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DYNAMIC MODEL FOR SIMULATION OF CHECK VALVES IN PIPE SYSTEMS

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Abstract

In the planning phase of pipe systems, the behaviour of pipe systems under irregular operating conditions should be controlled. This checking process can be made with the aid of computer programs. These computer programs have numerical models for each element (pipe, pump, different valves, air vessel, etc.) of the pipe system. Check valves are pipe accessories that can prevent back-flows in the system after the shut-down of the pump. This paper presents a mathematical model, which can be applied to simulate the behaviour of check valves, not only in case of full closure, but also at changes of flow conditions in the pipe. The research in this field is sponsored by the fund OTKA (OTKA T029073).

Keywords: check valve, transient flow, numerical simulation.

1. Introduction

In the planning phase and later during the regular operation of pipe systems, the transient processes are the most dangerous events for the system. These processes should be checked numerically. Effects of extreme operating conditions of the pipe system in the planning phase are simulated with the help of computer programs. During regular operation, the comparison of measured and calculated values of both pressure and flow rate could reveal failures of the system. A computer code, which can simulate the processes in a pipe system, must have numerical-mathematical models of each pipe element.

Check valves are self-acting valves, which allow flow only in one direction. They are built into the system to prevent back-flows after shut-down of pumps. Imperfect operation of check valves can cause high pressure peaks, vibrations or even breaks in the pipe systems. Most of the mathematical models of check valves were developed for the worst case, specially for the full stop of flow in the pipe. PROVOOST [1] developed a dynamic model of check valves for full closure. Sometimes the change between two stationary operation points of the pipe system results in pressure peaks. In some systems more check valves are built in serially. In this case a bad model can produce self-excited oscillations, while the real system operates well. It is therefore necessary to have a numerical model, which simulates not only the closing process of check valves but also the operation point changes. Such models have been developed by FŰZY [2] and CSEMNICZKY [3] (based on the behaviour in steady flow). It is known by experience of numerical calculations, that this model is not correct. It tends to oscillations and instabilities of the numerical computations. A better model has been developed by comparing experiments and numerical computations of the same system.

2. Steady Flow Behaviour

The behaviour of check valves in steady flow is properly described by their characteristics. These characteristics show how the pressure drop and the hydraulic opening torque depend on the opening angle of the check valve. They are measured in steady flow. The drag coefficient (ζ) gives the relationship between pressure drop (Δp) and flow velocity (v) for a given opening angle (φ):

$$\Delta p = \zeta\ (\varphi)\, \frac{\rho}{2} v^2. \tag{1}$$

Because the drag coefficient changes from zero to infinity, this interval is transformed to interval $[0 - 1]$. The parameter after substituting ζ of check valves is the drag number ($K_\zeta\ (\varphi)$):

$$\zeta\ (\varphi) = \left(\frac{1}{K_\zeta\ (\varphi)} - 1 \right)^2 . \tag{2}$$

The other parameter is the coefficient of hydraulic torque ($K_M\ (\varphi)$), which gives the dependence of hydraulic opening torque (M_H) on the pressure drop for a check valve with nominal diameter D:

$$M_H = K_M\ (\varphi) \cdot \Delta p \cdot D^3. \tag{3}$$

In a steady flow the torque equation is the following:

$$M_H\ (\varphi, \Delta p) + M_c\ (\varphi) + M_T\ (\varphi) = 0, \tag{4}$$

where M_c is the torque caused by the eccentricity of the mass centre of valve disc and M_T is the torque of the mass load (m) on the lead-out rotation shaft. These are given by:

$$M_C = -K_C(\varphi) \cdot m_0 \cdot g \cdot D, \tag{5}$$
$$M_T = -k \cdot m \cdot g. \tag{6}$$

These dimensionless characteristics of operation in steady flow are the only characteristic values given by the check valve producers. These characteristics could be used to determine the operating point of the check valve after a transient process.

