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на тему

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Waterhammer Control in the Delivery Conduit of a Pump

Reported in the paper is an experimentally checked algorithm allowing to control the waterhammer development by operating the valve installed in the pumping system. The possibility is indicated of application of the algorithm mentioned for control of operation of hydraulic machines in systems with long pipelines in transient conditions.

1. Introduction

The waterhammer in closed conduits occurs due to sudden changes of the liquid flow conditions. In the pumping systems this effect appears mainly during the pump starting and stopping or control valves opening and closing.

The waterhammer entails the increase of the dynamic load of the pump and flow guiding elements mated with it. These effects lead to deterioration of the durability and reliability of components of the whole pumping system. Most frequently the waterhammer appears in systems composed of long pressure conduits in which fast slowing-down of the flowing liquid occurs. The particularly dangerous water-hammer can occur in pumped-storage power stations equipped with pump-turbines. These dangerous conditions are enhanced by long penstocks, short duration of the transient processes and wide range of changes of the machine operating parameters.

In a given pump or turbine system the waterhammer depends mainly on the control of transient states, i.e. on the changes of the machine rotor speed and on the operation mode of its control devices (valves, guide-rings or rotor blades).

Figures 1 and 2 show the results illustrating the influence of the valve closure mode on the pressure changes in the pump delivery conduit. Figure 1 refers to the case in which the pump driving motor is stopped and then the flow in the pump discharge nozzle is throttled by means of the valve. Figure 2 concerns a case in which the water flow in the turbining mode of the pump operation is being stopped by the valve. The results reported show not only that the valve closure mode has a great influence on the waterhammer but also indicate the need of proper operation of the control devices to ensure the waterhammer reduction.

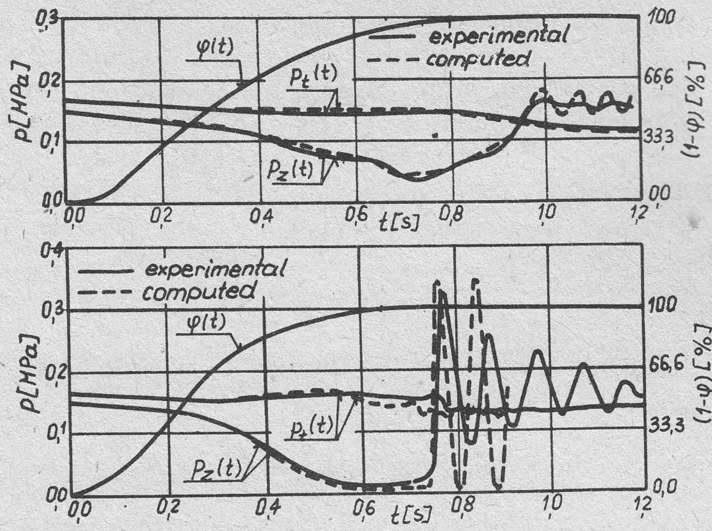

Fig. 1. Pressure changes in the pump discharge nozzle $p_t(t)$ and at the valve outlet $p_z(t)$ with the pump driving motor stopped and for different valve closing modes $\varphi(t)$

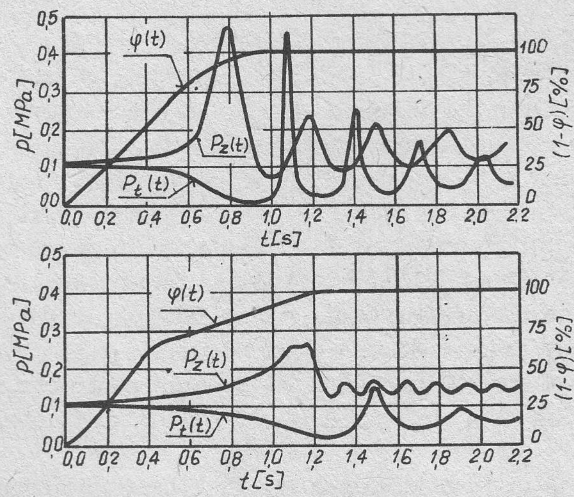

Fig. 2. Pressure changes in the pump discharge nozzle $p_t(t)$ and at the valve outlet $p_z(t)$ for different ways of the valve closing $\varphi(t)$ under turbining flow conditions

The comparison of the measured and calculated pressure variations (Fig. 1) shows a sufficiently good agreement between the theoretical and experimental data. Therefore, the purpose of developing numerical codes for waterhammer control is justified. Waterhammer control consists in forcing liquid flow in a prescribed manner. Most frequently keeping the pressure rise caused by waterhammer within the assumed limits and ensuring fast stabilization of transient processes is required.

The aim of this work was to develop and check experimentally a computer program for the valve control to ensure the proper pressure variations in the pump delivery conduit and to prove this program usability for the waterhammer reduction and shortening the duration of pressure stabilization.

The work reported in the paper was prepared at the Institute of Fluid-Flow Machinery of the Polish Academy of Sciences in Gdańsk within the frames of the Government-Sponsored Research-Development Program "Complex Development of Power Industry" PR-8.

