



Experimental Approach to Water Hammer Phenomenon

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Abstract. The paper indicates the frequent occurrence of transient states in hydraulic systems. Particular attention was paid to the phenomenon of water hammer - the causes and effects of this phenomenon. The necessity to modify the theoretical description of this phenomenon was indicated. The work focuses on the presentation of the development of the structure of experimental stands to study the phenomenon of water hammer. The research results obtained on the presented experimental stands were used to verify the theoretical considerations presented in other papers by the authors. The presented successive concepts of the test stand enable the research to be carried out to a greater extent and at the same time eliminate the disadvantages of the previous versions of the test stand. The stand presented as the final one also allows the testing of transients in hydraulic lines for various types of working fluid (oil, emulsion, distilled water).

Keywords: Hydraulics · Unsteady flow · Water hammer · Test stand · Experiment

1 Introduction

Unsteady states occur in hydraulic, water, heating, transmission and other systems more often than one might think. They are always accompanied by pressure pulsations, the peak values of which definitely exceed the set values, while the minimum ones may reach the level of the vapor pressure, thus the dangerous cavitation areas may occur. Therefore, both the maximum and minimum pressures may be responsible for the emerging system damage, among them the most dangerous are leaks related to damage to the walls of pipes or resulting from the unsealing of hydraulic connections, which result in significant financial losses, and in the case of oil spills, also a significant threat to the environment

[1]. Among the unsteady flows, there are accelerated, decelerated, reverse, pulsating flows and those which will be discussed in this paper, namely water hammer. Water hammer effect may be caused by the intentional closing of the valve, for example as a result of a power failure, or by deliberate control of the elements of hydraulic systems, including valves. During the impacts, there are intense changes in the basic parameters of the flow, i.e. pressure and velocity averaged in the cross-section. The intensity of the effect depends on the intensity of changes in these parameters. Quick-closing valves are often used in modern systems. It is their closure that often leads to the appearance of simple water hammers, characterized by the fact that closing the valve takes less time than the reflection (from reservoir) and return to the valve section of the original water hammer wave (increased pressure wave). In this case, the pressure increase may reach maximum values, which, when areas of vapour cavitation appear, could be almost twice as high as the pressures determined from formulas used in practice, e.g. the popular Joukowski formula.

Apart from the aforementioned water hammer, the hydraulic system often experience a periodic excitation. Such excitation comes, for example, from the flow pulsation of the displacement pump and is the result of the kinematics of the pump displacement elements [2, 3]. In a hydraulic system, due to the occurring impedance, pulsation of flow results in pressure pulsation [4]. The resulting pressure pulsation has frequencies corresponding to the flow pulsation [5, 6]. The real excitation signal in the hydrostatic system (in the form of pressure pulsation) is caused not only by the flow pulsation and can be approximated as a sinus series with different frequencies and amplitudes. Periodic pressure changes in the hydraulic system cause a number of negative effects on the components and the environment, including humans. The most important include:

- uneven operation of the hydraulic or pneumatic actuators,
- vibrations of the hydraulic system components, including hoses, which may lead to further critical damage (abrasion of hoses, fatigue load on the walls of the bodies of hydraulic components) [7],
- excitation of vibrations of hydraulic valve elements [8],
- increased noise of hydrostatic systems in a wide frequency range [5].

Transient states causing vibrations and noise in hydrostatic systems in relation to humans can cause, among others:

- disturbances in coordination of movements,
- increasing the time of visual or motor reaction [9],
- fatigue [10],
- changes in the nervous system [11],
- changes in peripheral vessels [12],
- changes in the osteoarticular system.

The harmful effect of vibroacoustic signals generated by transient states in the hydraulic system leads to a reduction in the quality of work performed and the formation of vibration disease in the human body. The resulting changes can be further

divided into acute and chronic [9]. Furthermore, one of the main requirements for modern hydraulically or pneumatically driven machines and equipment is precision [13]. Due to the occurrence of pressure pulsations of the working medium, the precision of hydraulic or pneumatic actuators can be significantly reduced.

For the above-mentioned reasons, the ability to model transients, especially the phenomenon of water hammer, is extremely important, in particular during the modernization of existing pressure systems as well as during the design of new ones. Due to the occurrence of reflected waves, the mathematical description of the water hammer phenomenon is based on the system of partial differential equations of the hyperbolic type [14]. For its solution and flow modeling, first of all, numerical programs are used, among which the dominant solution is based on the method of characteristics [15]. Analytical models are a minority and they have been derived for the simplest systems, e.g. for reservoir-pipe-valve system [16–18].