3. Earlier Dynamic Model

The models for the transient behaviour of check valves of FŰZY and CSEMNICZKY are based on the characteristics in steady flow. The dynamic equation of the valve disc is given below:

$$\ddot{\varphi} = \frac{M}{\Theta}, \tag{7}$$

where M is the sum of the torques acting on the valve shaft (*Fig.1*), Θ is the moment of inertia of the rotating parts including the moment of inertia of the mass load. For a moving disc there are new torque components:

$$M = M_c(\varphi) + M_T + M_H(\varphi, \Delta p) + M_b(\dot{\varphi}) + M_\omega(\dot{\varphi}) . \tag{8}$$

New torque components are the braking torque M_ω (of streaming fluid acting against the rotation of the disc), and the braking torque of the oil-damper M_b. These torque components are given by the following formulae:

$$M_\omega = -K_\omega \cdot m_0 \cdot D^2 |\dot{\varphi}| \dot{\varphi}, \tag{9}$$

$$M_b = \begin{cases} K_b(\varphi) \cdot \dot{\varphi}^2 & \dot{\varphi} < 0 \\ 0 & \dot{\varphi} \geq 0. \end{cases} \tag{10}$$

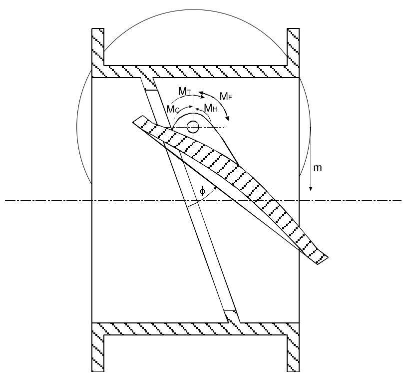

Fig. 1. Torques on the check valve disc

In these equations m_0 is the mass of the valve disc, K_ω the coefficient of braking torque (see CSEMNICZKY [3]) and $K_b(\varphi)$ the opening angle dependent dimensional damper coefficient.

4. Experimental Study of the Model

At the Department of Hydraulic Machines of the Budapest University of Technology and Economics a modular computer program has been developed to calculate transient processes in pipe systems. This program contains the check valve model with the above mechanical equations. In the laboratory of the Department a measuring equipment has been built for transient investigation of check valves (*Fig.*2). It is possible to generate two types of sudden changes. One of them is a jump in the mass load, the other one is a sudden change of flow rate by opening a bypass. Sudden opening of the bypass can be accomplished by piercing a foil at the port of a T-junction [4].

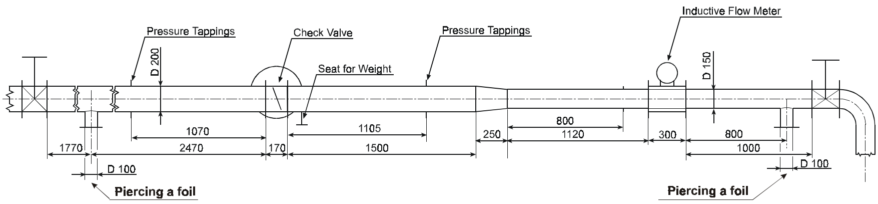

Fig. 2. Measuring equipment

During our experiments we have detected that there is a high friction at the rotation shaft. The experiments show high damping but the simulation with the original model gives oscillation of the disk valve (*Fig.*3). The torque of this friction was not included in the mechanical model.

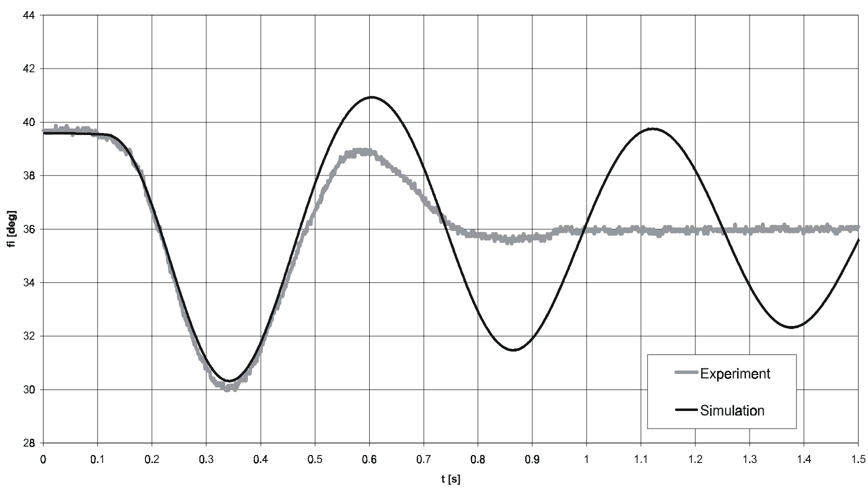

Fig. 3. Change of opening angle, piercing a foil before the check valve

5. Friction at the Rotation Shaft

The friction at the rotation shaft is modelled by Coulomb's law. There are two different cases, friction at rest and friction in motion. The friction at rest stabilises the position of the valve disk. When the sum of other torque components is less than the moment of friction at rest, there will not be a change of the disk position. The dynamic motion equation is modified with the friction:

$$\Theta \frac{\mathrm{d}^2\varphi}{\mathrm{d}t^2} = \begin{cases} 0 & |M| < M_{F_0\,\max} \\ M - \operatorname{sign}(\dot{\varphi}) \cdot M_F & \text{otherwise.} \end{cases} \tag{11}$$