2. Theoretical Basis of the Control Algorithm

The control of the waterhammer can be regarded as an inverse problem to the analysis of liquid transient flows in the pipelines. The changes in the pressure and flow velocity caused by a known perturbation such as for example a defined motion of the flow throttling device or a change of the pump rotational speed are predicted when a given system is analyzed. The problems connected with the waterhammer

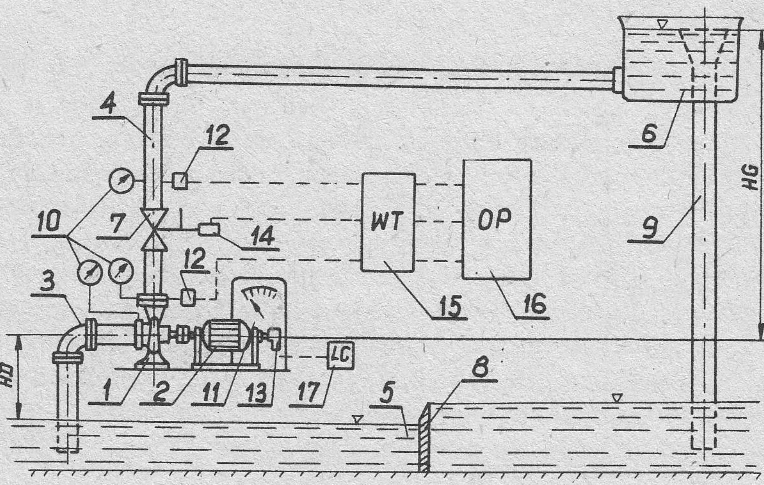

Fig. 3. Experimental stand scheme

1 — centrifugal pump 65PJM160, *2* — electrodynamometer KS26-4, *3* — suction tube (length 7.6 m, inner diameter 70 mm), *4* — delivery conduit (length 20.3 m, inner diameter 70 mm), *5* — suction reservoir, *6* — delivery reservoir, *7* — butterfly valve with a manual knob, *8* — Poncelet overflow, *9* — overflow tube, *10* — manometr, *11* — scales for measuring the turning moment, *12* — pressure transducer, *13* — rotational speed converter, *14* — converter of the butterfly revolution angle, *15* — amplifier, *16* — loop oscillograph, *17* — digital counter

control relate to determination of the perturbation in order to obtain the required change of the pressure and flow velocity. In this work the requirements are formulated for the waterhammer development at a selected point of the pumping system. The appropriate algorithm has been worked-out basing on the equations describing a transient flow in the pipelines and on boundary conditions representing the properties of the pumping system under consideration (Fig. 3).

2.1. Transient Flow of Liquid in Closed Conduits

The unsteady flow of liquids in the closed conduits is described by the flow continuity and motion equations formulated under following assumptions:

1. The flow is onedimensional.
2. The conduit walls and liquid are subject to elastic deformation.
3. The hydraulic losses are proportional to the square of the flow instantaneous velocity.
4. The flow velocity in the conduit can be regarded small when compared to the propagation speed of perturbations in the flow.

Under above assumptions the equations of motion and continuity take the following form [1]:

$$g\frac{\partial H}{\partial x}+\frac{\partial v}{\partial t}+\frac{\lambda}{2D}\,v|v|=0, \tag{1}$$

$$\frac{\partial H}{\partial t}+\frac{a^2}{g}\frac{\partial v}{\partial x}=0, \tag{2}$$

where: x denotes the distance measuring coordinate, t is the time, H describes the static pressure head, v stands for the liquid flow velocity, $D=\sqrt{4F/\pi}$ is the conduit substitute diameter, F denotes the area of the inner cross-section of the conduit, g is the acceleration of gravity force and a stands for the pressure wave speed in the conduit.

Equations (1) and (2) form a closed system of hyperbolic partial differential equations in which $H(x,t)$ and $v(x,t)$ are the functions sought for. This system of equations can be solved numerically under arbitrary boundary conditions. Prevalently these equations are integrated by the method of characteristics. Recurring to this method Eqs (1) and (2) can be transformed into the following system of ordinary differential equations:

$$\frac{g}{a}\frac{dH}{dt}+\frac{dv}{dt}+\frac{\lambda v|v|}{2D}=0 \tag{3}$$

for

$$\frac{dx}{dt}=+a; \tag{4}$$

$$-\frac{g}{a}\frac{dH}{dt}+\frac{dv}{dt}+\frac{\lambda v|v|}{2D}=0 \tag{5}$$

for

$$\frac{dx}{dt}=-a. \tag{6}$$

Equations (4) and (6) define a set of curves called characteristics in the $x-t$ plane. The characteristics can be interpreted physically as the lines along which the flow perturbations are propagating. Equations (3) and (5) determine relationships between the liquid flow parameters which are fulfilled along above mentioned lines. Taking into consideration the boundary conditions, Eqs (3) ÷ (6) can be solved numerically using the difference method of the first or higher order.

