EFFECTIVE PROTECTION OF LONG PIPELINES BY TRANSIENT PROCESS THROUGH MEMBRANE, START-GIVING VALVES AND FEEDER TANKS IN PUMPING SUPPLY SYSTEMS

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Abstract

The long pipelines have usually a length over 1000 m to several thousand meters. In that case a big direct water hammer arises during the transient process in pumping supply systems. Besides a dangerous interruptions of the water column could be observed. This circumstance compels the specialists to escape the long pipelines and to resort to a costly decision by pumping cascades with multiple pumping units. A new protection is proposed by set of membrane valve, start-giving valve and feeder tanks. This protection is more effective, because firstly it leads to construction of pipelines with small thickness and secondly it allows only one pumping station without appearance of cavitation along the pipeline. A practical implementation of this solution brings to an optimal economical decision and to low water hammer head provided the pipeline is tracing closed to the ground. A set of feeder tanks and air vessels are mounted lengthwise of the pipeline on the peak bends. The result of operation shows that the feeder tanks supply the system by water and don't permit the creation of cavitation zones. The realization was made by authors' software product by means of Object-Oriented Tools (OOT) for transient flow analysis. The set was tested in laboratory and was investigated by field study during the operation of a real water supply pumping station. The numerical results were compared with "Kistler" transducer system.

Keywords: Transient process; Membrane valve; Start-giving valve; Feeder tank; Back-pressure valve; Cavitation; Water supply; Object-Oriented Tools; Air vessel;Start-giving head

1. INTRODUCTION

The protection consists of a membrane valve (MV) and a start-giving valve (SGV) on a branch of the pipeline – Fig. 1. With each pump a back-pressure valve (bpv) is included (Fig. 2). Feeder tanks (FT) with air vessels (AV) are set on the peak bends of the pipeline. A control valve (cv) is mounted before the branch and elevation of upper level (EUL), resp. elevation of low level (ELL) are shown. The transient process includes five stages as follows:

Stage 1. The flow is directed to the upper reservoir. At the same time the pump revolutions and the head are reduced. In the end of this stage the MV begins to flow and the head decreases till the value predicted by SGV (S_1).

Stage 2. Generally the minimum of the headline had been reached before the moment when unsteady flow arrived at the upper reservoir. The MV is open and the feeder tanks support the piezometric line high at the dangerous peak pipeline points (S_2).

Stage 3. The flow in the pipeline is directed partial to the MV(velocity is negative). One part of unsteady flow with positive velocity takes up the space of the pipeline and another one is occupied by steady motion. The MV continues to be opening while the SGV is closing till the limit of start-giving head (SGH). The piezometric line is raising slowly (S_3).

Stage 4. When the full length of the pipeline is spread all over by unsteady motion, the two valves are open completely. This state will continue to the total closing of SGV (S_4).

Stage 5. A new filling of membrane chamber begins depending on the size of diaphragm (S_5).