M_{F_0} is the torque of friction at rest, M_F is the torque of the friction in motion. M is the sum of the other torque components. The torques of friction can be given by the following formulae:

$$M_{F_0\,\max} = \mu_0 F_N \frac{d_s}{2} \tag{12}$$

and

$$M_F = \mu F_N \frac{d_s}{2}, \tag{13}$$

where d_s is the diameter of the sliding bearings, μ and μ_0 are the coefficients of friction in motion and at rest, respectively. We used the numerical values of coefficients corresponding to the bearing's materials. F_N is the normal force at the bearing. The normal force is defined by the following formula:

$$F_N = \sqrt{[(m + m_\Theta)\, g + \Delta p \cdot A \cdot \sin(\varphi + \alpha)]^2 + [\Delta p \cdot A \cdot \cos(\varphi + \alpha)]^2}. \tag{14}$$

In this formula m_Θ and m are the masses of rotating parts and that of the mass load respectively, A the area of the valve disc and α the angle of the valve disk at fully closed state relative from vertical.

Considering friction improved our model, but it still does not describe properly the behaviour of the real check valve [6].

6. Steady Flow Characteristics of the Check Valve

It must be checked whether the model according to *Eqs*. (1)–(3) can be used in transient operation, too. These characteristics were determined by stationary streamlines. During the change of the opening angle of the valve the streamlines also change. The reason why these characteristics are used is that these are the only known characteristics of the check valve. In *Eq*. (1) the incident velocity should be the relative velocity of the fluid and the valve disc. Consider (*Fig.4*), that when the disc opens the part over the shaft moves (v_1) against the flow, the part below the shaft moves (v_2) in the direction of the flow. The relative velocities are:

$$\mathbf{v_{rel_1}} = \mathbf{v} + \mathbf{r_1} \times \dot{\varphi} \tag{15}$$

and

$$\mathbf{v_{rel_2}} = \mathbf{v} - \mathbf{r_2} \times \dot{\varphi}. \tag{16}$$

r_1 and r_2 are the distances of the centres of mass of the two parts of the disc.

With these relative velocities the renewed dynamic formula of the hydraulic torque can be defined:

$$M_h = K_M(\varphi) \cdot D^3 \cdot \zeta(\varphi) \frac{\rho}{2} \frac{1}{A} \left(A_1 v_{\mathrm{rel}_1}^2 + A_2 v_{\mathrm{rel}_2}^2 \right). \tag{17}$$

Here A is the area of the valve disc, A_1 and A_2 are the areas of the valve disc parts. The formula gives the steady state value in case of steady flow. This formula enables the more exact simulation with the use of stationary characteristics.

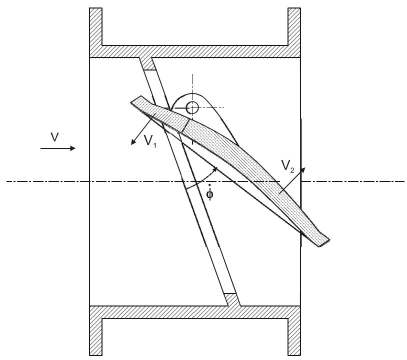

Fig. 4. Relative velocities

7. Modification of the Moment of Inertia

The reduced moment of inertia of the rotating parts of the check valve consists of two parts. The dynamic equation should include the acceleration of the valve disc and that of the mass load. The revised equation is:

$$\Theta_0 \ddot{\varphi} + m \cdot k \cdot a = M, \tag{18}$$

where Θ_0 is the moment of inertia of the valve disc and of rotating parts, m is the mass load and 'a' is the acceleration of the mass load during the opening or closing process. There is a kinematic coupling between the acceleration of the mass load and the angular acceleration. If the arm of the mass load is k, this is given by the formula:

$$a = k \cdot \ddot{\varphi}. \tag{19}$$

From *Eqs*. (18) and (19) the reduced moment of inertia can be defined as:

$$\Theta_r = \Theta_0 + m \cdot k^2. \tag{20}$$

When a disc rotates in a viscous fluid, it takes some fluid with it. This fluid increases the accelerated mass. Therefore it also increases the moment of inertia of the rotating parts. It should be recommended to increase the moment of inertia to consider that of the accelerated fluid by some fraction. The valve disc has an asymmetrical form. Therefore it transports a varying volume of fluid depending on the rotation direction. The added mass and the increase of the moment of inertia also depends on the rotation direction (*Fig. 5*). By introducing the mass raising factor f, the moment of inertia of the whole system is:

$$\Theta = (1 + f) \cdot \Theta_0 + m \cdot k^2. \tag{21}$$

The values of factor f which proved to be satisfactory in our simulation tests were 0.1 and 0.3 respectively in the positive and negative acceleration of the valve disc.

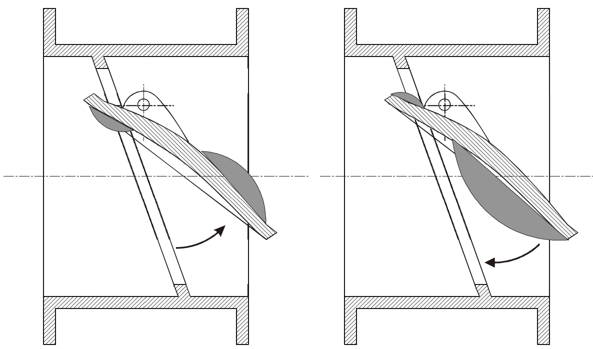

Fig. 5. The raise of the moment of inertia by attached fluid for both directions of rotation

8. The Results of the New Model

The program simulating transient processes in pipe systems has been developed at the Department of Hydraulic Machines. The modular structure of the program enables easy change of the models of the different pipe accessories. With the change of the original check valve model, simulations were executed. We have investigated rapid changes in the system generated by step functions in the mass load or in the flow rate through the check valve. The rapid change of mass load is a relatively slow

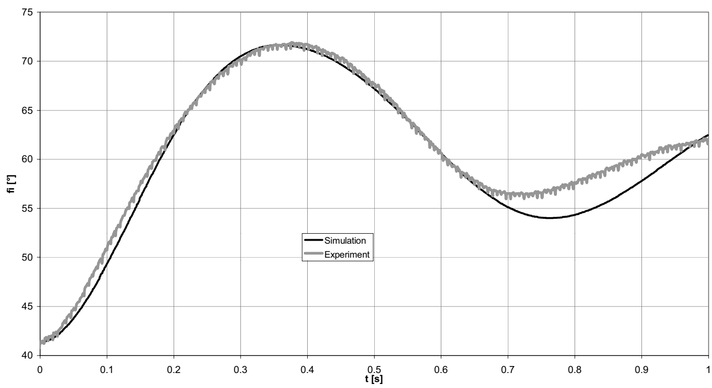

Fig. 6. Change of opening angle after cutting off mass load

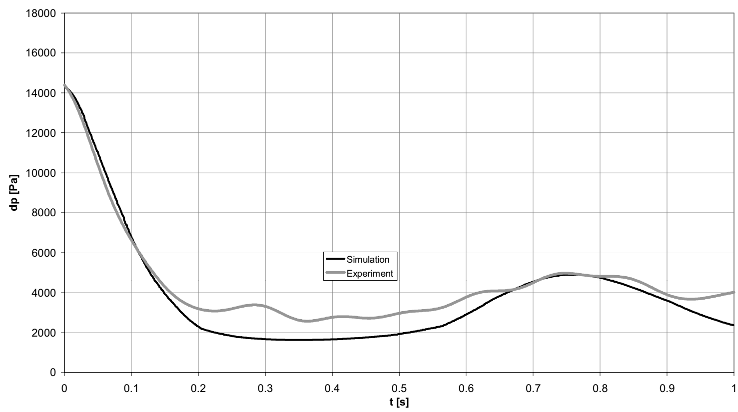

Fig. 7. Change of pressure drop after cutting off mass load

process. In this case the change is transmitted to the system over the inertial mass of the rotating parts. A comparison of the results of measurements and numerical simulations is presented in *Figs.* 6 and 7. *Fig.* 6 shows the change of the opening angle and *Fig.* 7 the pressure drop through the check valve over the time. Cutting off the mass has occurred at 0 s. The working point of the pump does not change much because the pressure drop of the check valve is about 6% of that of the whole measuring equipment.