2.2. Boundary Conditions

Initial conditions. Determined by the steady flow parameters.

Suction and delivery reservoir. It is assumed that during the transient flow the levels of liquid in the suction and delivery reservoirs remain constant.

Valve. During the displacement of the valve spindle the flow rate and the pressure difference between the valve inlet and outlet are changing. The changes of these parameters force a transient liquid flow within the whole pumping system. The difference ΔH of pressure head at the valve inlet and outlet under transient flow conditions is assumed to change according to the following formula:

$$\Delta H(t) = \frac{1}{2g} \frac{v(t)\,|v(t)|}{[C_p(\varphi(t))]^2}. \tag{7}$$

The function $C_p(\varphi(t))$ depends on the valve throttling capacity and on the drive of valve closing device (in case under consideration on the valve butterfly drive). The throttling properties have been determined giving the relationship between the flow factor C_p and the valve opening φ (the angle of the valve butterfly position). This last relationship has been determined basing on the tests under steady flow conditions (Fig. 4). The valve drive evolution is given by the function $\varphi(t)$. The control of the waterhammer exiges that this function is previously determined.

Pump. The changes of the pump operation conditions occurring during the transient flow have been described by the following equations:

a) equation of the rotating masses motion for the pump-electric motor system in the form

$$\frac{\pi I}{30} \frac{dn}{dt} = M_e(n) - M(Q, n). \tag{8}$$

In Eq. (8) I stands for the rotating masses moment of inertia, n for the rotational speed, Q denotes the flow rate, M is the moment of hydraulic forces, and M_e the moment of electric motor electrodynamic forces;

b) equation of energy balance of the pump in the form

$$H_t - H_s + \frac{Q^2}{2g}\left(\frac{1}{F_t^2} - \frac{1}{F_s^2}\right) = H_e(Q, n). \tag{9}$$

In Eq. (9) H_t and H_s stand for the pressure head in the pump discharge and suction nozzle, respectively; H_e is the pump total head, F_s and F_t denote the areas of the suction and discharge nozzle, respectively.

The functions $M(Q, n)$ and $H_e(Q, n)$ have been determined basing on the experimental characteristics of the pump obtained for the steady flow conditions (Fig. 5). The function $M_e(n)$ has been recovered out from the experimental static characteristics of the motor.

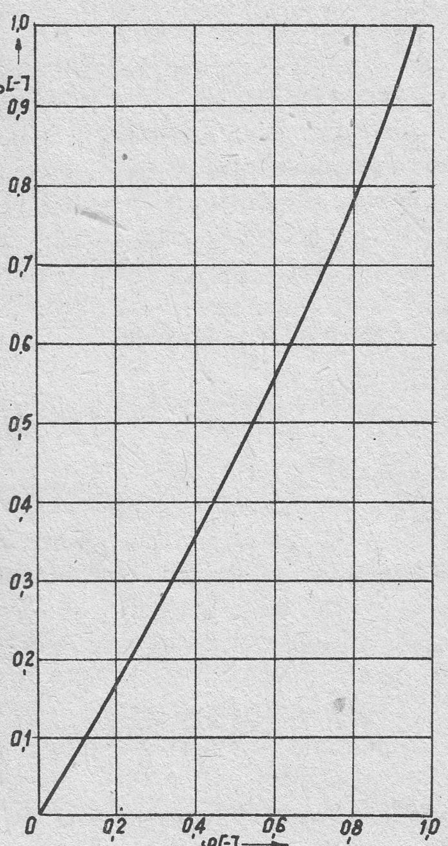

Fig. 4. Relationship between the valve flow factor, C_p, and its opening φ

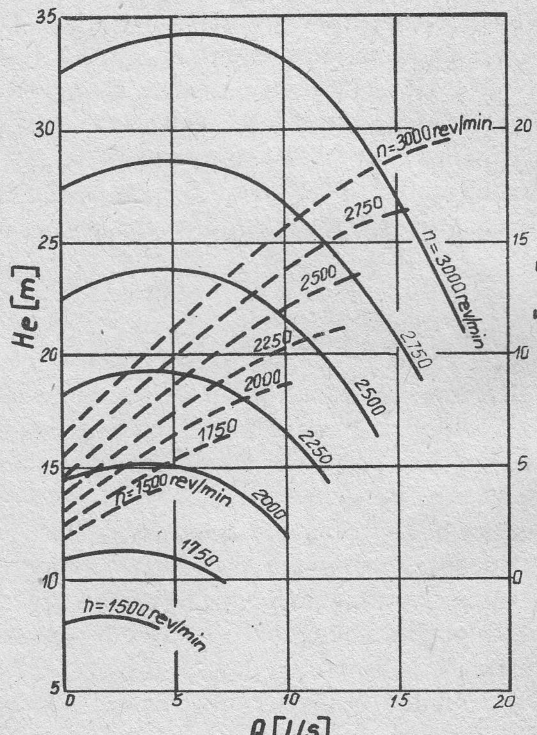

Fig. 5. 65PJM160 pump characteristics
———— head-discharge curves $H_e(Q)$,
– – – – moment-discharge curves $M(Q)$

