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UNSTEADY FLUID FLOW IN PRESSURISED CLOSED PIPES IN EXPERIMENTAL BENCH EXAMPLES

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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. A complete analytical description of this phenomenon has not yet been developed. New theoretical models are still being developed and need to be verified experimentally. The paper focuses on presenting the development of experimental stands for the study of water hammer in hydraulic pipes. Subsequent modifications of the experimental stands for the speed of the shut-off valves and their tightness, as well as for the minimisation of the occurrence of the effects of the pulsation of the performance of the pump supplying the line under investigation, are presented. 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, airplane crash.

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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 (Leishear, 2020). These phenomena are particularly dangerous in the hydraulic systems of responsible machinery and equipment (Hunko et al., 2022). Hydraulic systems play a special role in aircraft (Zhang et al., 2014), where even the smallest malfunction in the system can lead to a serious threat to human life and the destruction of equipment of enormous value. On the other hand, the authors of the paper (Ułanowicz et al., 2020) proposed a method for estimating the service life of a hydraulic drive assembly based on its condition monitoring. Condition monitoring makes it possible to detect the pre-failure condition of a hydraulic assembly at the right time. Comprehensively collected causes of aircraft accidents and hydraulic system failures are included in the paper (Saleh et al., 2010). The author of the paper (Karpenko, 2022) describes the failure of an aircraft landing gear wheel hydraulic system resulting from a damaged hydraulic pipe. As an example, the Bombardier Challenger 605 aircraft is equipped with three independent hydraulic systems operating up to pressures of 21 MPa. These systems drive the primary and auxiliary flight controls, the aircraft's landing wheels, the wheel brakes and the front control wheel. To increase the reliability of these systems, each has two hydraulic pumps: a main pump and an emergency pump. The main pumps are usually driven from an engine (and called engine-driven pump, EDP) and the emergency pumps from an AC motor (called ACMP). A high-profile example of an aircraft hydraulic component failure was the case of Indonesian airline Sriwijaya Air's Boeing B737-200 flight SJ041 on 24 December 2011.

a)

b)

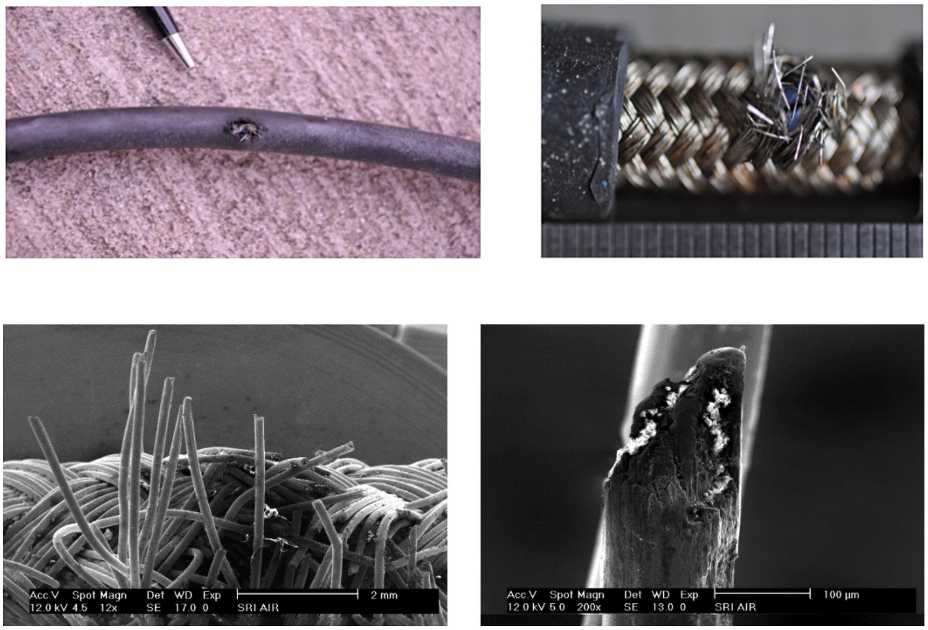

Figure 1. Damaged hydraulic pipe on a Boeing B737 aircraft (National Transport Safety Committee, 2014)

