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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machines

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PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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ADAM ADAMKOWSKI[1]

Effect of flow throttling configurations of plug-type check valve on waterhammer in a pump delivery pipeline – experimental investigation[2]

A new plug-type check valve has been designed and tested to reduce the intensity of waterhammer in pump systems. Novelty of the design lies in simultaneous application of two ways of flow throttling within the valve. One of them takes place in the configuration formed between a damping annulus mounted to the valve case and a sharpedged annulus of the valve head. The other is accomplished in the configuration of valve head - valve seat by means of a properly shaped valve head.

Experimental tests have been carried out on a pump test rig for different configurations of a specially prepared valve. Waterhammer is a result of valve self-closing after disconnection of the pump motor. Application of only one of the proposed methods of flow throttling without use of the other does not result in diminishing the impact of waterhammer which remains comparable to that of the undamped valve. Only simultaneous throttling in both mentioned configurations can produce a desired effect.

The main conclusion of the paper is that the newly constructed valve is capable of considerably reducing the level of waterhammer in pipeline systems due to pump power failure.

Nomenclature

A	– cross sectional area,	m	– mass,
b	– boundary layer thickness,	n	– pump rotational speed,
C_F	– hydraulic thrust coefficient,	n_{11}	– pump unit rotational speed,
C_Q	– flow discharge coefficient	p	– pressure,
D	– pipe inner diameter,	Q	– volumetric flow rate,
D_w	– pump impeller diameter,	Q_{11}	– pump unit discharge,
e	– pipe wall thickness	t	– time,
E	– elasticity modulus,	T	– pump torque,
F	– force,	T_{11}	– unit pump torque,
g	– acceleration of gravity,	v	– flow velocity,
H	– pressure head,	x	– distance coordinate,
H_p	– pump head	z	– valve head position,
h	– height,	β	– valve axis angle,
I	– moment of inertia,	ρ	– liquid density.

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Subscripts

d	–	of pump discharge side,	o	–	of initial conditions,
f	–	of friction,	p	–	of pump,
g	–	of check valve head,	s	–	of pump suction side,
h	–	of hydraulic interaction,	v	–	of check valve,
i	–	of internal flow interaction,	z	–	of valve outlet.
l	–	of liquid,			

1. Introduction

Check valves are often installed in pump systems to prevent flow reversal under conditions of pump power failure, thus preventing the change of direction of the pump impeller rotation. The position of the valve closing element depends on working regime of the pump. During pumping the closing element is in "open" position (the valve can be completely or partially open), during stoppage time is in "closed" position. After an abrupt disconnection of the pump motor, the valve automatically closes due to a flow decrease.

The speed at which the valve closes can vary, mainly according to its dynamic properties and the rate of flow decrease, thus according to the pressure difference between the delivery and suction reservoirs, and the moment of inertia of rotating elements of the pump unit. Often in practice, temporary changes in flow direction occur before the valve can reach its closed position. Especially in high-pressure systems or when the pumps work in parallel, the time-lag between the occurrence of reverse flow and the valve closure may be relatively large. Hydraulic forces acting on the closing element of the valve during flow reversal increase the speed of valve closure at its final stages. Two very unfavourable phenomena are observed then [2-4, 10, 15, 16, 18, 19, 20]:

- slamming of the valve, that is knocking of the closing element against the valve seat accompanied by a high level of noise,
- waterhammer in the pipeline caused by a sudden stop of reverse flow.

Greater velocity of reverse flow results in higher intensity of these "shock phenomena". Besides undesirable noise and excessive dynamic load of components of the flow system, these phenomena are responsible for a large number of failures and breakdowns [1, 13, 15, 23]. According to Wilkinson [23], about 16% of breakdowns in British power stations were caused by inappropriate operation of check valves.

New constructions of check valves are being worked out in order to minimize the intensity of valve slamming and hydraulic transients in the pipelines. Considering types of closing elements, check valves can be divided into two main groups:

1. swing valves-check valves with rotating closing elements,
2. plug-type valves-check valves with translating closing elements.

Within each group one can distinguish damped or undamped valves. In the undamped valves, the position of the valve closing element is controlled by flowing liquid only. In the damped valves, the position of the head is additionally influenced by a damping device installed to reduce the speed of the final stages of valve closure. The damped check valves, hitherto not often used, are now winning the favour of manufacturers to become part of their broadened assortment of products.

However, undamped check valves are also being improved. Endeavours of construction engineers are centred on increasing the speed of valve closure through, for example, (1) application of specific weight and spring configurations to increase the impact of external forces acting in the direction of closing or (2) application of many, instead of one, valve moving elements to reduce their inertia [7, 8, 10, 15, 19]. These measures give often satisfactory results as far as reducing the intensity of shock phenomena because for particular installations one can often select such check valves which fulfill technical requirements determined by an acceptable range of pressure variations. However, this is not always possible because one can not either increase closing forces nor reduce inertia of closing elements above certain limits. In order to prevent the pipelines from damage done by waterhammer, very expensive air accumulators are sometimes used to keep down oscillations of pressure, however not to reduce the intensity of valve slamming. A less expensive construction, suggested by some authors, is grounded on application of check valves equipped with damping devices. If properly designed, they can be suitable for reducing the effects of both valve slamming and hydraulic transients.

Reviewing works published over recent years, one can notice a considerable progress in recognition of dynamic behaviour of undamped check valves and their effect on the shock phenomena. In the papers [4, 10, 11, 15, 17, 19], dynamic properties of different check valves are displayed in the form of so-called dynamic characteristics proposed by Provoost [16]. The characteristics present the relations between the maximum velocity of reverse flow and its deceleration in the pipeline. These characteristics can be helpful in finding appropriate valves for particular installations to prevent destructive impact of the shock phenomena. Therefore, manufacturers of check valves are being persuaded to supply their products with dynamic characteristics in a similar way which is customary on the pump market where performance characteristics are part and parcel of the product [20]. This way of doing business is strongly necessitated by the fact that mathematical modelling of dynamic behaviour of valves still comes across major difficulties. So far, no satisfactory method of predicting performance characteristics of check valves under transient flow has been worked out [15-17, 19]. Another, so far, not resolved problem is precise determination of hydraulic and friction forces acting on the valve closing elements and this in turn is crucial for further precise determination of a function describing valve closure and hydraulic transients. It is clear that one obstacle in developing reliable mathematical model to account for dynamic behaviour of check valves is certainly shortage of experimental evidence.

Information about damped check valves can be found in [5, 9, 11, 12, 14, 22]. Ellis [5], Kane [9], Wang [22] presented results of their investigations on swing valves with external damping devices, intended to keep down the speed of the final stages of valve closure. It should be noted that these works are mostly theoretical. However, Kruisbrik, Throley [11], drawing on dimensional analysis, made efforts to formulate dynamic characteristics of damped check valves using a model of swing valve with an external damping device. A bottom line of this paper is extension of Provoost's [16] dynamic characteristics on damped check valves. It is suggested that experimental investigations should be carried out in order to confirm usefulness of these characteristics. Plug-type check valves are subject of papers [12, 14], especially in terms of their effect on hydraulic transients. Results of theoretical analysis of a plug-type valve with a cylindrical closing element slowed down along its whole displacement thanks to special clearances in the cylinder, are presented by Kubie [12]. This design is not one of the most successful due to an undesired delay in early stages of valve closure. Müller [14] outlined the problem of application and investigation of plug-type valves pointing out advantages that can be achieved from damping the final stages of valve closure.