The chosen description of the pump transient state omits the forces generated by the liquid flow within the pump impeller and the electrodynamic forces due to the fast alternating inductive currents generated via the changes of the pump driving electric motor parameters. The errors resulting from neglection of the transient flow in the pump are not very evident in cases in which the mass of liquid contained in the pump is small compared to that contained in the conduits and when the pressure pulsations generated by the pump blading are small compared to the waterhammer effects. Omission of the transient conditions of the electric motor operation is based on the fact that electromagnetic time constants are very small when compared with the time constants characterizing mechanical transient states.

3. Algorithm and Control Program

Equations (3) ÷ (6) together with the boundary conditions constituted the basis for the working out of an algorithm and numerical program for control of the waterhammer course by means of a valve installed within the pumping system.

Numerical integration of Eqs (3) ÷ (6) consists in determining the grid of characteristics in the $x-t$ plane basing on Eqs (4) and (6) and in calculating the discrete values of v and H in nodes of this grid based on Eqs (3) and (5) and taking into account the boundary conditions. Fig. 6 shows the grid of characteristics in the $x-t$ plane limited by the suction and delivery pipes. This grid was obtained by dividing the suction and pressure pipes into sections keeping in mind that the time Δt of the pressure wave propagation along each of these sections should be identical.

To carry out the numerical calculations Eqs (3) ÷ (6) have been replaced by the following system of difference equations [5]:

$$H_{i,j}-H_{i,j+1}+E_j(Q_{i,j}-Q_{i,j+1})+R_j(Q^2_{i,j}+Q_{i,j}Q_{i,j+1}+Q^2_{i,j+1})=0 \tag{10}$$