In order to better understand the water hammer phenomenon, experimental research is necessary in a wide spectrum of the dimensionless parameter defining it, namely the water hammer number $Wh = \frac{vL}{cR^2}$ [19]. These studies have been conducted at the Wrocław University of Science and Technology since 2007 [20] (by Zygmunt Kudźma and Michał Stosiak) with the participation of theoreticians from the then Szczecin University of Technology (Zbigniew Zarzycki and Sylwester Kudźma) and, in recent years, of the West Pomeranian University of Technology in Szczecin (Kamil Urbanowicz) [21, 22].

This paper will describe in detail the modification process of the hydraulic water hammer test stand built at the Wrocław University of Science and Technology. Examples of the obtained results will be presented and a discussion will be held on the further necessity of this type of research as well modification of test stand.

2 Water Hammer Effect Test Stand

In order to conduct experimental research and verify the modified mathematical models of the water hammer effect, a number of test stands were built in the laboratory of the Department of Technical Systems Operation and Maintenance, Wrocław University of Science and Technology. The first stand is presented in Fig. 1 [20]. This stand made it possible to achieve a full water hammer effect for lines with a minimum length of 13.1 m (assuming the pressure wave speed of 1,309 m/s and the switching time of the 4/2 directional control valve of 20 ms).

In the test stand, a steel pipe ($L = 18$ m, $d = 9$ mm), (section between points 7 and 9, Fig. 1) is supplied by a variable displacement pump (6) PTOZ2-K1-100R1 with manual adjustment. The pressure (constant and variable components) at the beginning and end of the tested pipe was measured with strain gauge pressure transducers 7 and 9 in Fig. 1. Due to the significant distance between the main pump (PTOZ2) and the oil tank, an auxiliary pump (1) was installed in the suction line of the pump (6) ensuring appropriate suction conditions. Pressure in the PTOZ2 pump suction line was monitored with a vacuum gauge (5). The adjustable throttle valve (11) was used to set the value of the loading pressure after a sudden switching of the 4/2 directional valve (10). The check valves (12) were used to eliminate the possible influence of the reflected pressure wave

between the directional control valve (10) and the oil tank. The cooler (14) stabilized the oil temperature with an accuracy of ± 1 °C. The test stand used L-HL68 hydraulic oil with a viscosity of 68 cSt at 40 °C. Safety valves 2 and 8 protected the relevant parts of the hydraulic system against overload. The flow meter (13) measured the value of the volumetric flow rate. Instantaneous pressure values at measurement points were recorded on a Tektronix TDS-224 digital oscilloscope and a computer with a dedicated WaveStar software.

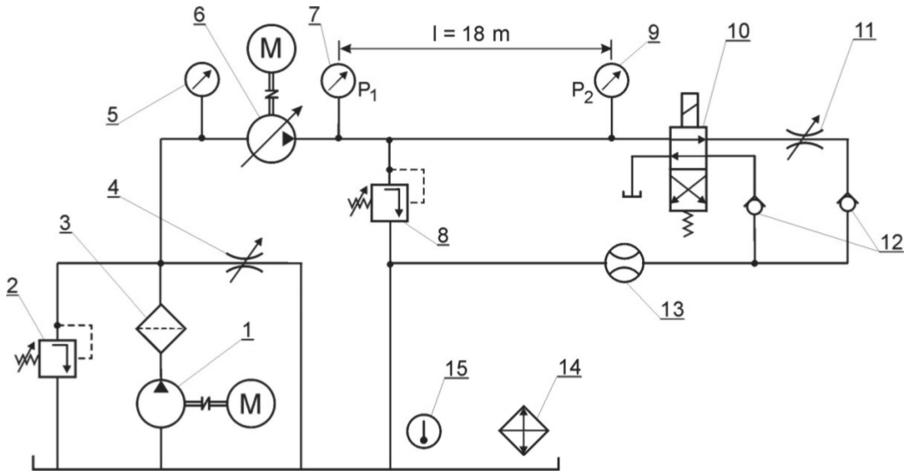


Fig. 1. Diagram of the stand for testing the water hammer effect in closed pressure lines: 1 – fixed displacement pump, 2 – safety valve, 3 – oil filter, 4 – adjustable throttle valve, 5 – pressure gauge, 6 – variable displacement pump, 7 – pressure sensor, 8 – safety valve, 9 – pressure sensor, 10 – directional control valve 4/2, 11 – adjustable throttle valve, 12 – check valve, 13 – flowmeter, 14 – fluid cooler, 15 thermometer, [20].