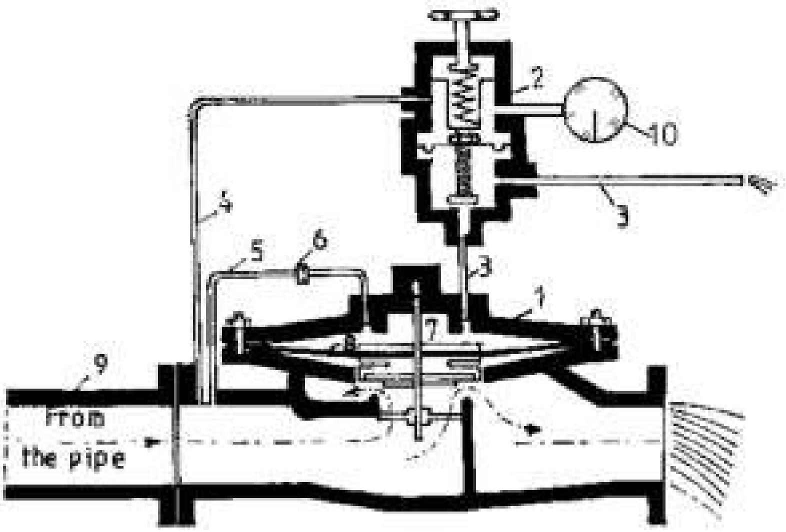

Fig. 1 Membrane valve MV; 2.Start-giving valve SGV; 3.Evacuation tube; 4.Start giving tube; 5.Filling tube; 6.Diaphragm; 7.Membrane chamber MCh; 8.Membrane; 9.Branch;10. Manometer

2. DEVELOPMENT OF NUMERICAL MODEL

The proposed algorithm will be considered having in minded a compound pipeline and the main equations of unsteady motion. The different hydraulic structures can be coupled together and connected with one object, i.e. the closing of one valve determines the operation of another one and so on. Thus the *Object Oriented Tools* (OOT) could be presented by means of nodes, branches, grid points and connections associated with one common structure, P. Ingeduld at al. (1996). This principle is discussed further bearing the above five stages.

2.1 REDUCTION OF PUMP HEAD TO THE PREDICTED VALUE BY SGV

The integration of the main equations of unsteady motion is carried out by Euler's method along characteristic direction - N.Nikolov, M.Maradjieva (2003):

$$y_i = y_0 + \frac{a_1}{g}\left(v_i - v_{i-1}\right) + \frac{a_1}{g}\sum_{j=i-1}^{i-\mathrm{int}_1+1}\left(v_j - v_{j-1}\right) + \frac{1}{g}\left(\Delta p_1^e a_1 + \Delta p_2^0 a_2\right)\left(v_{i-\mathrm{int}_1+1} - \frac{d_1^2}{d_2^2}v_{i-\mathrm{int}_1}\right) +$$

$$+\sum_{m=2}^{s}\left\{\frac{a_m}{g}\frac{d_1^2}{d_m^2}\left[v_{i-\sum_{m=1}^{i-1}\mathrm{int}_m} - v_{i-\sum_{m=1}^{i-1}\mathrm{int}_m-1}\right] + \frac{a_m}{g}\frac{d_1^2}{d_m^2}\sum_{j=i-\sum_{m=1}^{i}\mathrm{int}_m-1}^{i-\sum_{m=1}^{i-3}\mathrm{int}_m+2}\left(v_j - v_{j-1}\right) + \right.$$

$$\left. + \frac{1}{g}\left(\Delta p_m^e a_m + \Delta p_{m+1}^0 a_{m+1}\right)\left[\frac{d_1^2}{d_m^2}v_{i-\sum_{m=1}^{i-2}\mathrm{int}_m+1} - \frac{d_1^2}{d_{m+1}^2}v_{i-\sum_{m=1}^{i-1}\mathrm{int}_m}\right]\right\} + \frac{\Delta l_1}{C_1^2 R_1^1}\frac{v_i^2 + v_{i-1}^2}{2}sign\left(v_i\right) +$$

$$+\frac{\Delta l_1}{C_1^2 R_1}\sum_{j=i-1}^{i-int_1+2}\frac{v_j^2 + v_{j-1}^2}{2}sign\left(v_j\right) + \frac{\Delta l_1^e}{C_1^2 R_1}\frac{v_{i-\mathrm{int}_1+1}^2 + \left(v_{i-\mathrm{int}_1+1} + \Delta v_1^e\right)^2}{2}sign\left(v_{i-\mathrm{int}_1+1}\right) +$$

$$+\underbrace{friction unsteady\ losses\ of\ other\ sections} + ..\underbrace{friction steady\ losses\ of\ all\ sections} \qquad (1)$$