for

$$x_j-x_{j+1}=a_j\frac{\Delta t}{2}; \tag{11}$$

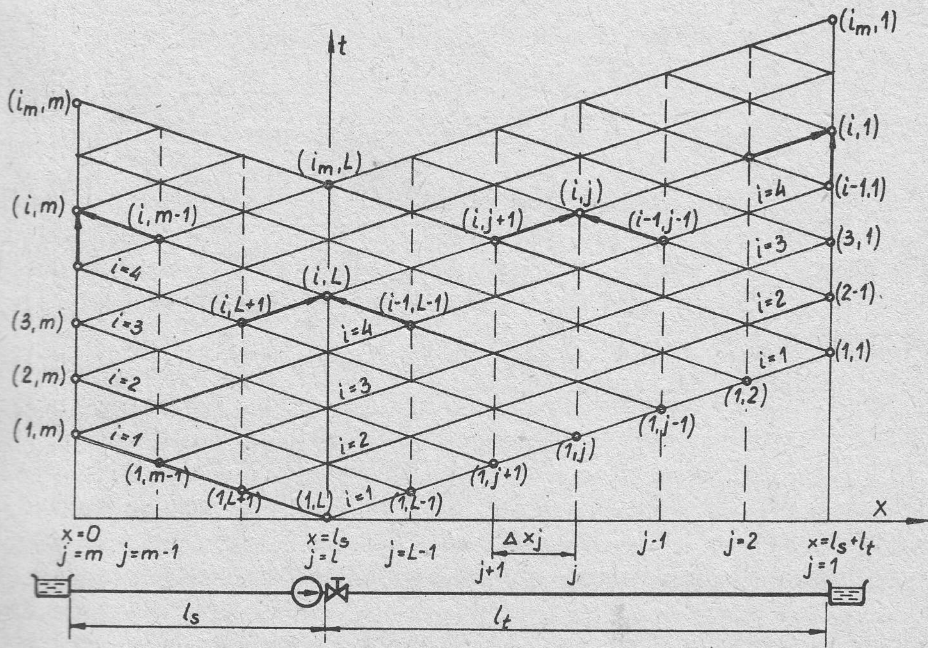

Fig. 6. Characteristics grid

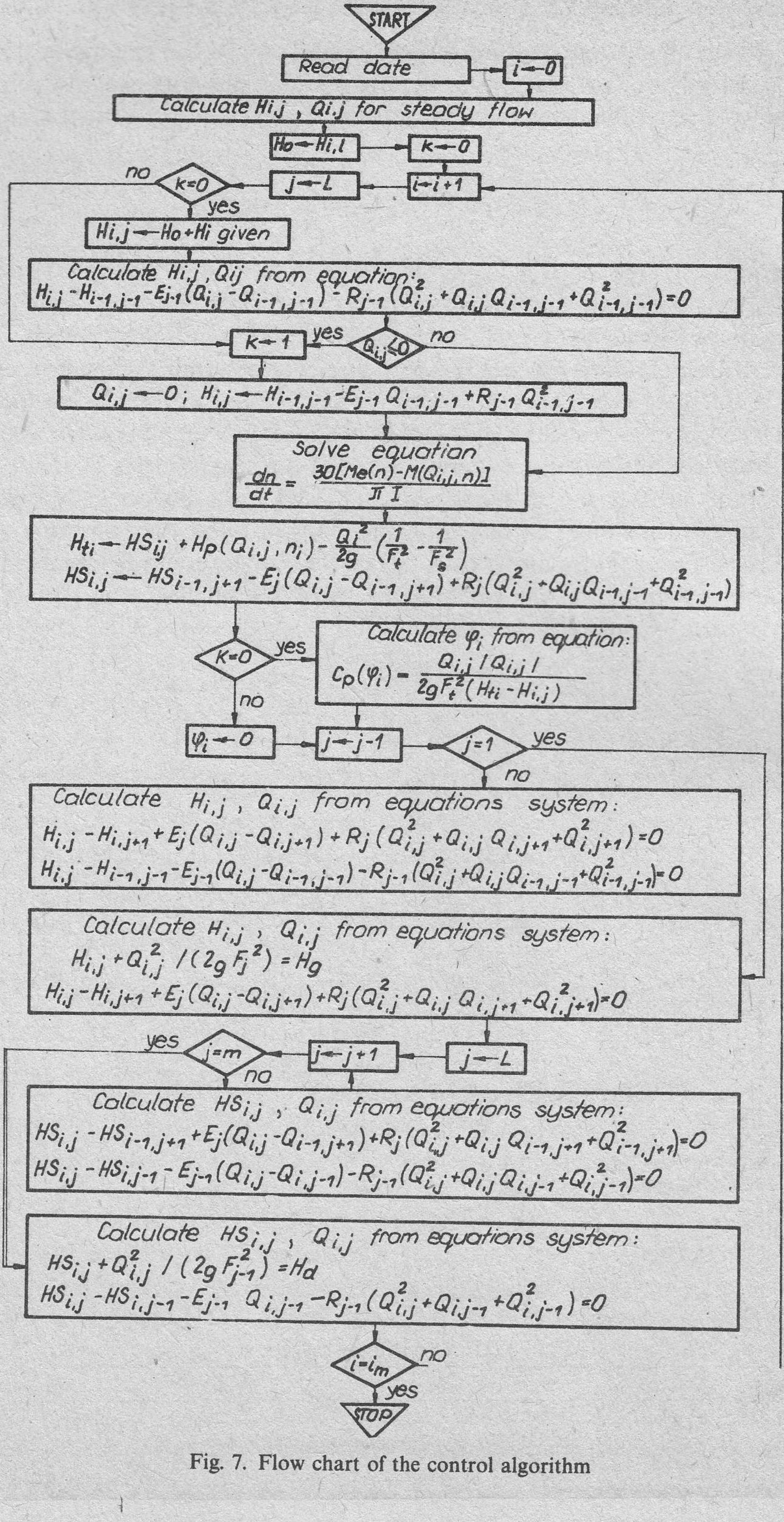

Fig. 7. Flow chart of the control algorithm

$$H_{i,j}-H_{i-1,j-1}-E_{j-1}(Q_{i,j}-Q_{i-1,j-1})-R_{j-1}(Q_{i,j}^2+Q_{i,j}Q_{i-1,j-1}+Q_{i-1,j-1}^2)=0 \quad (12)$$

for

$$x_{j-1}-x_j=-a_{j-1}\frac{\Delta t}{2}. \quad (13)$$

In Eqs (10) and (12) E_j and R_j stand for

$$E_j=a_j/(gF_j)$$
$$R_j=\lambda_j(x_j-x_{j+1})/(2gD_jF_j^2)\varepsilon_j$$

respectively, where

$$\varepsilon_j=\begin{cases} 1 \text{ for } (Q_{i,j}+Q_{i,j+1})>0 \\ 0 \text{ for } (Q_{i,j}+Q_{i,j+1})=0 \\ -1 \text{ for } (Q_{i,j}+Q_{i,j+1})<0 \end{cases}$$

Indices i and j numbering the H and Q values correspond to the modes of characteristics (Fig. 6) while index j numbering the values of a, λ, D, F, E and R refers these values to the j^{th} segment of the pipeline section.

The algorithm of calculations is shown in Fig. 7 in the form of a flow chart; the computer program being described in detail in reference [3].

4. Experimental Verification of the Control Program

4.1. Experimental Stand

The experiments have been carried out at the laboratory of the Institute of Fluid-Flow Machinery of the Polish Academy of Sciences in Gdańsk. The experimental stand is schematically shown in Fig. 3. The pump (1) coupled with an electrodynamometer (2) sucks water out of the reservoir (5) and delivers it into the reservoir (6). The water level in both containers is almost constant. The level of water in the delivery reservoir is kept constant by means of an overflow (9). The butterfly valve (7) is used to generate the waterhammer and to change the pump operation parameters. The valve can be opened and closed both manually and mechanically. The mechanical closing of the valve is actuated by a properly shaped driving cam.