In the quasi-steady state (prior to transient), the pressure was recorded as a function of time, and then, using specialized software, the amplitude-frequency spectra were determined, as shown in Fig. 2.

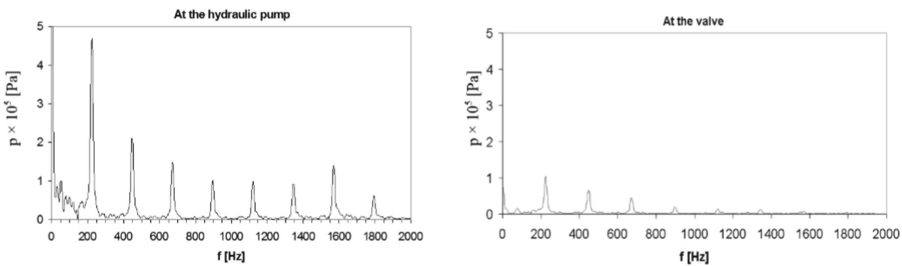


Fig. 2. Amplitude-frequency spectrum of pressure pulsation caused by pump flow pulsation ($\nu = 100$ cSt, mean pressure at the valve 1.2 MPa, mean pump flow rate 50 dm³/min) [20].

As can be seen in Fig. 2, in both spectra the dominant component is the fundamental component resulting from the pulsation of the flow of the pump, $f_1 = 225$ Hz. Successive components of the spectrum are its multiple.

These components will also be visible in time domain graph of pressure during the water hammer (lines 3 and 4). This is shown in Fig. 3 in comparison to the simulation runs (lines 1 and 2). The method of determining the simulation runs has been given in [8] and will not be discussed in detail in this paper. The simulation runs do not take into account the pressure pulsation resulting from the pulsation of the displacement pump flow.

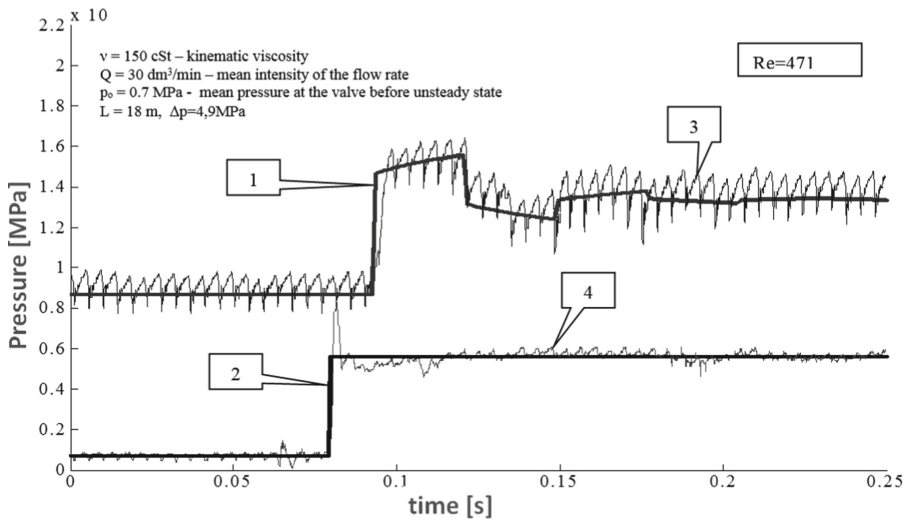


Fig. 3. Pressure run in water hammer effect and theoretical simulations [20].