where $a = \sqrt{\dfrac{g/\gamma}{1/\varepsilon + d/\mathrm{E}\delta}}$ is the wave velocity, g the gravity acceleration, γ the volume weight of the fluid, R_1^1 the hydraulic radius, accounting a correction by minor-loss coefficient of cv, R_1, R_m denote hydraulic radiuses of the first and the random section *m*, C_1 Chezy's coefficient of the first section, ε, E the modulus of elasticity of the fluid and the pipe material, δ, d the pipe wall thicknesses and diameters for a given section, $\Delta l_1, \Delta l_1^e$ the integer step of length for first section and the remainder for first section respectively, *m, s* the current number of the section and the number of last section resp., $int_1^{'}$ the integer number of steps for the first section, $int_1 = int_1^{'} + 1$ the integer number of points for first section, $\Delta t = T_{prop} / p$ the step in time for total wave propagation, $p, \Delta p_m^e, \Delta p_{m+1}^0$ denote the number of integer step for time, the last and initial remainders resp., *i* the index for time, *k* the index for place as follows from Fig. 2.

The initial conditions are introduced in the following form:

$$y(t=0) = y_0 + h_{loss_0}, \quad v(t=0) = v_0 \qquad (2)$$

The head for an arbitrary cross section indicated as "k" can be defined by Euler-MacLauren method, N. Nikolov, M. Maradjieva (2003), using the following formula:

$$p_i^k / \gamma + \Delta H_m^k = y_i^k \qquad (3)$$

where *i* is the index for time, *k* the index for place, *m* is the index for a given section (Fig. 2), $p_i^k / \gamma = h_m^k$.

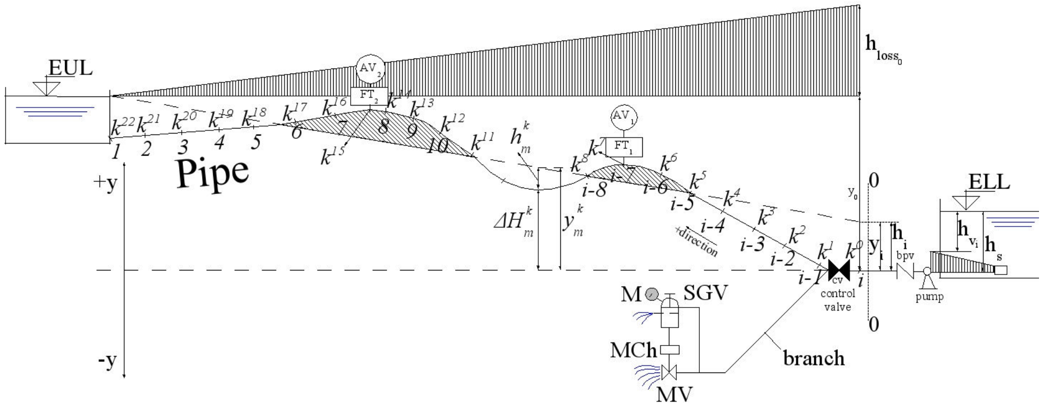

Fig. 2 Scheme of pumping station with MV and SGV on branch, AV on feeder tanks and BPV