In the light of the above presented review, one can acknowledge that know-how concerning check valves is incomplete. A striking fact is lack of reliable mathematical models and well-documented experimental investigations. It refers in particular to damped check valves. Moreover, problems connected with design of new improved check valves to reduce the impact of the shock phenomena have not met even a slightest response in the literature.

Better access to results of experiments hitherto carried out as well as more investigations relevant to the topic could certainly lead to better recognition of phenomena involved in operating of the check valves and to verification of mathematical models and finding the causes of their divergence, if any, from the experiments. In order to bridge the gap, a physical model of a new original damped check valve has been put forward and appropriate experimental investigations have been conducted at the Liquid Dynamics Department of the Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Gdańsk. The experimental investigations aimed at determination of the influence of proposed methods of damping on hydraulic transients.

2. Physical model of damped plug-type check valve

A flow diagram of the proposed damped plug-type check valve is presented in Fig. 1. A cylindrical closing element (valve head) is equipped with a conical element 1, sharpedged annulus of the head 2 and pivot 3. During valve closure, the head is pressed down towards the surface of the valve seat 4. Valve opening (in Fig. 1, the valve head is in open position) is caused by liquid flow of rate Q. Let us consider the behaviour of the closing element when the flow rate decreases. Valve closure begins at the instant when the resultant hydraulic force acting on the valve head in the axial direction begins having lower values than the axial component of gravity and frictional forces combined. Displacement of the head is accompanied by internal flow of rate Q_i towards the upper part of the valve case

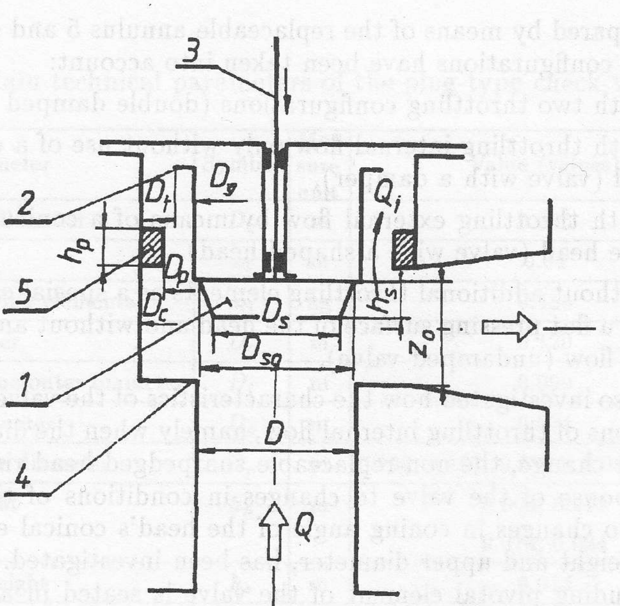

Fig. 1. A flow diagram of the proposed damped plug-type check valve.

above the head. Internal flow is not throttled during early stages of valve closure. It is assumed that the hydrometric area between the inner wall of the valve case and the sharpedged annulus of the head is not smaller than the area of the valve inlet section. When the head goes down, internal flow is throttled according to the size of the clearance between the two damping annuli. As a result of this throttling, the speed of the head in the final stages of valve closure should be reduced. Besides throttling internal flow, there is also a configuration of throttling external flow consisted of the conical element 1 and the valve seat 4. When the head moves down towards the seat, the pass between these two elements becomes narrower. Application of the conical element leads to a change in flow characteristics of the valve.

Novelty of the above proposed design lies in application of two configurations of throttling liquid flow within the valve. One of them consists in throttling internal flow in the clearance between the sharpedged annulus of the head and the annulus of the case, thus reducing the speed of valve closure in its final stages. The other increases throttling external flow through the valve when it is partially open.

An experimental model of plug-type check valve has been constructed, based on the above concept, in order to conduct experimental investigations. Responding to the necessity of investigating the influence of the valve on hydraulic transients, various damping configurations with respect to internal and external flow have been prepared by means of the replaceable annulus 5 and conical element 1. The following configurations have been taken into account:

- valve with two throttling configurations (double damped valve),
- valve with throttling internal flow only without use of a conical element of the head (valve with a damper),
- valve with throttling external flow by means of a conically shaped end of the valve head (valve with a shaped head),
- valve without additional throttling elements as a special case of damped valve with a flat pressing surface of the head and without annuli for throttling internal flow (undamped valve).

It has been also investigated how the characteristics of the valve respond to changes in conditions of throttling internal flow, namely when the diameter and height of the damper change, the non-replaceable sharpedged head ring assumed. Likewise, the response of the valve to changes in conditions of throttling external flow, that is to changes in coning angle of the head's conical element, assuming its constant height and upper diameter, has been investigated.

The protruding pivotal element of the valve is seated in a low-friction slide bearings. This construction enables us to measure the hydraulic force exerted by the liquid on the valve head and to record the position of the head. Mounting of the additional weights is also possible. Main technical data referring to the replaceable and non-replaceable elements of the valve are shown in Tab.1.

3. Experimental rig

Experimental investigations of the plug-type check valve were carried out on a pump test rig in the laboratory of Liquid Dynamics Department at the Institute of Fluid Flow Machinery, Polish Academy of Sciences, Gdańsk, Poland. A diagram of the hydraulic installation of the rig is presented in Fig. 2. The installation consists of a suction reservoir, short suction pipeline, centrifugal pump, proposed plug-type check valve, butterfly valve, delivery pipeline, delivery reservoir and over-flow pipeline with a control valve to set pressure in the delivery reservoir. The check valve has been installed in vertical position in the delivery pipeline close to the outlet orifice of the pump within a distance of two inner diameters of the pipeline from the delivery nozzle. An electrodynamometer with controllable rotational speed is used as the pump motor. The shaft of the pump is connected to the shaft of the electrodynamometer by means of an electromagnetic coupling to allow quick disconnection of the motor.

Main technical data referring to the components of the test rig have been inserted in Tab. 2, whereas static characteristics of the applied 65PJM160-type centrifugal pump obtained during previous tests are presented in Fig. 3. The characteristics of the pump are presented in the form of unit characteristics

Table 1. Main technical parameters of the plug-type check valve

Parameter	Symbol	Measure unit	Value (values)
Pipe inlet/outlet diameter	D_v	m	0.066
Maximum lift	z_0	m	0.04
Total mass of moving elements	m_i	kg	0.456
Case inner diameter	D_c	m	0.120
Sharpedged annulus outer diameter	D_t	m	0.099
Head cylinder diameter	D_g	m	0.074
Case annulus inner diameter	D_p	m	0.103, 0.101, 0.0996, 0.0994, 0.0992
Case annulus height	h_p	m	0.008, 0.012 0.016, 0.024
Conical element height	h_s	m	0.014
Conical element upper diameter	D_{sg}	m	0.0658
Conical element lower diameter	D_s	m	0.051, 0.055, 0.059, 0.062, 0.0645

$n_{11} - Q_{11}$ and $n_{11} - T_{11}$ where

$n_{11} = \frac{n \cdot D_w}{\sqrt{H_p}}$ – is the pump unit rotational speed,

$Q_{11} = \frac{Q}{D_w^2 \sqrt{h_p}}$ – pump unit discharge,

$T_{11} = \frac{T}{D_w^3 H_p}$ – pump unit torque.

The test rig has been provided with a measuring equipment capable of finding static and dynamic characteristics of the valve in steady and unsteady flow regimes.