The measurement system installed at the experimental stand allows to record the pump operation parameters and flow parameters in the pipeline under steady flow conditions as well as the pressure variations occurring in the pressure pipeline in transient states.

These variations were measured and recorded using the inductive pressure transducer (12) connected in a half-bridge system with strain-gauge amplifier (15) and a loop oscillograph (16). The process of butterfly valve closing was recorded simultaneously with the pressure variations. To this aim a potentiometer (14) and a strain-gauge amplifier (15) were used together with a loop oscillograph (16).

4.2. Experiments

The pump static characteristics (Fig. 5), the valve throttling characteristic (Fig. 4), the frictional losses factor in the pipelines and the initial parameters — the water flow rate Q_0, and the pump rotational speed n_0 — were determined under steady state conditions.

The control program was used to determine the function $\varphi(t)$ describing the valve closing for three assumed cases of waterhammer course in the delivery conduit. In all the cases under consideration the same initial conditions of water flow are assumed together with the same value of the total valve closure time and the same constraint connected with the maximum velocity of the valve butterfly position angle change. Under the last assumption the maximum valve closure speed should occur in the range between 100 per cent and 40 per cent of full valve opening. The pre-set pressure-time function at the valve outlet and, the corresponding to them, calculated valve opening and pressure at the pump outlet are given in Fig. 8 as the time curves.

The calculated valve closing-time functions $\varphi(t)$ were attained during experimental runs using the properly shaped cams controlling the valve butterfly position.

The measured pressure and valve opening have been compared with the assumed or calculated values (Fig. 8). The comparison indicates that in the first two cases (Figs 8a and 8b) the agreement between theoretical and experimental data is sufficiently good. The last remark refers only to the valve closing phase. The maximum difference between measured and calculated values of the pressure is in this phase less than 15 per cent with respect to the pressure under the steady state conditions.

Certain differences of pressure values which appear in this phase can be linked to the small, but perhaps significant, differences between real and numerically determined valve closing-time curves. In the third case large differences in the valve closing phase (Fig. 8c) appeared.

The causes for these differences can be sought in the appearance of the zones filled with vapour-gas bubbles (cavitation zones) at those places in the conduit in which the pressure falls down to its critical value close to the liquid evaporation pressure at given temperature [4]. The disturbance of the flow continuity, often referred to as liquid column separation has appeared also in the second case at the end of the valve closing process (Fig. 8b). The differences noticed in the phase of attenuating pressure free oscillation regard mainly the damping intensity and the frequency of oscillations. Comparing them with the calculated data the oscillations *are in the reality significantly more strongly damped down and their frequency is* lower. It can be suggested that these differences are due mainly to the following phenomena neglected in the theoretical model:

— gas and vapour cavitation [1, 2, 4],

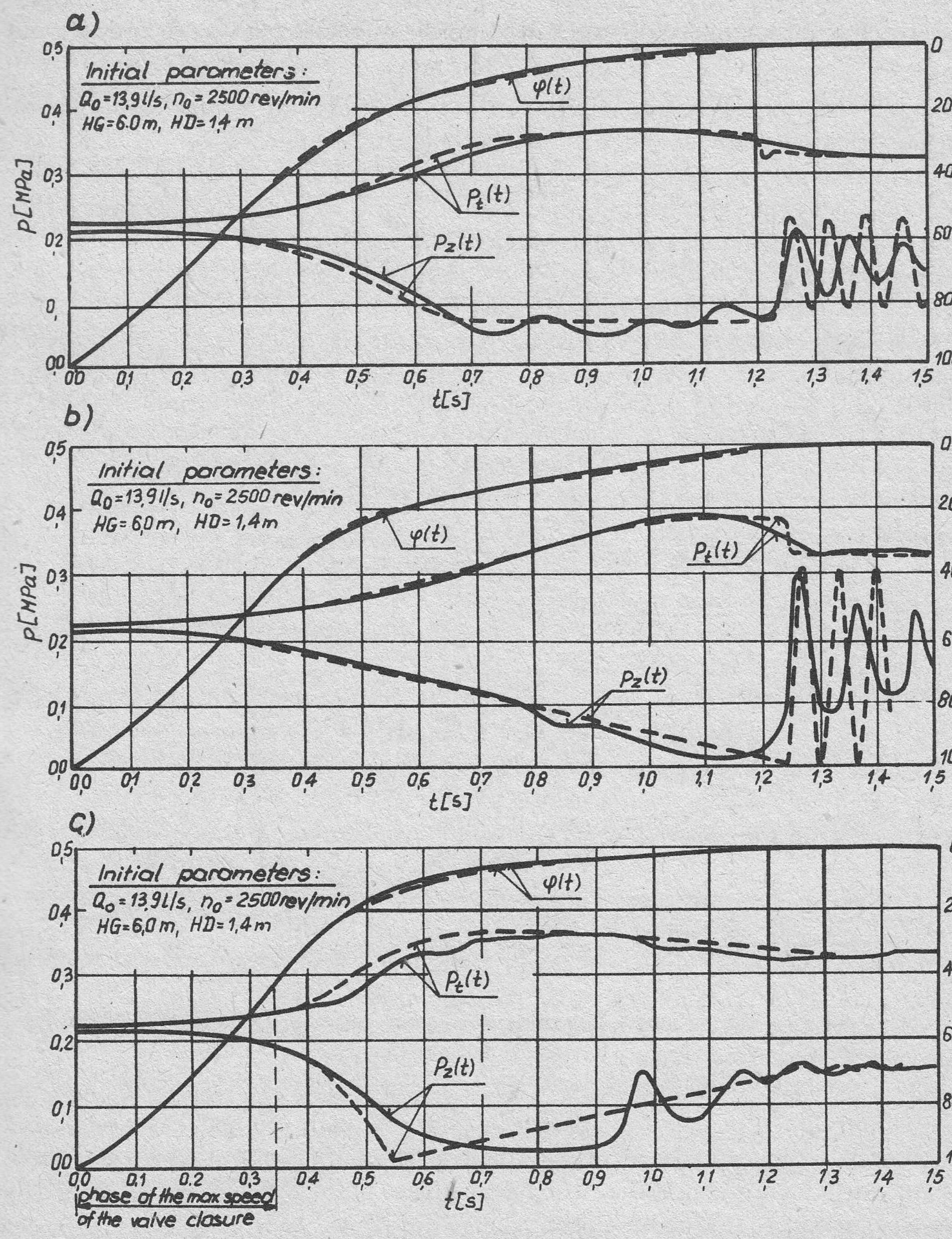

Fig. 8. Calculated valve closure modes $\varphi(t)$ and pressure values vs time curves at the pump outlet $p_t(t)$ for a given pressure-time functions at the valve outlet $p_z(t)$ and respective experimental curves
– – – – – – – preset and calculated curves, ———— experimental curves

— change of the pressure wave speed caused by the presence of free gases in the liquid [1, 2],
— energy dissipation in the walls of conduits and their supports and interaction between the pressure waves in the liquid and the stress waves in the conduit wall [1, 4].

5. Final Remarks and Conclusions

Figure 9 shows the results of experiments performed for an arbitrarily chosen valve closing mode and compared with experimental data taken from Fig. 8, i.e. with the results obtained by applying the control program. This comparison makes clear the usefulness of waterhammer control and allows to estimate the influence of different valve closing modes on waterhammer. All cases given in Fig. 9 are characterized by identical time of valve closure and the same initial conditions of water flow. Highest pressure values occurred in the case of arbitrary valve closing mode (curves no. *4*). The closing of the valve carried out according to curve no. *3* caused the fastest supression of the pressure changes while the same process carried out according to curve no. *1* allowed to avoid the column separation of liquid.

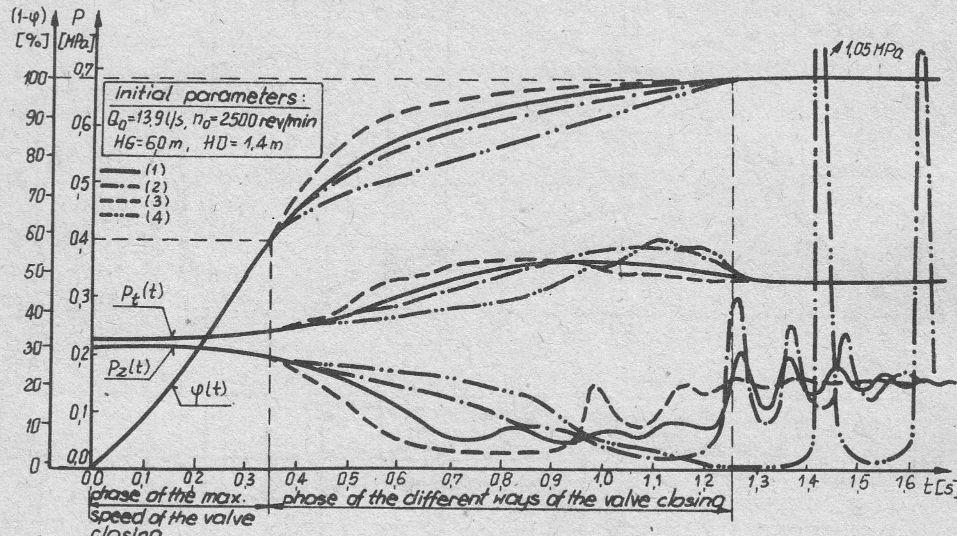

Fig. 9. Measured valve closure modes $\varphi(t)$ and the respective pressure variations at the valve outlet $p_z(t)$ and inlet $p_t(t)$

Attention should be paid to the order in which the valve closing curves are traced. This order corresponds to the increase of the amplitude of the pressure free oscillations.

Also worth mentioning is the strong dependence of the pressure variations on the valve closing course (the valve closing curves are placed very close to one another and small modification in the closing procedure results in large changes of the waterhammer).