The flight was travelling from Sultan Syarif Kasim II Airport (Pekanbaru) to Soekarno Hatta Airport (Jakarta) with 134 people on board. The pilot first observed a drop in the hydraulic oil level in one of the systems at an altitude of 2,000 feet. As a result of a hydraulic system leak, the trailing flap was not raised. At an altitude of 5,000 feet, the pilot observed a complete loss of hydraulic oil. The pilot made the decision to divert and make an emergency landing at the airport as soon as possible. The landing took place with significant problems. During the landing, at the beginning of the aircraft's touchdown, the crew felt that there was a deceleration due to the brake being applied, then at a speed of about 60 kts (about 111 km/h) they felt no deceleration at all, despite increasing the pressure on the reverser. The pilot used an asymmetric reverser to stay aligned with the runway centreline. The main wheels of the aircraft came to a stop about 20 metres from the end of the runway. In this case, tragedy was narrowly missed. A meticulous visual inspection revealed that the cause of the failure was fatigue damage to the flexible hydraulic line as a result of its vibration, Figure 1. Hydraulic flexible pipes that are weakened by prolonged vibration are particularly susceptible to damage as a result of the water hammer that occurs in them.

Figures 1a and 1b in the macro and micro room respectively show damage to the outer protective layer and the steel braid carrying the internal loads.

One of the most recent examples of hydraulic failure in an aircraft is the case of a DHL Aero Expreso Boeing 757-200 performing a flight from San Jose (Costa Rica) to Guatemala City (Guatemala) on 07 April 2022. The aircraft was climbing off the runway and then the crew aborted the climb at approximately FL210 announcing "Mayday" due to a hydraulic failure. The aircraft then resumed its approach to the runway. After touchdown, the crew lost control of the aircraft, which rotated and skidded off the runway, causing the aircraft to break into two pieces (Figure 2).

These and many other examples demonstrate the continuing need to study, in particular, the unsteady states of hydraulic systems.

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

Figure 2. The crash of a Boeing 757-200 at San Jose Airport on 7.04.2022 (Hradecky, 2022)

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 (Stryczek, 2022; Karpenko & Bogdevičius, 2020). In a hydraulic system, due to the occurring impedance, pulsation of flow results in pressure pulsation (Bogdevičius & Lingaitis, 2005). The resulting pressure pulsation has frequencies corresponding to the flow pulsation (Kudźma, 2012; Karpenko & Bogdevičius, 2018). 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 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) (Karpenko, 2021);
- excitation of vibrations of hydraulic valve elements (Kudźma, 2012);
- increased noise of hydrostatic systems in a wide frequency range (Kudźma, 2012).
- 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 (Engel & Zawieska, 2010);
- fatigue (Tonin et al., 2016);
- changes in the nervous system (Gužas & Viršilas, 2009);
- changes in peripheral vessels (Chaban et al., 2021);
- changes in the osteoarticular system.

In addition to the effects mentioned above, the harmful effects of vibroacoustic signals generated by transients in the hydraulic system lead to a reduction in the quality of the work performed by humans and the development of vibration sickness in the human body (Engel & Zawieska, 2010).

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 (Wylie & Streeter, 1993). 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 (Urbanowicz et al., 2021a). Analytical models are a minority and they have been derived for the simplest systems, e.g. for reservoir-pipe-valve system (Urbanowicz et al., 2021a).

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 $Wh = \frac{\nu L}{cR^2}$, where: ν – kinematic viscosity of liquid, L – pipe lenght, c – wave speed, R – pipe inner radius (Urbanowicz et al., 2021a). These studies have been conducted at the Wrocław University of Science and Technology since 2007 (Zarzycki et al., 2007) (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) (Urbanowicz et al., 2021b).

This paper describes 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 are presented and a discussion is provided on the further necessity of this type of research as well modification of test stand.

1. Basic analytical description of fluid flow unsteady states in pipes

Research works on understanding and describing the phenomenon of water hammer have been in progress for more than 120 years. Many researchers have tackled the problem of providing an analytical description of the phenomenon that is consistent with observation and experience. Some of the best-known researchers working on the analytical description of hydraulic shock include: Zielke (Zielke, 1968), Rich (frictionless, quasi-steady cases) (Rich, 1945), Muto, Takahashi (Muto & Takahashi, 1985), Mei, Jing (unsteady cases) (Mei & Jing, 2016), Zarzycki (Zarzycki, 2000), Urbanowicz (Urbanowicz et al., 2021a), Adamkowski (Adamkowski et al., 2021).