4. Investigation of static characteristics of the valve

The valve has been investigated under steady flow conditions for five different conical elements as in Tab. 1., the sixth being of zero-thickness - valve head with a flat pressing surface. Within each case, two series of investigations have been performed in order to determine the influence of flow throttling on static characteristics:

- with a case annulus having inner diameter of $D_p = 99.2 \cdot 10^{-3}$ m,
- without a case annulus.

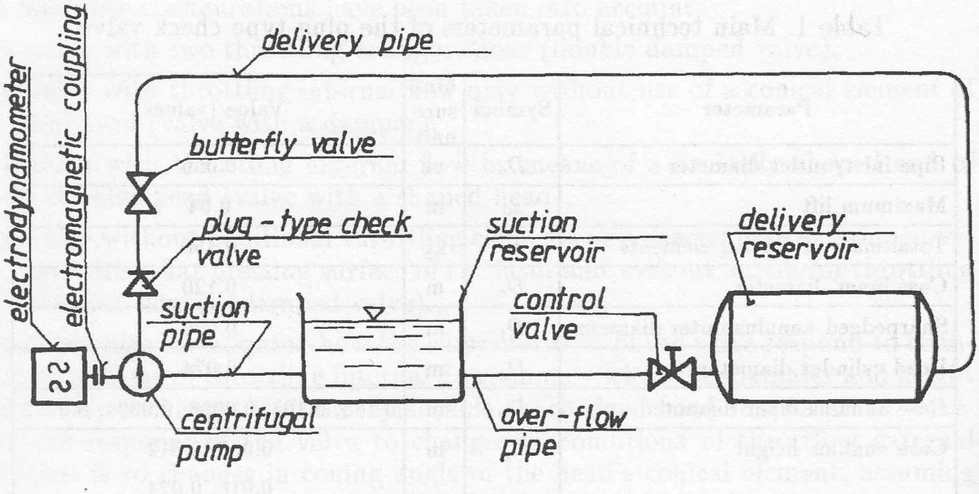

Fig. 2. Diagram of the test rig.

The investigations have been conducted for two opposite flow directions within a wide range of flow rate. The flow rate has been controlled by proper adjustment of the pump rotational speed whereas the change in flow direction has been achieved thanks to reverse installation of the valve. During the investigations, the following quantities have been measured:

- valve head position,
- volumetric flow rate,
- pressure drop in the valve,
- hydraulic thrust on the valve head.

A diagram of applied measuring equipment is shown in Fig. 4.

A strain gauge dynamometer has been applied in order to control the position of the valve head and measure hydraulic force. One end of the dynamometer has been connected to the valve pivot whereas the other has been connected to the valve case by a screw which also controls the position of the valve head. Pressure drop across the valve Δp_v has been determined from pressure measurements before and behind the valve at a distance of two and twelve inner diameters of the pipeline, respectively. Measured values of pressure have been recalculated then to obtain the pressure at the inlet and at the outlet of the valve, subtracting pressure losses due to friction in the pipeline stretches two and twelve diameters long, respectively. The friction coefficient has been taken from the graph of Colebrook and White.

Based on these investigations, the relations between the pressure drop in the valve (Δp_v), flow rate (Q) and hydraulic thrust (F_h) have been found for different positions of the valve head and with respect to all assumed configurations. No

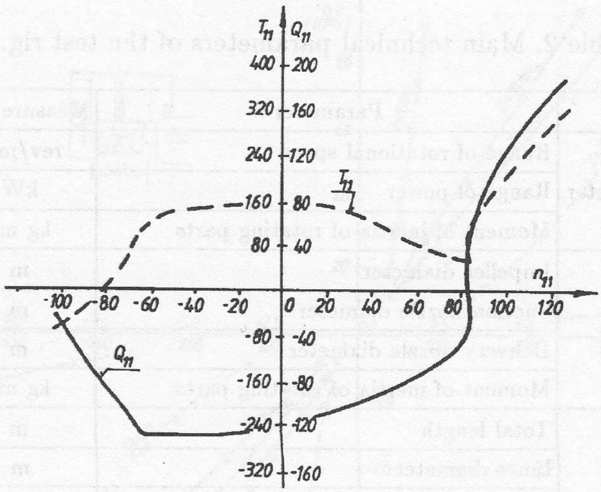

Fig. 3. Unit characteristics of the 65PJM160 centrifugal pump.

Fig. 4. Diagram of measuring system to investigate the static characteristics of the valve.

Table 2. Main technical parameters of the test rig.

Element	Parameter	Measure unit	Value
Electrodynamometer	Range of rotational speed	rev/min	±3000.0
	Range of power	kW	±10.0
	Moment of inertia of rotating parts	kg m^2	0.15
Centrifugal pump	Impeller diameter	m	0.16
	Suction nozzle diameter	m	0.08
	Delivery nozzle diameter	m	0.065
	Moment of inertia of rotating parts	kg m^2	0.007
Suction pipeline	Total length	m	6.2
	Inner diameter	m	0.08
	Wall thickness	m	0.0035
Delivery pipeline	Total length	m	29.3
	Inner diameter	m	0.066
	Wall thickness	m	0.002
Delivery reservoir	Total volume	m^3	1.6
	Volume of the air at atmospheric pressure	m^3	0.9
Open-air suction reservoir	Maximum volume	m	8.0
	Head range	m	0÷2

influence of the annulus 5 on the above relations has been found. Chosen results of investigations of the valve with the conical element having lower diameter of $64.5 \cdot 10^{-3}$ m are displayed in Fig. 5. Measured values on the $\Delta p_v(Q)$ curves apparently lie very close to square functions. Values referring to the characteristics of $\Delta p_v(F_h)$ seem to belong to linear functions. This suggests that the experimental curves should not be sensitive to the Reynolds number and could be approximated by square and linear functions, respectively.
Values of the flow discharge coefficient for each investigated position of the valve head can be determined by means of the least square approximation for $\Delta p_v(Q)$. This coefficient referred to the inlet cross sectional area A_v has been calculated from the formula:

$$C_Q = \frac{|\,Q\,|}{A_\nu \sqrt{2\,|\,\Delta p_\nu\,|\,/\rho}} \,.$$

Then, the hydraulic thrust coefficient has been evaluated based on linear approximation for $\Delta p_v(F_h)$. This coefficient has been defined as the ratio of the

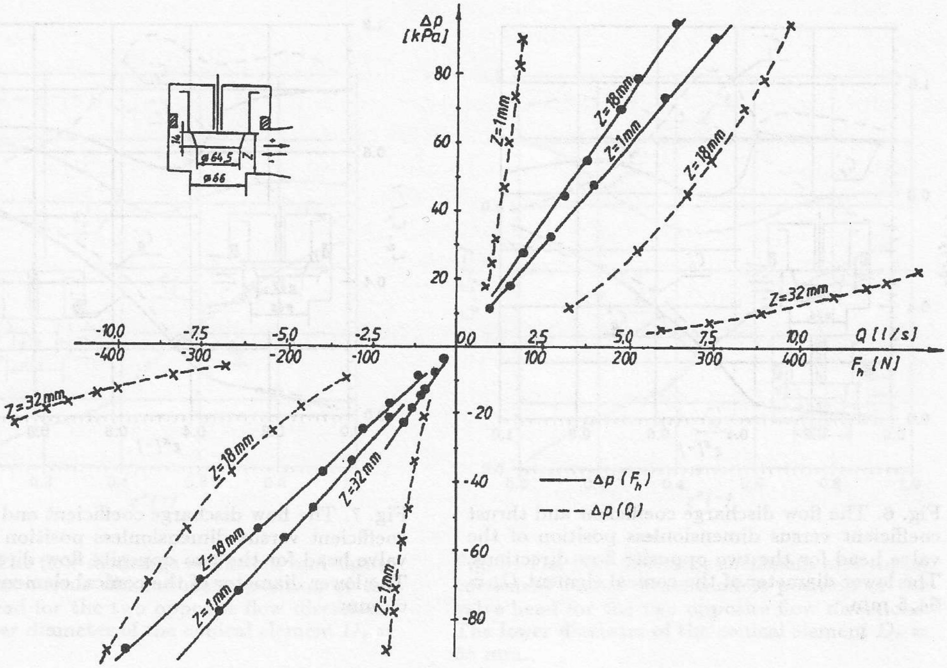

Fig. 5. Characteristics of the plug-type check valve with the conical element of lower diameter $D_s = 64.5 \cdot 10^{-3}$ m.