The development of analytical considerations on the occurrence of water hammer in hydraulic lines of small diameters [22] forced the design of a new test stand. The constructed test stand made it possible to test relatively long and short hydraulic lines, and its limitation was the actuation time of shut-off valves. Ultimately, experimental tests were carried out for steel pipes with a length of 7 and 2.3 m and an internal diameter of 4 mm. At the ends of the tested pipes, strain gauge pressure sensors (13) and (14) AT-5230 were installed, both with a measuring range of 25 MPa and a measuring accuracy of ± 0.025 MPa. A quick shut-off valve (17) of poppet type was installed at the end section of the tested pipe, whose task was to suddenly stop the flow. The leakage rate on this valve is indicated by the manufacturer in drops per minute. The valve was controlled by a conventional electromagnetic coil. The actuation times of both valves (at the beginning and the end of the tested pipe) were selected to be identical and equal 10 ms. An accumulator (9) with a volume of 13 dm^3 was installed in the system, the role of which was to allow the reflection of the wave running in the direction from the valve (17) to the accumulator (9) after closing the valve (17) at the end of the line. A check valve (11) was installed on the accumulator supply line to ensure unidirectional flow.

The test system contained ISO32 mineral oil with a reduced viscosity ($1 \cdot 10^{-4} \text{ m}^2/\text{s}$ at 20°C). The tested pipe was supplied by an external gear micropump, driven by an electric motor with an inverter. This solution allowed for a smooth change of the flow rate in the tested micropipe. The scheme of the test stand is shown in Fig. 4.

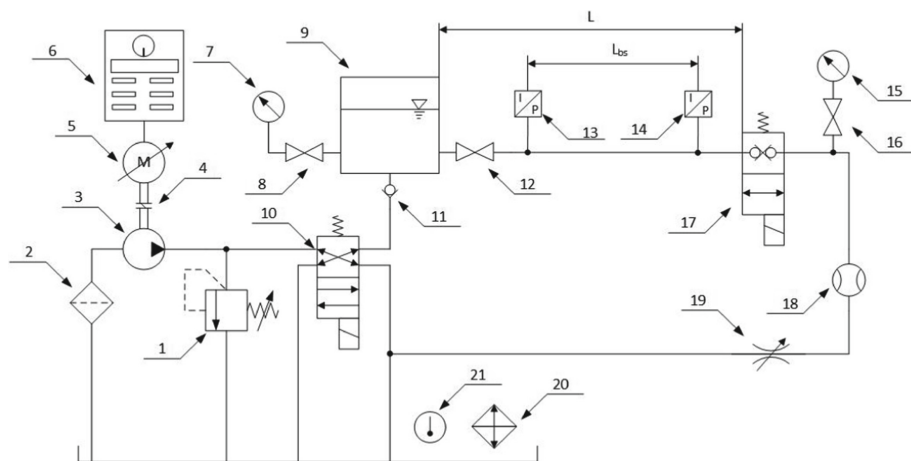


Fig. 4. Diagram of the hydraulic system of the test stand: 1 – pressure relief valve, 2 – suction filter, 3 – gear micropump, 4 – clutch, 5 – electric motor with adjustable speed, 6 – control cabinet, 7 – pressure gauge, 8 – shut-off valve, 9 – liquid tank (accumulator), 10 – electrically controlled directional control valve 4/2, 11 – check valve, 12 – shut-off valve, 13, 14 – strain gauge pressure transducers, 15 – manometer, 16 – shut-off valve, 17 – poppet shut-off valve, 18 – flow meter, 19 – adjustable throttle valve, 20 – oil cooler, 21 – thermometer.

Using the stand shown in Fig. 4, the experimental results of the pressure as a function of time at the end of the pipe (just before the shut-off valve 17) were obtained and used to validate the results of the numerical solutions. Figure 5 shows an example of the results obtained for the following conditions: oil temperature 21°C , average initial velocity of the liquid flow in the pipe 1.83 m/s , pressure wave speed $1,273 \text{ m/s}$. The simulation results were obtained for the conditions of sudden shut-off valve closure and unsteady friction losses.

The presented pressure results obtained from the experimental measurements (dashed line) has a pressure pulsation component originating from the pulsation of the displacement pump flow. However, the simulation solution does not take this into account. The elimination of pressure pulsation caused by pulsation of the displacement pump flow can be achieved by using a hydropneumatic accumulator - provided that the natural frequency of the accumulator corresponds to the pulsation frequency, i.e. it is usually in the range $175\text{--}350 \text{ Hz}$.