The analytical description of water hammer in pipes is made by means of partial differential equations of the hyperbolic type describing the variation as a function of time t and pipe length x of the pressure parameters p and fluid flow velocity v:

$$\begin{cases} \dfrac{\partial p}{\partial t} + \rho c^2 \dfrac{\partial v}{\partial x} = 0 \\ \rho \dfrac{\partial v}{\partial t} + \dfrac{\partial p}{\partial x} + \dfrac{2}{R}\tau = 0 \end{cases}, \tag{1}$$

where: ρ – liquid density, c – speed of propagation of the pressure wave, R – pipe radius, τ – friction or instantaneous shear stress on the duct wall.

The instantaneous shear stress τ on the duct wall is the sum of the quasi-state quantity τ_{wq} and the time-varying quantity τ_{wn}:

$$\tau = \tau_{wq} + \tau_{wn}. \quad (2)$$

Analytical considerations usually adopt the following models to describe τ:

- frictionless model ($\tau = 0$);
- quasi-state friction losses model ($\tau = \tau_{wq}$);
- unsteady friction losses model ($\tau = \tau_{wq} + \tau_{wn}$).

The choice of friction loss model influences the numerical calculation results obtained and their accordance with experimental results. However, to date, a complete analytical model (in the time domain) with unsteady friction has not been developed.

The paper goes on to outline the development of experimental test rigs for the study of water hammer.

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 Figure 3 (Zarzycki et al., 2007). This stand made it possible to achieve a full water hammer effect for pipes with a minimum length of 13.1 m (assuming the pressure wave speed of 1309 m/s and the switching time of the 4/2 directional control valve of 20 ms).

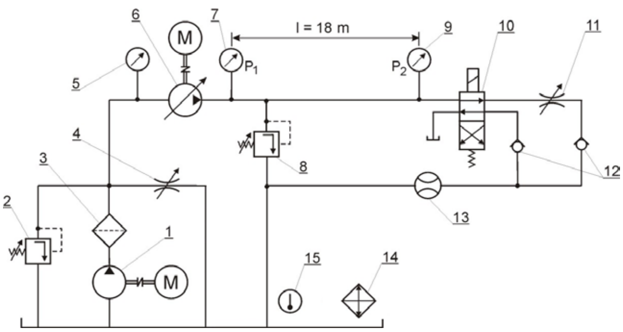

Figure 3. Diagram of the stand for testing the water hammer effect in closed pressure lines: 1 – fixed displacement pump, 2 – safety valve, 3 – oil filter, 4 – adjustable throttle valve, 5 – pressure gauge, 6 – variable displacement pump, 7 – pressure sensor, 8 – safety valve, 9 – pressure sensor, 10 – directional control valve 4/2, 11 – adjustable throttle valve, 12 – check valve, 13 – flowmeter, 14 – fluid cooler, 15 thermometer, (Zarzycki et al., 2007)

In the test stand, a steel pipe (L = 18 m, d = 9 mm), (section between points 7 and 9, Figure 3) 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 Figure 3. 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 pressure 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.

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 Figure 4.

As can be seen in Figure 4, 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 or other unsteady states. This stand can also be used to experimentally determine the speed of pressure wave propagation in a hydraulic pipe. Knowing the length of the pipe (distances between pressure sensors) and the time lag of, for

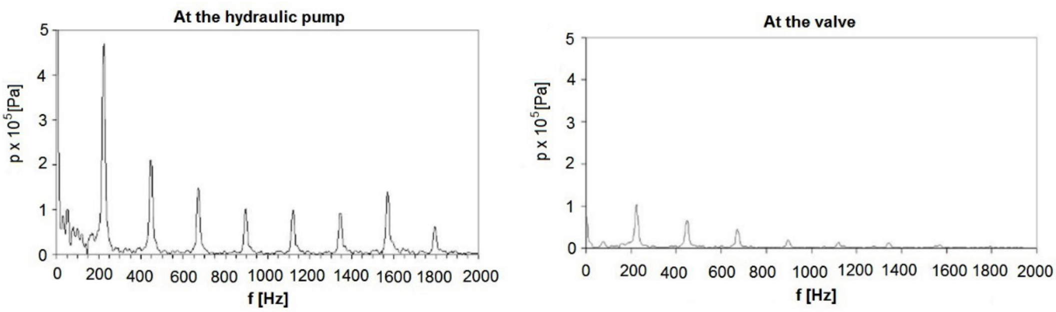

Figure 4. Amplitude-frequency spectrum of pressure pulsation caused by pump flow pulsation (ν = 100 cSt, mean pressure at the valve 1.2 MPa, mean pump flow rate 50 dm^3/min) (Zarzycki et al., 2007)