hydraulic thrust on the valve head and the force due to pressure difference between the inlet and the outlet of the valve

$$C_F = \frac{F_h}{A_\nu \cdot \Delta p_\nu} .$$

In Figs. 6-11, the flow discharge coefficient C_Q and the hydraulic thrust coefficient C_F referring to the two flow directions and different configurations are plotted against dimensionless position of the valve head ($z^* = z/z_o$). It becomes clear from comparison of the presented graphs that the shape of the valve head has a significant effect on the characteristics $C_Q(z^*)$ and $C_F(z^*)$. Application of conical elements with larger diameters D_s leads to smaller values of C_Q for a constant position of the head. Thus, the pressure drop across the valve and the hydraulic thrust become larger. It should be also noted that the influence of the conical element on these coefficients becomes smaller when valve opening increases and is insignificant when the valve is fully open.

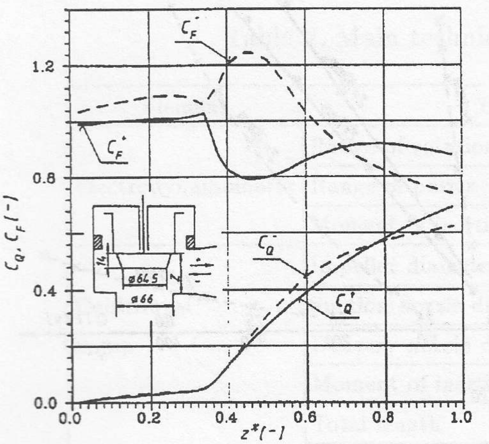

Fig. 6. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 64, 5 mm.

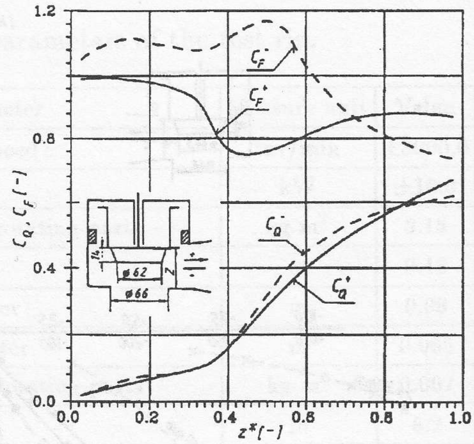

Fig. 7. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 62 mm.

5. Experiments under transient conditions

The experiments described in this chapter concern the situation of disconnecting the pump motor (by means of an electromagnetic coupling). The investigations have been carried out for all prepared configurations of the valve. For reliable comparison, each configuration has been tested starting from the same steady working conditions. These conditions have been determined as:

- pump rotational speed $n_o = 1500$ rev/ min ,
- volumetric flow rate $Q_o = 5.0 \cdot 10^{-3}$ m^3/s,
- water head in the suction reservoir above the centre line of the horizontal delivery pipe $H_{do} = 0$,
- absolute pressure in the delivery reservoir, referred to that of the delivery pipe level $p = 170$ kPa,
- temperature of water 9°C.

When the pump was stopped the following quantities were measured:

- pump rotational speed (n),
- valve head position (z),
- inlet pressure (p_d),
- outlet pressure (p_z), - volumetric flow rate (Q).

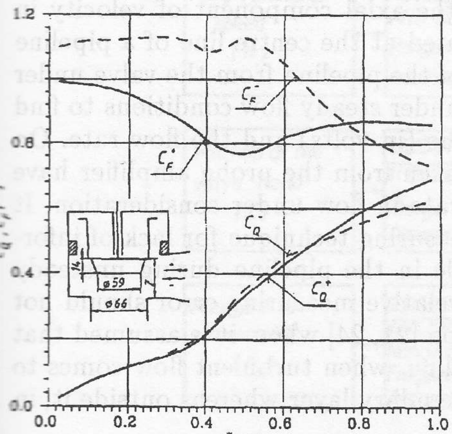

Fig. 8. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 59 mm.

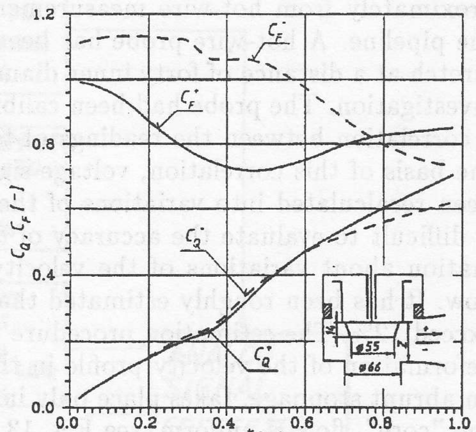

Fig. 9. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 55 mm.

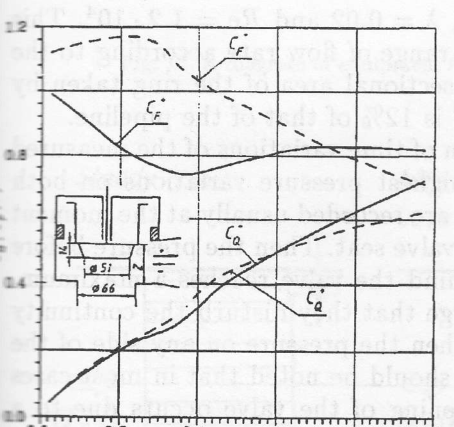

Fig. 10. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 51 mm.

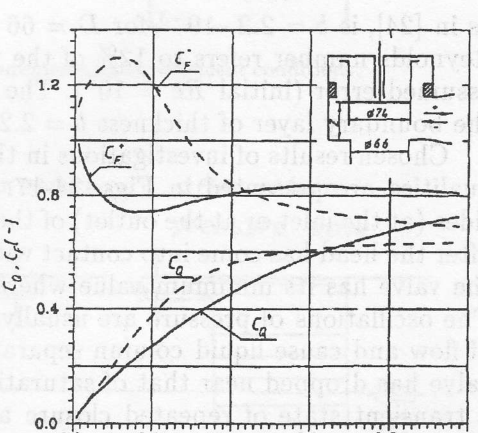

Fig. 11. The flow discharge coefficient and thrust coefficient versus dimensionless position of the valve head for the two opposite flow directions. The lower diameter of the conical element D_s = 0.

A diagram of measuring equipment for measurements of these quantities is presented in Fig. 12. Time variations of the flow rate have been determined approximately from hot-wire measurements of the axial component of velocity in the pipeline. A hot-wire probe has been mounted at the centre line of a pipeline stretch at a distance of forty inner diameters of the pipeline from the valve under investigation. The probe had been calibrated under steady flow conditions to find a correlation between the readings of the probe (in volts) and the flow rate. On the basis of this correlation, voltage signals taken from the probe amplifier have been recalculated into variations of the flow rate in flow under consideration. It is difficult to evaluate the accuracy of this measuring technique for lack of information about variations of the velocity profile in the pipeline during unsteady flow. It has been roughly estimated that the relative measuring error should not exceed 12%. The estimation procedure based on [21, 24] where it is assumed that deformation of the velocity profile in the pipeline, when turbulent flow comes to an abrupt stoppage, takes place only in the boundary layer whereas outside it, in the "core", flow is uniform, see Fig. 13.