The control program discussed in this paper allows for waterhammer reduction for the given time of valve closure or to shorten this time while keeping the waterhammer effects in the admissible limits. Additionaly, the stabilization of free pressure oscillations can be hastened by diminishing the difference between the pressure value at the moment of complete valve closure and its value in the equilibrium state at the considered cross-section of the conduit (Fig. 8c). For above listed reasons this program can be used to improve the operation under transient conditions of the hydraulic machines mated with long conduits.

Use of the program presents, however, certain disadvantage. The valve closing mode for which the total closure time would agree with its required value and which would eliminate the pressure free oscillations can be determined only by the method of successive approximations. This entails a long time of calculations and suggests that the program ought to be further improved. The program requires also improvements from the point of view of phenomena connected with the transient liquid flow in conduits. On the other hand an experimental verification in case of positive waterhammer is needed.

References

[1] E. B. Wylie, V. L. Streeter, *Fluid transients*. Mc Graw-Hill, New York, 1979.

[2] A. Adamkowski, *Analiza uderzenia hydraulicznego i sterowanie jego przebiegiem za pomocą zaworu w układzie pompowym* (Waterhammer analysis and control by means of a valve installed in the pump system). Oprac. IMP PAN 5/85.

[3] A. Adamkowski, *Program obliczeniowy SNP* (Numerical program SNP). Oprac. IMP PAN 6/85.

[4] A. Adamkowski, K. Steller, *Zmiany ciśnienia spowodowane naruszeniem ciągłości przepływu cieczy w układzie pompowym* (Changes of pressure caused by the flow continuity disturbance in the pump system). Zeszyty Naukowe Politechniki Śląskiej, Energetyka, z. 87, 1984.

[5] S. Bednarczyk, *Ruch nieustalony cieczy w przewodach pod ciśnieniem* (Transient liquid flow in conduits under pressure). Arch. Hydrot., t. XXI, z. 4, 1974.

Sterowanie uderzeniem hydraulicznym w przewodzie tłocznym

Streszczenie

We wstępie podkreślono znaczenie efektów dynamicznych związanych z nieustalonym przepływem cieczy a szczególnie z uderzeniem hydraulicznym. Wskazano na potrzebę sterowania uderzeniem hydraulicznym w układach pompowych, a zwłaszcza w układach z odwracalnymi maszynami hydraulicznymi. Na podstawie wyników wcześniej przeprowadzonych badań zilustrowano wpływ wymuszenia nieustalonego przepływu (sposobu zamykania zaworu) na przebieg uderzenia hydraulicznego (rys. 1 i 2). Te wyniki wskazują na możliwość ograniczenia poziomu uderzenia hydraulicznego za pomocą odpowiedniej regulacji natężenia przepływu.

Dalej przedstawiono algorytm sterowania przebiegiem uderzenia hydraulicznego za pomocą zaworu. Wyznaczono trzy przebiegi zamykania zaworu dla założonych przebiegów zmian ciśnienia w wybranym przekroju rurociągu tłocznego (rys. 8). We wszystkich przypadkach przyjęto takie same początkowe warunki przepływu oraz ten sam czas całkowitego zamykania zaworu. Wyniki obliczeń sprawdzono doświadczalnie na stanowisku pompowym w laboratorium Instytutu Maszyn Przepływowych PAN w Gdańsku.

Na zakończenie wskazano na kierunki dalszych prac w dziedzinie uderzeń hydraulicznych, a szczególnie na konieczność opracowania odpowiednich programów do sterowania uruchamianiem i zatrzymywaniem maszyn hydraulicznych współpracujących z długimi przewodami w celu obniżenia obciążeń hydrodynamicznych.

Управление гидравлическим ударом в напорном трубопроводе

Резюме

Во вступлении подчёркивается значение динамических эффектов связанных с неустановившимся течением жидкости, а особенно с гидравлическим ударом.

Указывается на необходимость управления гидравлическим ударом в насосных системах, а особенно в системах с обратимыми гидравлическими машинами. На основе результатов раньше произведённых исследований иллюстрируется влияние вынуждения неустановившегося течения (способа закрытия клапана) на ход гидравлического удара (рис. 1 и 2). Эти результаты показывают на возможность ограничивания уровня гидравлического удара с помощью соответствующей регуляции интенсивности течения.

В дальнейшей части представлен альгоритм управления ходом гидравлического удара с помощью клапана. Определены три хода закрытия клапана для предположенных распределений изменений давления в избранном сечении напорного трубопровода (рис. 8). Во всех случаях приняты такие же самые начальные условия течения и то же самое время полного закрытия клапана. Результаты расчётов проверены экспериментально на стенде для исследования насосов в лаборатории Института проточных машин Польской Академии Наук в Гданьске.

В заключении указывается на направления дальнейших работ в области гидравлических ударов, а особенно на необходимость обработки соответствующих программ для управления пуском и останавливанием гидравлических машин, содействующих с длинными трубопроводами с целью уменьшения гидродинамической нагрузки.