example, the onset of pressure build-up after a surge in the system load, the propagation velocity of the pressure wave in the tested pipe can be determined. The value of this velocity will be right for a test pipe filled with a particular oil at a particular temperature and for a fixed value of the dissolved air content and the static pressure in the test pipe. This is shown in Figure 5 for a rigid pipe (internal diameter 9 mm). Figure 6 shows the results for the flexible cable (9 mm internal diameter, one steel braid). The test conditions were the same as for the rigid pipe. Both figures clearly show the dynamic pressure increase caused by the step override of valve 10 in Figure 5 and the pressure pulsation caused by the flow pulsation of the displacement pump output. Test conditions: discharge pressure p_t = (0-20) MPa, pump suction pressure p_s = 0.01 MPa, pump shaft speed e.g. n_p = 25 1/s (1500 rpm).

For the test results presented here, the value of the pressure wave propagation velocity depending on the material of the hydraulic pipe can be determined. This velocity is: 1285.7 m/s and 818.2 m/s for rigid and flexible pipe respectively. However, a reliable determination of the pressure wave propagation velocity requires a series of tests under identical conditions.

The development of analytical considerations on the occurrence of water hammer in hydraulic pipes of small diameters (Urbanowicz et al., 2021b) forced the design of a new test stand. The constructed test stand made it possible to test relatively long and short hydraulic pipes, 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 pipe. 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 ($1{\cdot}10^{-4}$ m^2/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 Figure 7.

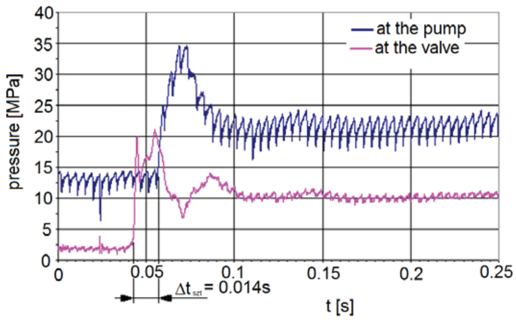

Figure 5. Pressure run in water hammer effect for rigid pipe, Δt_{szt} = 0,014 s (Zarzycki et al., 2007; Kudźma, 2012)

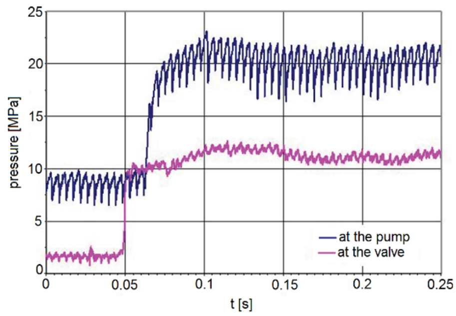

Figure 6. Pressure run in water hammer effect for flexible pipe, Δt_{el} = 0,022 s (Zarzycki et al., 2007; Kudźma, 2012)

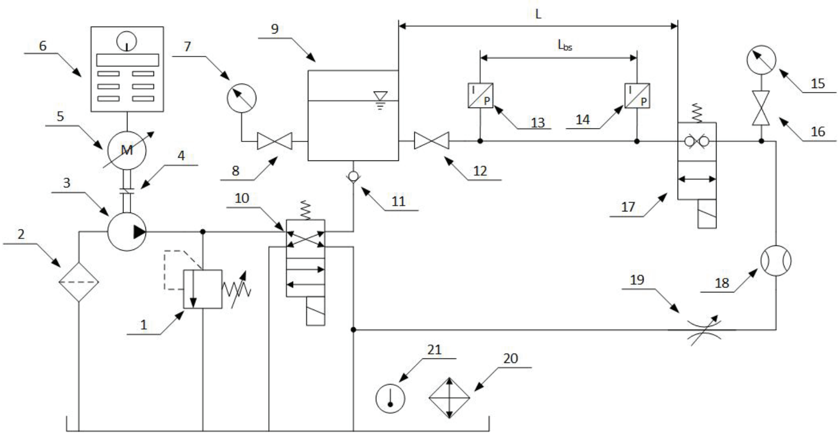

Figure 7. Diagram of the hydraulic system of the test stand: 1 – pressure relief valve, 2 – suction filter, 3 – gear micropump, 4 – clutch, 5 – electric motor with adjustable speed, 6 – control cabinet, 7 – pressure gauge, 8 – shut-off valve , 9 – liquid tank (accumulator), 10 – electrically controlled 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 Figure 7, 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 8 shows an example of the results obtained for the following conditions: oil temperature is 21° C, average flow rate of the liquid in the pipe is 1.15 dm^3/min, pressure wave speed is 1273 m/s.