In the light of these assumptions, the measuring error should not be higher than the ratio of cross sectional areas taken by the boundary layer (in the form of a ring) and the pipeline. The boundary layer thickness, determined from the formula

$$b = \frac{8 \cdot D}{\lambda \cdot Re}$$

as in [24], is $b = 2.2 \cdot 10^{-3}$ for $D = 66 \cdot 10^{-3}$, $\lambda = 0.02$ and $Re = 1.2 \cdot 10^4$. This Reynolds number refers to 12% of the tested range of flow rate according to the assumed error (initial $Re = 10^5$). The cross sectional area of the ring taken by the boundary layer of thickness $b = 2.2 \cdot 10^{-3}$ is 12% of that of the pipeline.

Chosen results of investigations in the form of time variations of the measured qualities are presented in Figs. 14-17. The highest pressure variations on both sides (at the inlet or at the outlet) of the valve are recorded usually at the moment after the head has come into contact with the valve seat. Then the pressure before the valve has its minimum value whereas behind the valve reaches a maximum. The oscillations of pressure are usually so large that they disturb the continuity of flow and cause liquid column separation when the pressure on any side of the valve has dropped near that of saturation. It should be noted that in most cases a transient state of repeated closure and opening of the valve occurs due to a temporary drop of pressure at the outlet below the pressure at the inlet.

Comparing the presented transients, one can see a considerable influence of the applied methods of damping liquid flow in the valve on variations of pressure within the system and on the speed of valve closure in its final stages. This speed may be deemed a measure of the intensity of valve slamming. For further considerations and presentation of some more important results during the course of investigations, the following dimensionless parameters of the valve have been

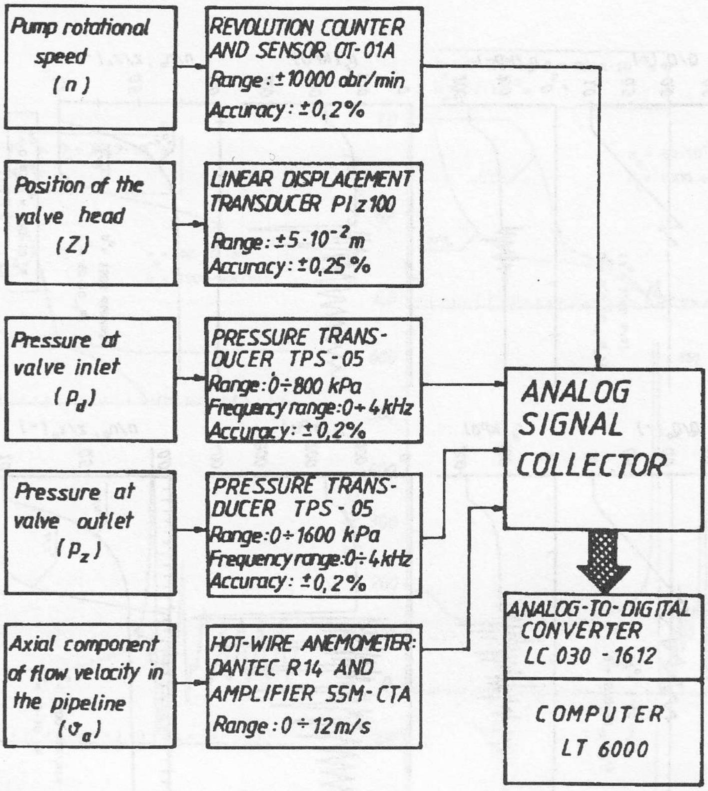

Fig. 12. A diagram of equipment for measurements under transient conditions.

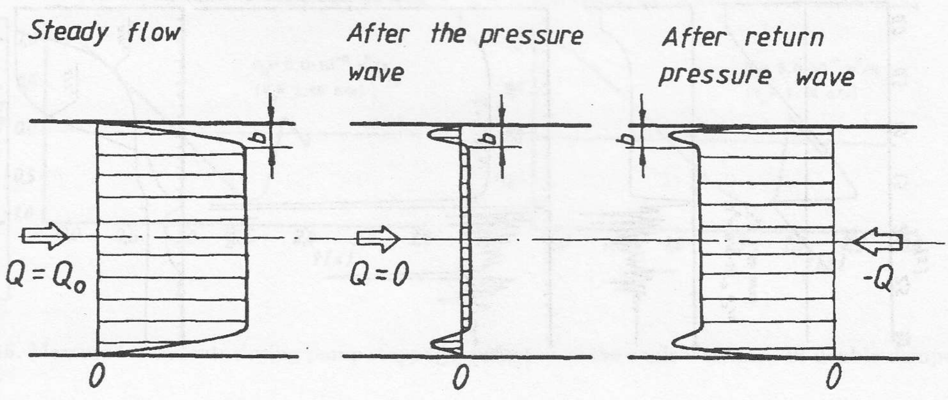

Fig. 13. Model of velocity distribution in turbulent pipe flow.

