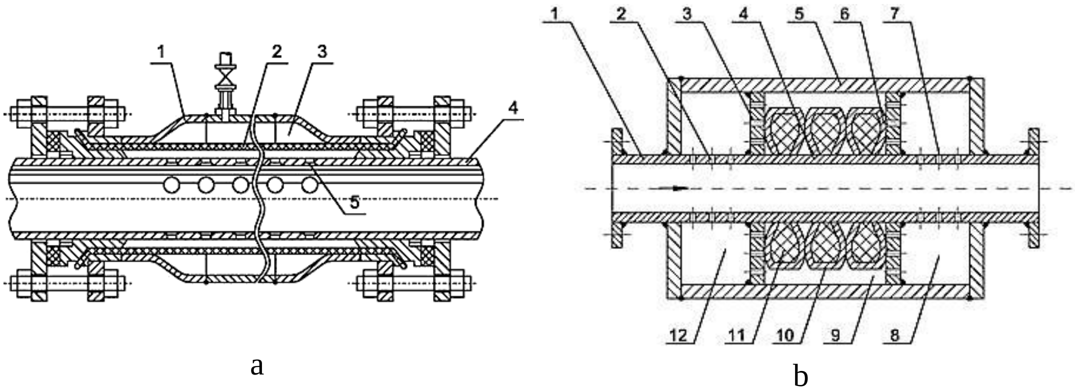


Fig. 6: Pressure stabilizers functioning under the principle of oscillations damping: a – with the compressed air and elastic inserts as a damping element; b – with automobile tires and foam rubber as a damping element

With the use of pressure stabilizers of this type it is possible to reach almost full damping of hydraulic hammer. An illustration to this statement may be an oscillogram in Figure 7, that shows fluid behavior characteristics during hydraulic hammer beginning with the pressure reducing wave without the stabilizer vs. after its installation. The pressure stabilizer used during this test included perforation holes and elastic insert made of polymeric material separating the casing into air and fluid pockets, i.e. had a structure similar to the one depicted in Figure 6a [12].

However, despite the high efficiency of operation these devices have a number of substantial disadvantages. First of all among these it worth emphasizing a heavy load which elastic elements are subjected to that results in higher risk of their rupture. Besides even with sufficient strength their elasticity is limited and it means that a situation when hydraulic hammer is not fully compensated is possible. As a result despite the number of successful designs the problem of decreasing the consequences of hydraulic hammers shall not be considered uniquely solved.

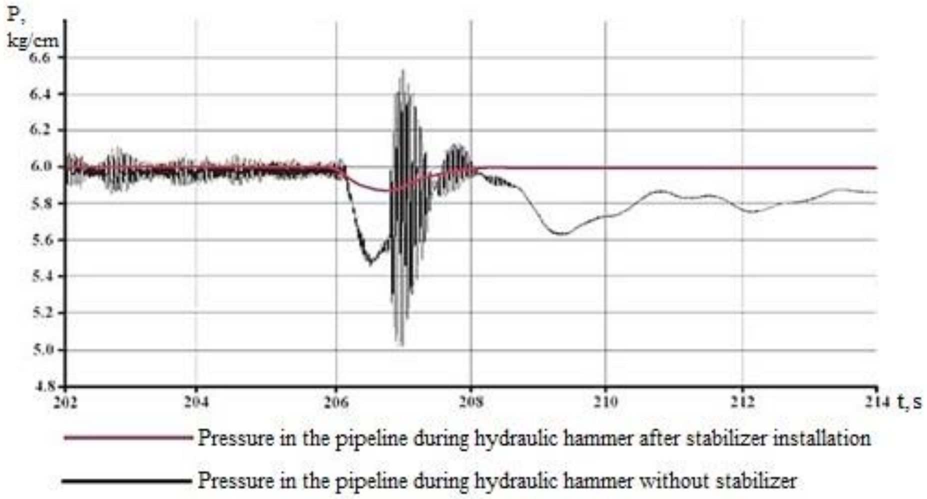


Fig. 7: Amplitude-frequency response characteristics of fluid before and after installation of the pressure stabilizer

4 Air injection to the pipeline as a way of hydraulic hammer relief

From the physical nature of hydraulic hammer there comes one more way to reduce pressure oscillations amplitude. Pressure increase value during the water hammer may be calculated by the formula:

$$\Delta p = \rho c v_0, \tag{1}$$

where ρ – is density of the fluid transported in the pipeline;
c – fluid disturbance spreading velocity (sound speed);
v_0 – velocity of fluid stationery flow.

Obviously hydraulic hammer power should be reduced by decrease of each of factors. Change of velocity v_0 may be considered unacceptable as it is regulated by peculiarities of the technologic process. Consequently it only remains to affect on the value of pressure increase during hydraulic hammer by changing parameters ρ and c. An obvious way of their reducing is air injection to the pipeline decreasing simultaneously both fluid density and sound speed connected to it.