The presented pressure results obtained from the experimental measurements has a pressure pulsation component originating from the pulsation of the displacement pump flow. However, mathematical models and the simulation solution do 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 175-350 Hz.

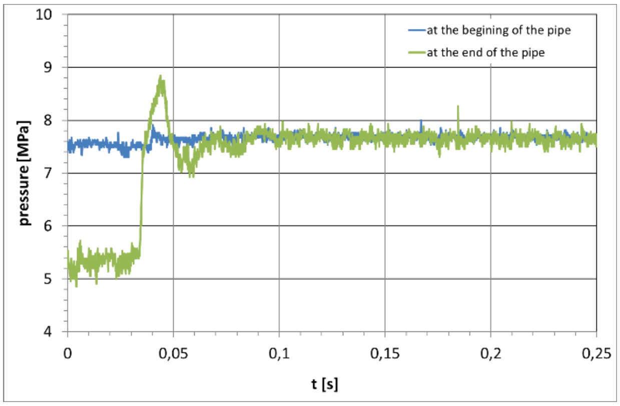

Figure 8. Pressure as a function of time inside tested pipe

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 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 Figure 9.

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

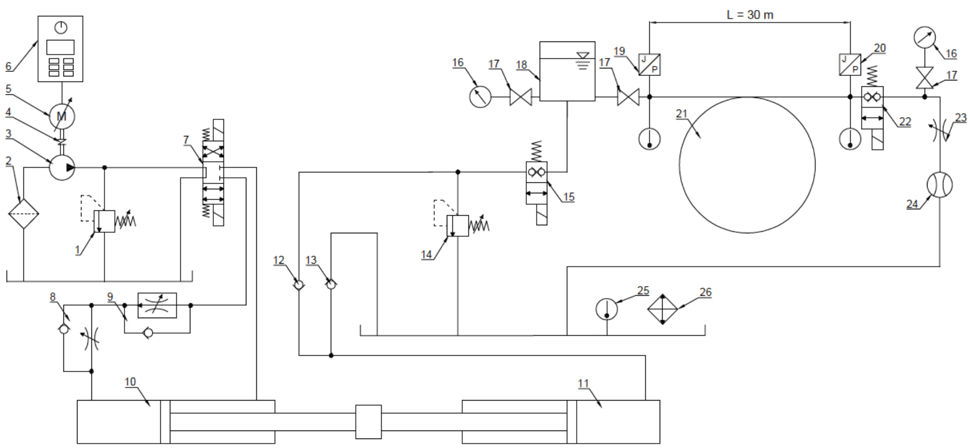

Figure 9. Diagram of the system for water hammer effect testing for a 30 m long pipe: 1 – pressure relief valve, 2 – suction filter, 3 – pump, 4 – clutch, 5 – electric motor, 6 – control cabinet, 7 – 4/3 electrically controlled directional control valve, 8 – throttle-check valve, 9 – two-way flow control 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

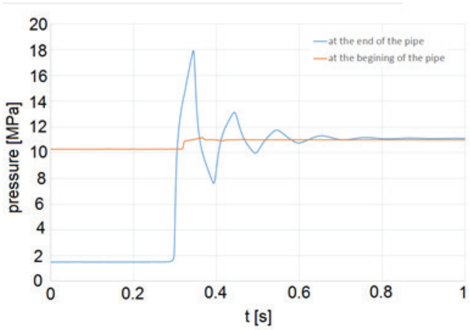

Figure 10. Pressure as a function of time during water hammer in a 30 m long pipe

(20) with a measuring range up to 40 MPa and accuracy of +/–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 pipe. The viscosity of the oil was $12 \cdot 10^{-6}$ m^2/s, flow rate 6.8 dm^3/min. 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).

As can be seen from Figure 10, 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. The pressure waveform in Figure 10 is less attenuated than in Figure 9, and this is due to the difference in viscosity of the working fluid. In the system whose diagram is shown in Figure 9, the viscosity of the fluid is about 10 times higher than in the system shown in Figure 10.

Conclusions

As demonstrated by examples of serious accidents it is necessary to describe the water hammer phenomenon accurately in order to predict it and to predict the maximum pressures occurring inside the pipe during this phenomenon.

The paper presents the development of the concept of test stands for researching the phenomenon of water hammer effect. All this stands have been built and the exemplary results of experimental studies of dynamic pressure change have been recorded. In the latest version of the test stand, 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).

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 affect 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. 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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