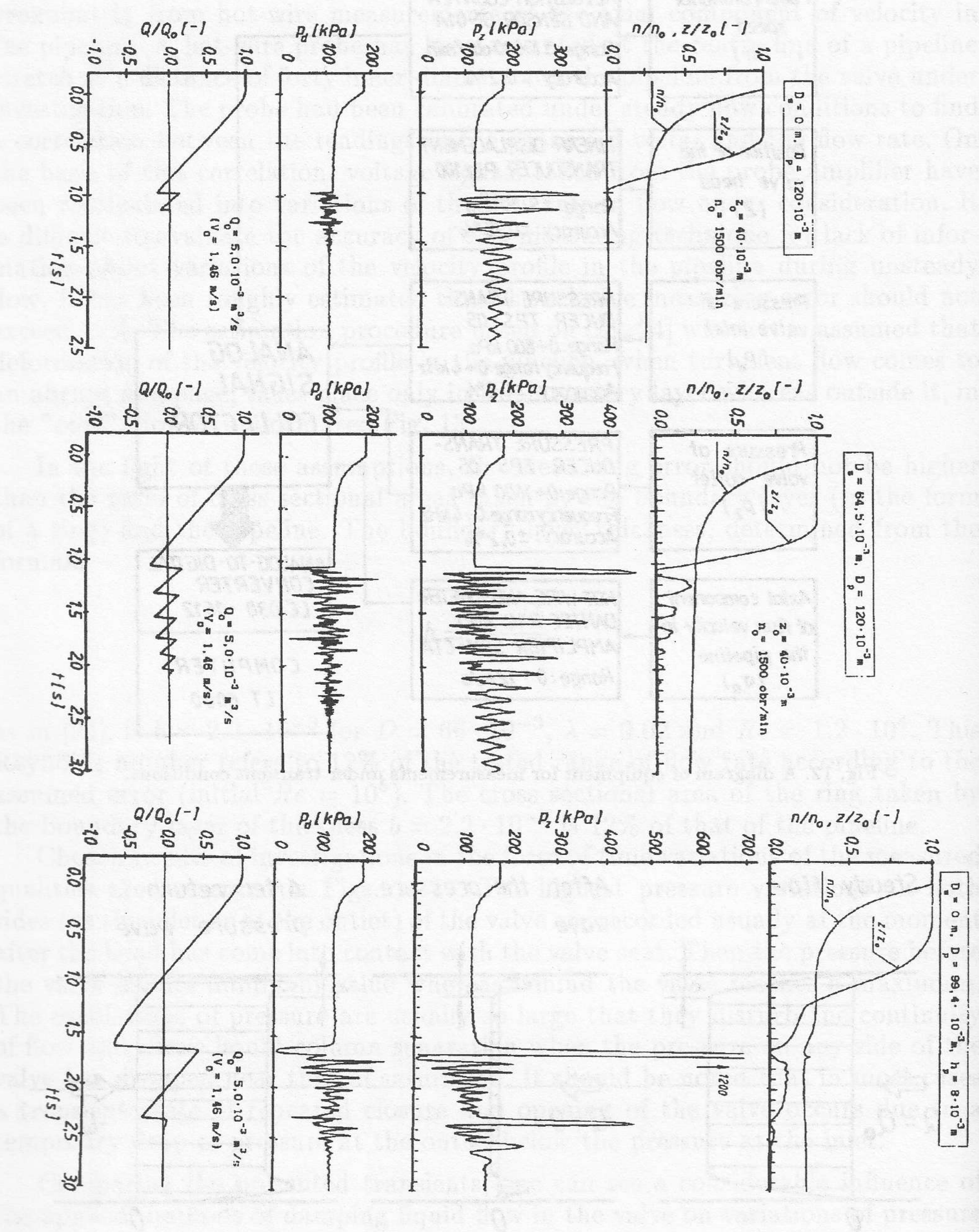

Fig. 14. Measured transients during pump stoppage: a. for undamped valve, b. for valve with conically shaped head (valve without damper), c. for valve with damper only (with flat press-down surface).

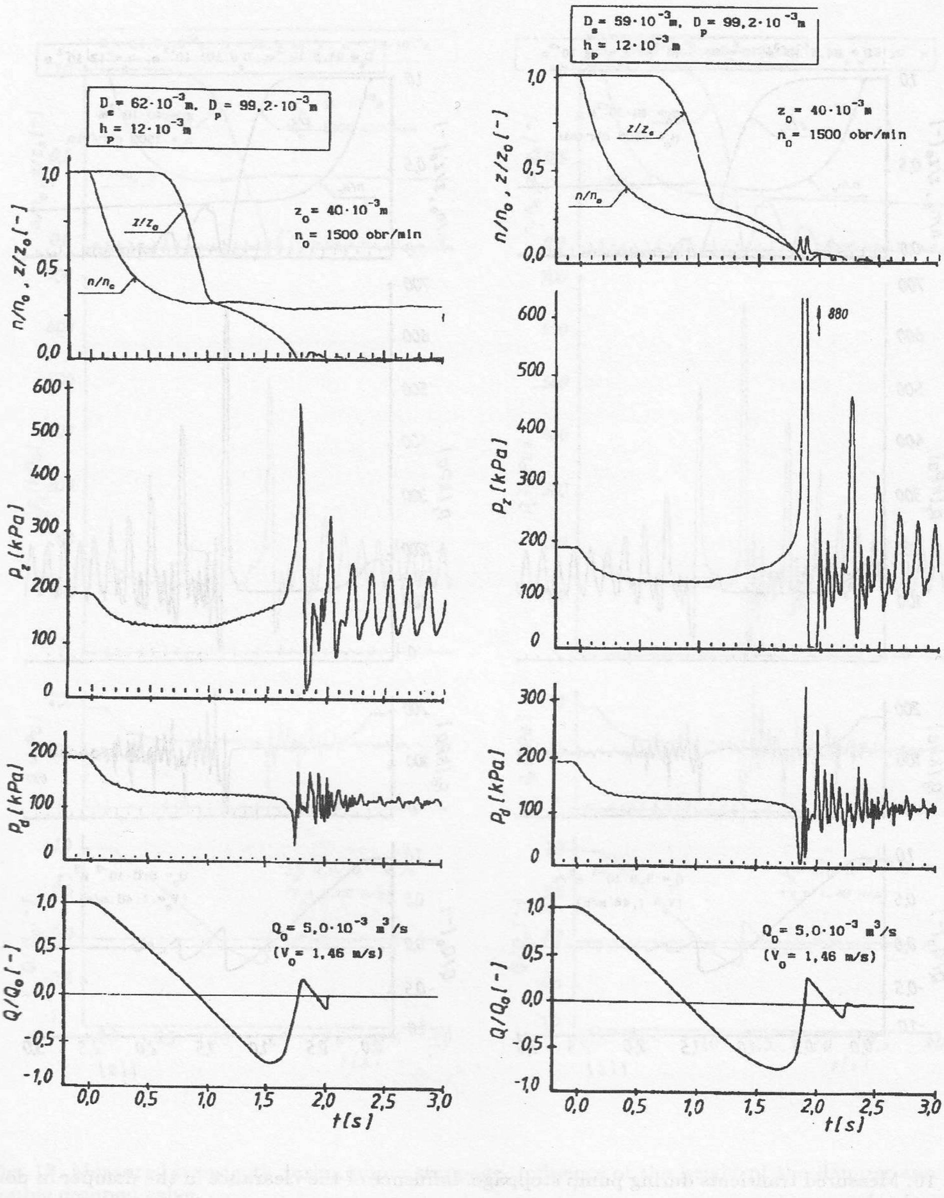

Fig. 15. Measured transients during pump stoppage. Influence of the conical element in double damped valve.