In *Figs.* 8, 9, and 10 time histories for piercing a foil are presented. Piercing the foil behind the check valve happened at $t = 0.15$ s. *Fig.* 8 shows the change of the opening angle, *Fig.* 9 and 10 show the pressure at the pressure tapping before and behind the check valve. This change acts directly on the hydraulic system, there is a big change in the point of operation of the pump. There is a difference of measured

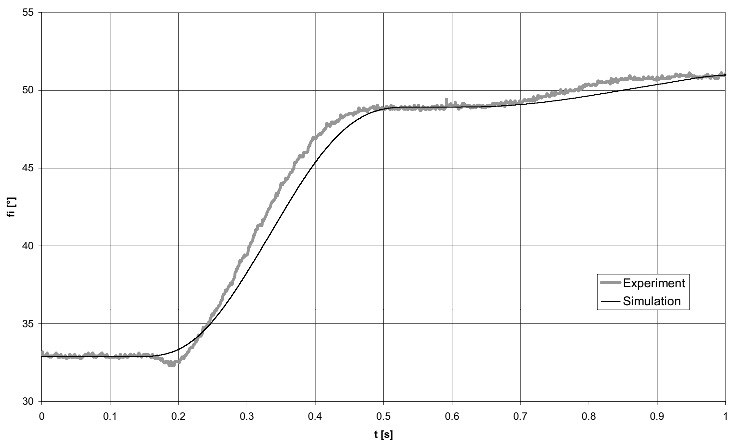

Fig. 8. Change of opening angle after piercing a foil

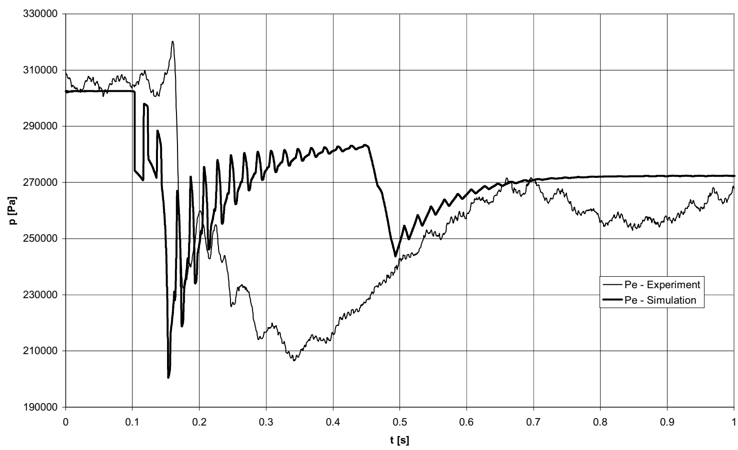

Fig. 9. Change of pressure before the check valve after piercing a foil

and computed pressures. The reason for this difference is that the shortages of mathematical models of other components of the complete investigated system (pump, valves, elastic pipes, etc.) influence the numerical results too.

Comparing measurements with simulations of the same changes prove that the developed model can simulate transient processes in pipe systems containing check valves with acceptable accuracy. Our model must also be tested with results from measurement of other systems.

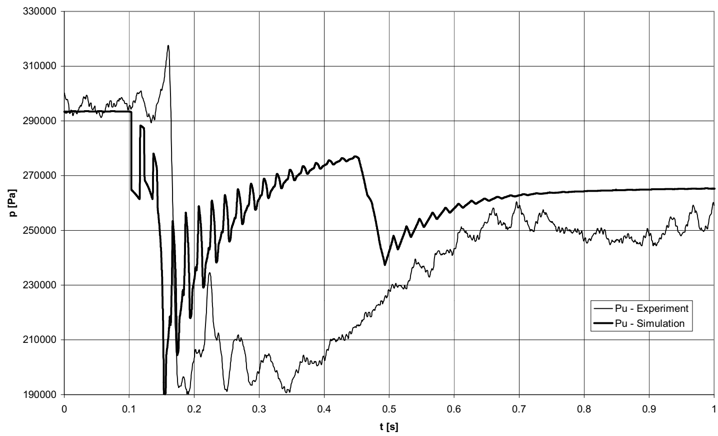

Fig. 10. Change of pressure behind the check valve after piercing a foil

9. Conclusions

On the basis of the old dynamical model of check valves by CSEMNICZKY and FŰZY we have developed a new model by correcting the motion equation of the valve disk. Considering the friction on the shaft, the rotation direction dependency of the moment of inertia of the attached fluid and the relative incidence velocity in the hydraulic torque the simulation results of the improved model are more realistic than those of the earlier one. By improving the numerical models of the other pipe components the simulation can be even more exact.

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