In further development of the water hammer effect test stand, an attempt was made to minimize the pressure pulsation from the pulsation of the displacement pump flow through changes in the supply system of the tested pipe. Unlike in the systems presented so far, the tested pipe was supplied by a hydraulic actuator acting as a single-piston displacement pump. This actuator was driven by a second actuator, which was powered

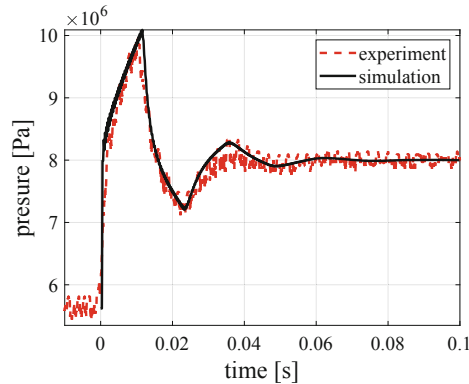


Fig. 5. Pressure as a function of time at the shut-off valve: $\nu = 100$ cSt, $Q = 1.38$ dm³/min.

by a hydraulic power unit with a constant displacement pump with an electric motor with adjustable rotational speed. Due to this solution, the impact of pulsation of the displacement pump flow on the tested pipe was limited under the condition of smooth operation of the supply actuator. Moreover, this solution increased the capacitance of the system on the supply side. The diagram of the hydraulic system of the modified test stand is shown in Fig. 6.

The tested copper pipe (length 30 m, internal diameter 4 mm) was wound on a drum with a diameter of 1.2 m and supplied by an actuator (11). The adjustable throttle valve (23) is responsible for generating the mean pressure for the set flow rate. After their actuation, the quick shut-off valves (15) and (22) are used to cause a water hammer effect in the pipe. Simultaneously with the actuation of the shut-off valves (15) and (22), the pump (3) is connected with the oil tank by switching the directional control valve (7) into the neutral position.

During the operation of the pump (3) to supply the actuator (10), its performance is stabilized by a two-way flow control valve (9). This valve also causes that the pump works under a constant load resulting from the setting of the pressure relief valve (1), which eliminates changes in its performance caused by the actual characteristics of the pump. Dynamic changes in pressure at the beginning and at the end of the tested pipe were recorded using pressure transducers (19) and (20) with a measuring range up to 40 MPa and accuracy of $\pm 0.15\%$ (BSFL), and the flow rate with a flow meter (24). The pressure transducers used were integrated with the fluid temperature meters. Thus, the liquid temperature was recorded at the beginning and end of a 30 m long cable. The modular structure of the test stand enables the use of different working fluids in two parts of the test stand. The pump with power supply has its own tank. On the other hand, the part of the test stand where the tested pipe is located has its own tank, separate from the power supply. Therefore, in the presented stand, various fluids can be used in the test part, even those that are not recommended by the manufacturer of the pump (3) supplying the driving cylinder (10).

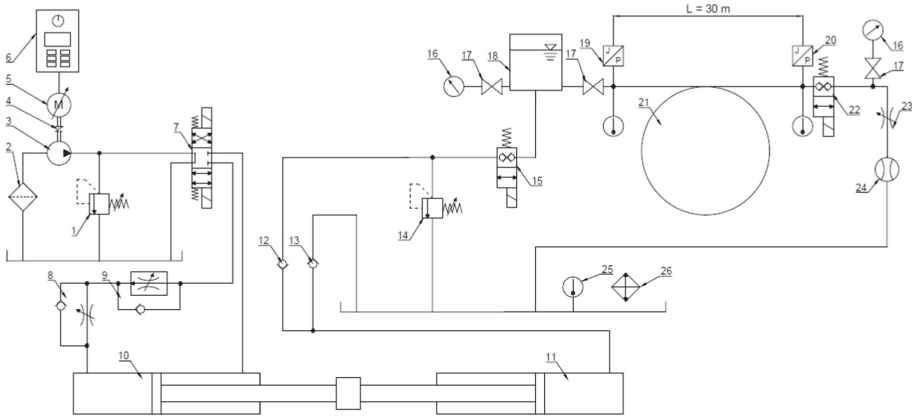


Fig. 6. Diagram of the system for water hammer effect testing for a 30 m long pipe: 1 – pressure relief valve, 2 – suction filter, 3 – pump, 4 – clutch, 5 – electric motor, 6 – control cabinet, 7 – 4/3 electrically controlled directional control valve, 8 – throttle-check valve, 9 – two-way flow control valve, 10 – driving actuator, 11 – driven actuator (source of flow in the tested line), 12, 13 – check valves, 14 – pressure relief valve, 15 – electrically operated cut-off valve, 16 – pressure gauge, 17 – shut-off valve, 18 – accumulator, 19, 20 – strain gauge pressure transducer, 21 – tested pipe, 22 – electrically controlled shut-off valve, 23 – adjustable throttle valve, 24 – flow meter, 25 – thermometer, 26 – oil cooler.