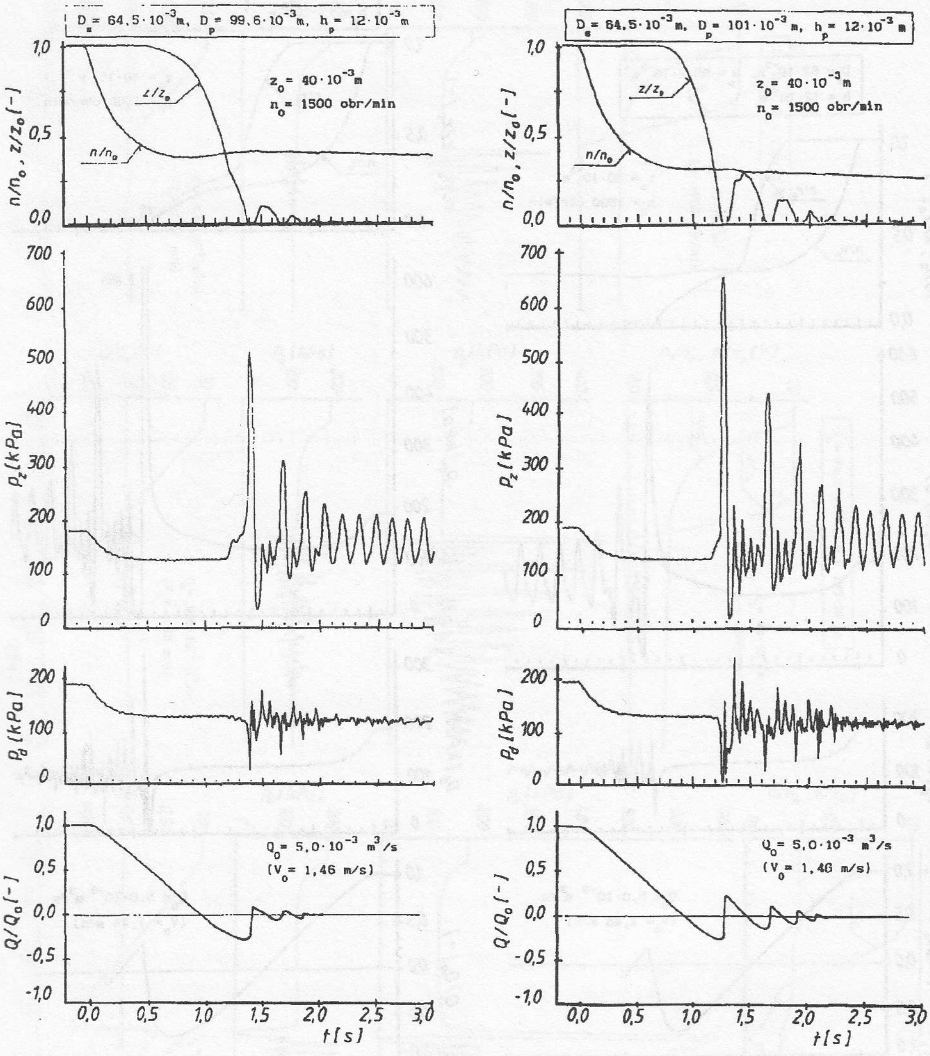

Fig. 16. Measured transients during pump stoppage. Influence of the clearance in the damper in double damped valve.

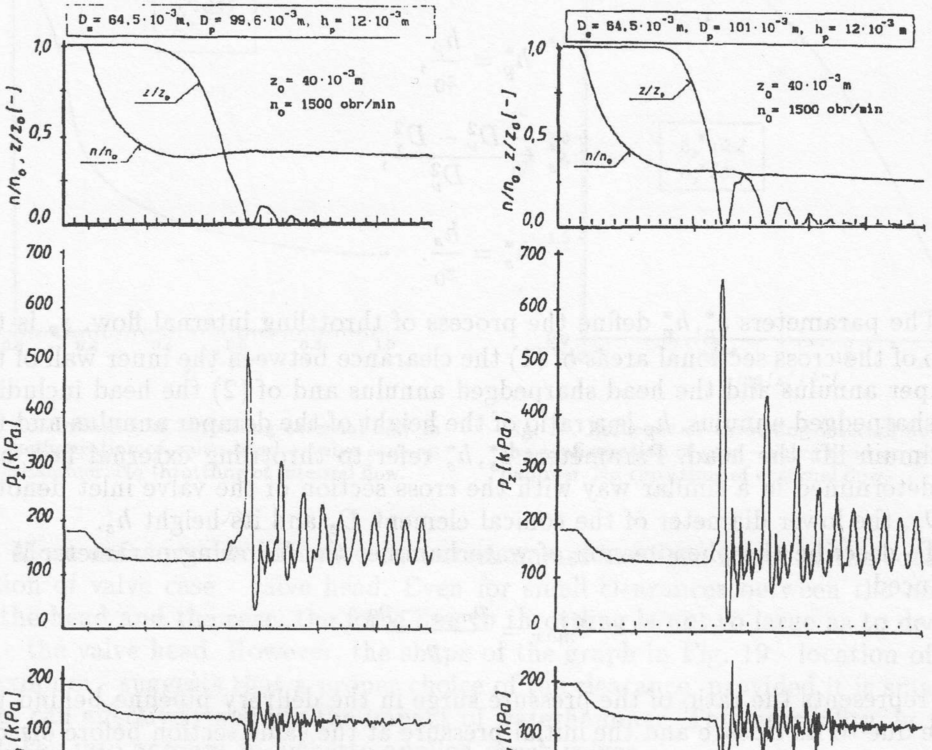

Fig. 17. Measured transients during pump stoppage. Influence of the height of the damper annulus in double damped valve.

made use of:

$$s_p^* = \frac{D_p^2 - D_t^2}{D_t^2},$$

$$h_p^* = \frac{h_p}{z_0},$$

$$s_s^* = \frac{D_\nu^2 - D_s^2}{D_\nu^2},$$

$$h_s^* = \frac{h_s}{z_0}.$$

The parameters s_p^*, h_p^* define the process of throttling internal flow. s_p is the ratio of the cross sectional areas of (1) the clearance between the inner wall of the damper annulus and the head sharpedged annulus and of (2) the head including the sharpedged annulus. h_p is a ratio of the height of the damper annulus and the maximum lift the head. Parameters s_s^*, h_s^* refer to throttling external flow and are determined in a similar way with the cross section of the valve inlet denoted as D_v, the lower diameter of the conical element D_s and its height h_s.

To describe the phenomenon of waterhammer the following parameter is introduced

$$\delta_{max} = \frac{p_{max} - p_0}{p_0}$$

and represents the ratio of the pressure surge in the delivery pipeline behind the valve due to its closure and the initial pressure at the same section before disconnection of the pump motor.

The influence of each individual damping method on waterhammer is presented in Figs. 18, 19. From Fig. 18, one can say that application of a conical element of the head without a damper leads to an unfavourable increase in pressure surge, compared to the undamped valve. The situation becomes more unfavourable with the clearance between the side conical surface and the valve seat becoming smaller. In the author's opinion the situation may be explained as follows:

1. From the static characteristics of the valve (Fig.6-11) it is evident that slightly lower flow coefficient C_Q at the open position of the valve with the conical element results in somewhat higher hydraulic thrust on the valve head, so that in consequence the valve closure begins with greater delay.

2. During final stages of valve closure under the conditions of reverse flow, the "reverse" hydraulic thrust aggravated by throttling internal flow leads to a precipitation of valve closure.

3. As a result, reverse flow gets higher rate and then higher deceleration when it rapidly stops.

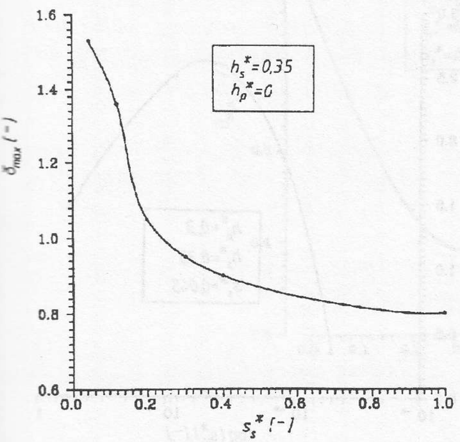

Fig. 18. Influence of throttling external flow in the configuration of valve head – valve seat on waterhammer. No throttling of internal flow.

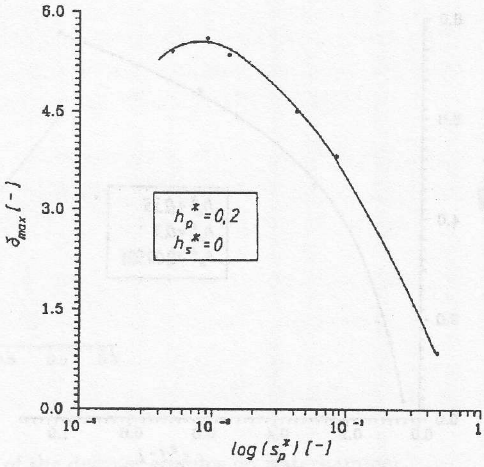

Fig. 19. Influence of throttling internal flow in the configuration of case – valve head on waterhammer. No throttling of external flow.