If there is no additives in the fluid then the sound speed is calculated by the formula [16, 18, 20]:

$$c = \sqrt{\frac{E_f/\rho}{1+\frac{D}{e}\frac{E_f}{E}}}, \tag{2}$$

where E_f – fluid elasticity module;
ρ – fluid density;

D – internal diameter of the pipeline;
e – pipe walls thickness;
E – module of pipe material elasticity.

According to the formula (2) for steel pipeline with the internal diameter D = 100 mm and wall thickness e = 4 mm, in which a water is transported with the temperature 15°C with the operating pressure p_0 = 3 atm, sound speed will equal to 1273 m/s.

However in real conditions the sound speed appears to be much lower and makes about 500 m/s for the same conditions. The main reason of it is namely presence of particles and/or undissolved air.

At the presence of undissolved air in the fluid the sound speed is calculated by the formula [11, 17, 21]:

$$c = \frac{1}{\sqrt{(1-\varepsilon_g)\rho_f\left(K+\frac{1}{E_f}\right)+\varepsilon_g\rho_f\left(\frac{1-R}{E_g}-\frac{R}{\Delta p}\right)+\varepsilon_g[\rho_g\varepsilon_g+\rho_f(1-\varepsilon_g)]\left(K+\frac{R}{\Delta p}\right)}}, \tag{3}$$

$$K = \frac{D}{eE_{pipe}},\ R = 1-\left(\frac{p}{p+\Delta p}\right)^{\frac{1}{\chi}}$$

where ε_g – bulk concentration of air expressed in unit fraction;
ρ_f – density of the fluid (water) filling the pipeline;
E_{pipe} = 200 GPa – modulus of elasticity of pipe material (low-carbon steel);
E_f = 2030 MPa – water modulus of elasticity;
χ = 1.42 – adiabatic exponent for air;
$E_g = \chi p$ – air modulus of elasticity;
P_g – air density at the pressure p;
$\Delta p = \rho v_0 c$ – increase of pressure during hydraulic hammer.

As in case with the two-phase flow $c = f(\Delta p)$, and $\Delta p = f(c)$, then the formula (3) is recursive, however it can easily be solved by the method of successive iterations. The sound speed in the flow at the conditions assumed above for different v_0 with respect to the concentration of combined air is depicted at the figure 8; the corresponding pressure increase during hydraulic hammer is depicted in Figure 9. During calculation of the last value it was considered that during air inlet to the pipeline the media density will decrease and will equal to:

$$\rho = [\rho_g\varepsilon_g + \rho_f(1-\varepsilon_g)]. \tag{4}$$

As it is seen on these figures even weak volume concentration of air in the fluid flow equal to 4% reduces the pressure increase during hydraulic hammer by 4÷5 times. To understand what air content is required in order to prevent pipeline damage is it necessary to move further from pipeline pressure to mechanical stress in its walls.

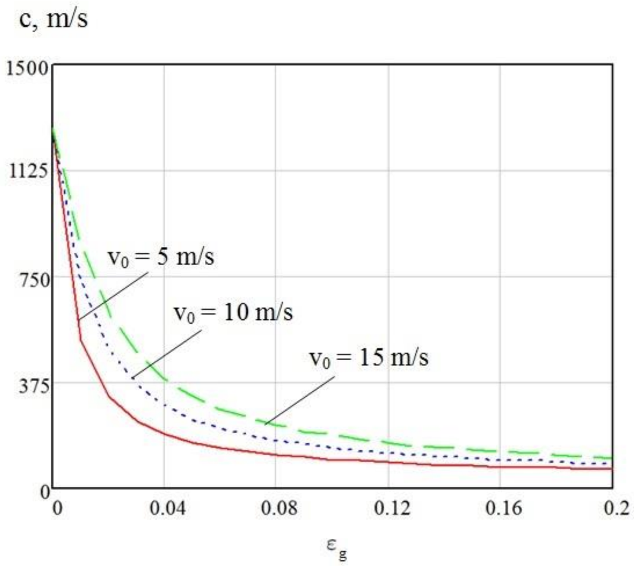


Fig. 8: Sound speeds in the two-phase flow with respect to the content of undissolved air

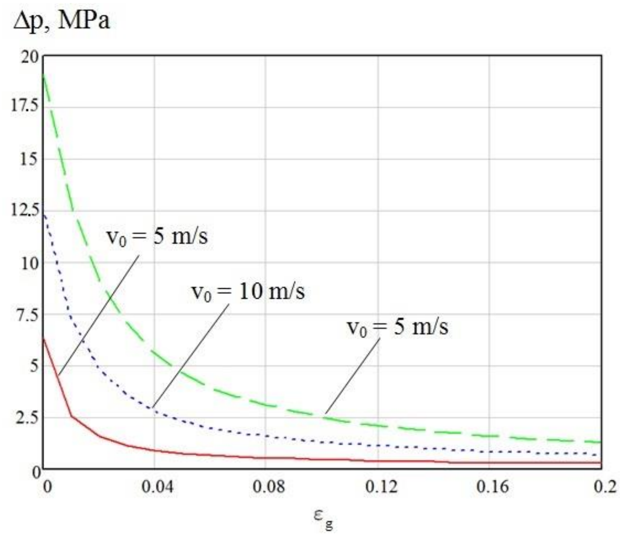


Fig. 9: Pressure increase value in the two-phase flow with respect to the content of undissolved air

As in the moment of hydraulic hammer the velocity of the fluid part with the highest pressure equals to 0 then the pipeline wall is affected only by static pressure equal $p_0 + \Delta p$. As in the considered case the pipe may be assumed a membrane shell then mechanical stresses are calculated by Laplace equation for a cylinder without including the membrane weight and liquid head.