As can be seen from Fig. 7, the use of a modified supply of the tested pipe eliminated the pressure pulsation caused by the pulsation of the pump flow. As a result, all the obtained dynamic pressure changes are smoothed, and thus they are ideal for comparisons with the results of numerical tests.

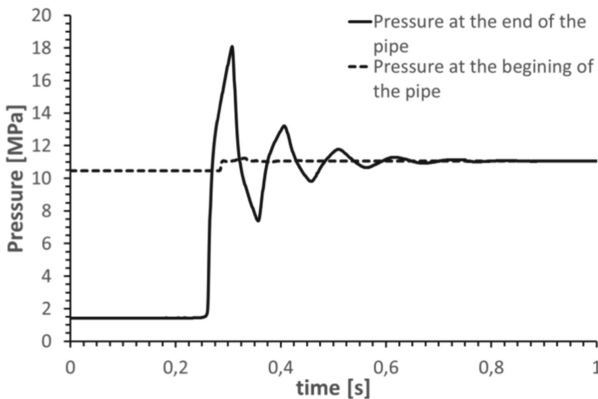


Fig. 7. Pressure as a function of time during water hammer in a 30 m long pipe: $\nu = 10 \text{ cSt}$, $Q = 7 \text{ dm}^3/\text{min}$.

3 Summary

The paper presents the development of the concept of test stands for researching the phenomenon of water hammer effect. All these stands have been built and the exemplary results of experimental studies of dynamic pressure change have been recorded. As demonstrated by experimental and numerical simulations presented in other studies, it is very important to assure an accurate stable initial flow velocity and dynamical stabilization of the working fluid temperature. In the latest version of our test stand shown in Fig. 6, the value of the initial flow rate was stabilized by a two-way flow control valve. In the same system, the two-way flow control valve cooperated with the pressure relief valve at the pump, thanks to which the pump worked at a set operating point, which ensured the stability of internal leaks (at a constant temperature of the working fluid). In Figs. 3, 5 and 7 we can observe the pressure time course recorded at the test stands. The pressure time course (at the valve, at the end of the tested pipe) shown in Fig. 7 is less damped than the pressure time course shown in Fig. 5: it has a larger maximum amplitude and the unsteady state lasts longer. This is due to the significant differences in viscosity of the working fluid. In the system in Fig. 6 the kinematic viscosity of the oil is 10 cSt, while in the system in Fig. 4 the kinematic viscosity of the oil is 100 cSt. The higher oil viscosity results in greater energy dispersion and greater internal friction damping.

As the further development of tests the medium of reduced viscosity (e.g. water) will be used as a working fluid and the flow rate will be increased so as to move into the strongly developed turbulent flow range with the possibility of cavitation during dynamic pressure changes. Moreover, the test stand will be extended with two hydropneumatic accumulators in which the gas pressure value will be controlled. A test pipe will be installed between the accumulators. This will make it possible to realise non-pulsatile flows and flows with cavitation. Also, the use of pneumatically or mechanically operated shut-off valves will shorten the valve actuation time, which will allow for the testing of pipes of shorter lengths.

In the future, it is also planned to test the flow in the plastic micropipes (with lower operating pressures). In this case, retarded strain occurs, which significantly affects the pressure changes.

The registered dynamic pressure changes will be used to verify the developed mathematical models of the water hammer effect. Correct models will be used to analyze newly designed and modernized pressure systems and will be used for follow-up (online - using algorithms based on artificial intelligence) protection of these systems against excessive, destructive pressure increases and leak detection. Accurate determination of maximum pressures during water hammer will allow determination of maximum loads to which the pipeline will be subjected during operation. In addition, reliable modelling of unsteady states in hydraulic systems will enable changes to be made to the hydraulic system at the design stage in order to reduce negative effects, such as vibroacoustic signals and critical damage to pipelines. As of today, the base of completed water hammer experiments is still limited. Most of the tests were performed for a medium with low viscosity, i.e. water. Hence, the implementation of new tests in a wide range of water hammer number Wh is so important.

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