More unfavourable influence on waterhammer has been found in the configuration of valve case - valve head. Even for small clearances between the annuli of the head and the case, the force due to throttling is not so large as to decelerate the valve head. However, the shape of the graph in Fig. 19 - location of the maximum - suggests that a proper choice of the clearance, provided it is small in size, can effectively subdue the impact of waterhammer. This possibility, in fact, is taken into account in hitherto applied check valves.

Effects of simultaneous application of the two ways of throttling are displayed in Figs. 20-22. In Fig. 20, the influence of the conical element in the double damped valve on waterhammer is presented. One can see that, unlike in Fig. 18, pressure surge becomes smaller with diminishing the size of the clearance between the side conical surface and the valve seat. This positive impact of the conical element is a result of a lower flow rate of reverse flow due to the increasing intensity of throttling internal flow with simultaneous decelerating of the head.

The influence of the throttling configuration of case annulus-head sharpedged annulus in the double damped valve is illustrated in Figs. 21, 22. In Fig. 21, a favourable influence of the configuration can be reported for a much larger clearance between the annuli than in the case without a conical element, see Fig. 19. Further reducing the size of the clearance subdues the impact of waterhammer more efficiently. In Fig. 22, the waterhammer parameter is presented as a function of the height of the case annulus. The highest rate of reducing the intensity of waterhammer has been achieved during deceleration of the valve head beginning from 30% of its maximum lift.

From the obtained results, one can conclude that the proposed design of double damped plug-type check valve considerably subdues the impact of waterhammer,

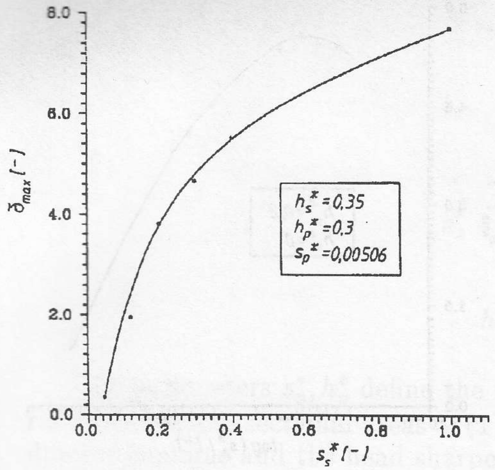

Fig 20. Double damped valve. Effect of the conical element of the valve head on waterhammer.

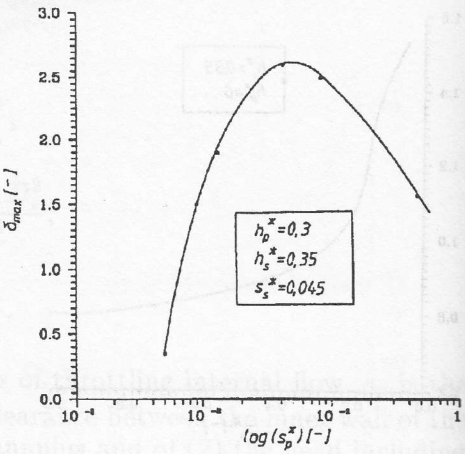

Fig 21. Double damped valve. Effect of the clearance in the damper on waterhammer.

compared to those damped check valves in which throttling devices are installed only to reduce the speed of valve closure in its final stages.

6. Conclusions

A damped plug-type check valve of original design has been constructed and tested in order to reduce the effect of waterhammer in pump systems. The novelty of the design lies in simultaneous application of two methods of throttling liquid flow within the valve. One of them takes place in the throttling clearance of the configuration of valve case-valve head in the clearance between the case annulus and the sharpedged annulus of the valve head. Like in check valves used hitherto, due to throttling internal flow in this clearance, decelerating of the valve head is achieved in the final stages of valve closure. The other method of throttling takes place in the configuration of valve head-valve seat by means of a conical element as the press-down surface of the head. This method leads to an increase in flow throttling when the valve is partially open.

Experimental investigations have been carried out on a pump test rig for different configurations of a specially prepared valve. Waterhammer has been a result of valve self-closing after disconnection of the pump motor. Application of one of the proposed methods of throttling (without application of the other) does not result in diminishing the impact of waterhammer which remains comparable to that of the undamped valve. Each conical element used in the investigations of the valve without throttling internal flow has led to an increase in pressure surge, compared to the valve with a flat press-down surface of the head. Similarly, for

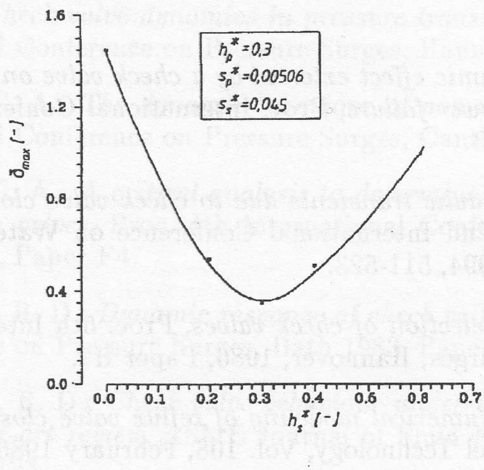

Fig. 22. Double damped valve. Effect of the height of the damper annulus on waterhammer.

each size of the clearance in the configuration of case annulus-head sharpedged annulus used in the investigations without a conical element, a much desired reduction of the impact of waterhammer has not been achieved. However, as the obtained results suggest, considerable reduction in size of this clearance can lead to reduction of pressure variations below those of the undamped valve.

Simultaneous throttling in both mentioned configurations significantly subdues the impact of waterhammer. The conical shape of the head's press-down surface has been positively verified in the valve with double throttling. As a result, for the smallest clearance between the case annulus and head sharpedged annulus and the largest conical element of the head applied, a considerable reduction of pressure variations has been found, compared to the undamped valve. This result is to confirm usefulness of the new construction of the plug-type check valve in diminishing the level of waterhammer in pump systems after pump power failure. Towards the end of the paper it should be underlined that there is still a need of a calculation method to be worked out with a view to appropriately selecting the parameters of the two presented throttling configurations. The results of experimental investigations could be then serviceable in validation of this calculation method.

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Badania doświadczalne wpływu układów dławieniowych wzniosowego zaworu zwrotnego na przebieg uderzenia hydraulicznego w przewodzie tłocznym pompy

Streszczenie

Opracowano i zbadano wzniosowy zawór zwrotny oryginalnej konstrukcji w celu zmniejszania poziomu uderzenia hydraulicznego w układach pompowych. Oryginalność rozwiązania polega na równoczesnym zastosowaniu dwóch sposobów dławienia przepływu cieczy w zaworze. Jeden z nich jest realizowany w szczelinie tłumika utworzonej w układzie: pierścień obudowy-ostrokrawędziowy pierścień elementu zamykającego (grzybka) zaworu. Drugi sposób jest realizowany w układzie: grzybek-gniazdo zaworu za pomocą ukształtowanego grzybka.

Badania doświadczalne przeprowadzono na stanowisku pompowym dla różnych konfiguracji specjalnie przygotowanego zaworu. Uderzenie hydrauliczne było wynikiem samoczynnego zamknięcia zaworu po odłączeniu napędu pompy. Poprzez pojedyncze zastosowanie proponowanych sposobów nie uzyskano korzystnego efektu łagodzenia uderzenia hydraulicznego w porównaniu do zaworu bez tłumienia. Równoczesne zastosowanie dwóch zaproponowanych sposobów znacznie zwiększyło efekt łagodzenia uderzenia hydraulicznego.

W rezultacie potwierdzono założoną przydatność opracowanego zaworu do zmniejszania poziomu uderzenia hydraulicznego w układzie pompowym w warunkach odłączenia napędu pompy.