Circumferential stress:

$$\sigma_t = D\frac{p_0+\Delta p}{2e}; \tag{5}$$

meridional stress:

$$\sigma_m = D\,\frac{p_0 + \Delta p}{4e}. \tag{6}$$

To evaluate the strength with respect to the material stress we use fourth (maximum-strain-energy) theory according to which:

$$\sigma_{eq} = \sqrt{(\sigma_t^2 - \sigma_m^2) + \sigma_m^2 + \sigma_t^2} \leq [\sigma] \tag{7}$$

For the selected low-carbon steel at the temperature 20°C the allowable stress $[\sigma]$ = 147 MPa.

The mechanical stresses calculated with the above described method are compared to the limit value in Figure 10. As it is depicted on the graph for the velocity v_0 equal to 10 m/s and 15 m/s and at the air volumetric content only 2.5% mechanical stresses during hydraulic hammer are close to the limit. At v_0 = 5 m/s the hydraulic hammer does not cause pipeline damage even in the one-phase flow, however even small quantity of combined air may significantly increase the strength margin. As we may see in the figure the optimal air content for different velocity values is varying approximately from 2% to 5% as it not only provides the pipeline preservation but also leaves some strength margin without increasing pipeline hydraulic resistance and without making difficulties during pumping.

The main limitation of this method of hydraulic hammer protection is the arrangement of air injection to the pipeline. For systems in which the centerline of the pump is higher than the fluid level in the pump sump the optimal way of aeration is air inlet under atmospheric pressure at the pump suction nozzle. Otherwise a compressed air supply is necessary.

During air inlet at the pump suction system aeration occurs along the whole length of the pipe. In connection with this the method is not applicable at pipelines of the compound section that require installation of air escape valves. In this case undissolved air will protect the pipeline against rupture only at the system section before the first air escape valve downstream where the air will be discharged.

Utilizing of this method is also complicated in case with high pressure pipelines as in this case to provide the required concentration of undissolved air a huge amount of air may be required as the air distinguishes itself with a good solvability.

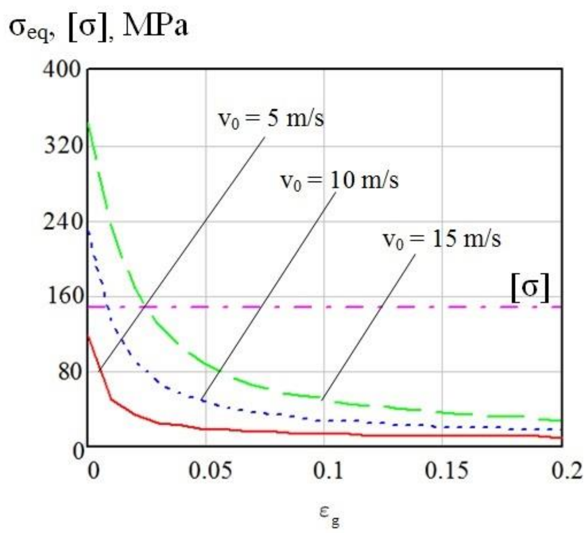


Fig. 10: Mechanical stress in the pipeline walls during hydraulic hammer with respect to the air volumetric concentration

Nevertheless air injection to the pipeline at the pump suction point may be used as a constantly functioning measure to prevent pressure drops in pressurized pipelines of water disposal. Also it worth considering that sewage water aeration will contribute to oxidation of organic pollutants [2, 8].

For pressurized wastewater disposal pipelines the recommended concentration of combined air makes 1-2%. In general, on the basis of the research findings this value does not exceed 5% that corresponds with the above described calculations. It was confirmed experimentally that this air concentration does not have a big negative impact to the pressurized pipeline functioning in a steady mode [3, 6, 15].

Thus inlet of small amount of air to the pipeline may be considered as an effective way of protection of a certain class of pipeline systems against rupture at hydraulic hammer, distinctive with low cost and high reliability at small requirements for additional maintenance.

5 Conclusion

Within the framework of the present research a mechanism of hydraulic hammer emergence was reviewed, thus it does not seem possible to avoid it within the technologic process.

Main principles of devices protecting against hydraulic hammer were reviewed, their advantages and disadvantages emphasized. Now the most prospective devices are ones with damping elements of different construction.

Air injection in the pipeline was reviewed as a way of pipeline protection against rupture during hydraulic hammer. Optimal volumetric concentration of air does not exceed 5%. This way of protection is best for pressurized pipelines of wastewater disposal system. The most rational way of aeration is arrangement of air inlet at the pump suction nozzle.

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