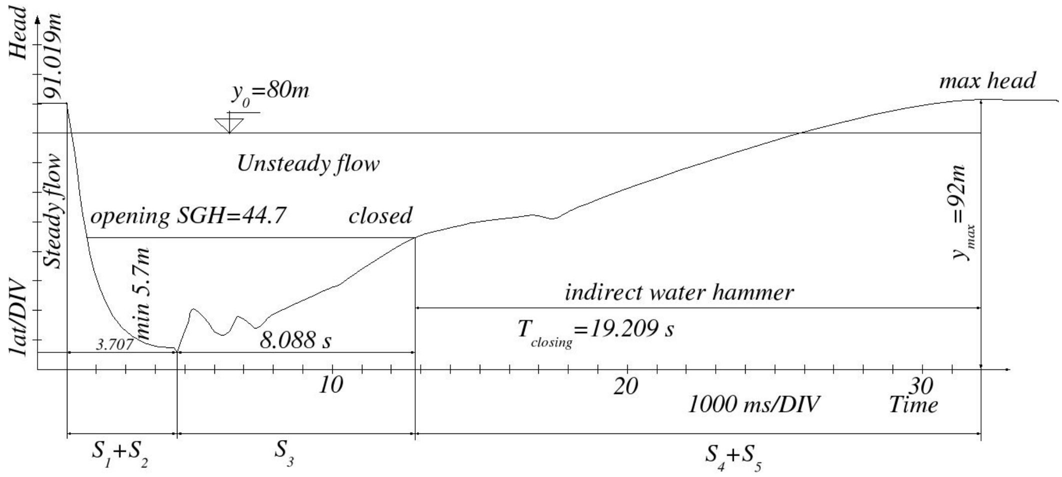

Fig. 3 Results obtained by numerical experiment

2.1.1 Boundary conditions

The boundary condition in the first section is $y_0 = const$ and in 0-0 is described by pump characteristics in I or IV quadrant approximated with cubic splines in the Hermite's form. The pump capacity is calculated at a moment $t_i = i\Delta t$ through the following equations:

$$H_{0i} \geq 0 \quad N_{0i} = \frac{9.81 Q_{0i} H_{0i}}{\eta_p \eta_m^{mech}} \tag{4}$$

If $\Delta H_i \rangle H_{0i}$ it is accepted $H_{0i} = \Delta H_i$ and for $H_{0i} < 0$, follows:

$$N_{0i} = \frac{9.81 Q_{0i} \left(H_{0i} + \Delta H_i \right)}{\eta_p \eta_m^{mech}} \tag{5}$$

where η_p the efficiency of the pump, η_m^{mech} the mechanical efficiency of the motor; H_{0i} and Q_{0i} are the head and pump discharge for the rated revolutions n_0. The losses ΔH_i are:

$$\Delta H_i = A n_0^2 + B n_0 Q_{0i} + H_{0i} \tag{6}$$

The coefficients A,B are determined by two points of the rating curves with the highest efficiencies η_i. The following relationships are used for the revolutions:

$$n_i \geq 0.3n_0 \quad n_i = \frac{k_i}{k_i / n_{i-1} + \Delta t} \tag{7}$$

$$n_i < 0.3n_0 \quad n_i = n_{i-1} - \frac{\Delta t}{k_i}(0.3n_i)^2 \tag{8}$$

where $k_i = \dfrac{n_0^3 [GD^2]}{365000\overline{N_{0i}}}$, $\overline{N_{0i}} = \dfrac{N_{0i-1} + N_{oi}}{2}$,[GD^2] is the rotary moment of the aggregate and the units in SI are given in the paper N. Nikolov, M. Maradjieva (2003).

The head H_i, the velocity v_i^s and the discharge Q_i^p of the pump at the moment $t_i = i\Delta t$ and random revolutions n_i are determined by the similarity laws for *N* number of pumps:

$$H_i = H_{0i}\left(\frac{n_i}{n_0}\right)^2, v_i^s = Q_i^p / NF_{pipe}, \quad Q_i^p = NQ_{0i}\left(\frac{n_i}{n_0}\right) \tag{9}$$

The head h_i of the rotary pump with suction is calculated by the formulas:

- submerged pump: $h_{vi} = h_s - h_{loss\,i} \qquad h_i = H_i + h_{vi}$ (10)
- unsubmerged pump: $h_{vi} = h_s + h_{loss\,i} \qquad h_i = H_i - h_{vi}$ (11)

where h_{vi} is the vacuum height of the pump, h_s is the suction height (Fig. 2).

An iterative simulation related to discharge Q_{0i} is used to achieve the condition for accuracy.

2.2 THE HEAD UNDER THE SGH TO THE MINIMUM ONE IN TWO CASES

The opening, resp. the closing of SGV is completed for an equal value of the head (SGH). After opening of the SGV the emptying of membrane chamber begins through the evacuating tube. Meanwhile the opening of MV begins and a flowing process in the atmosphere follows. After spreading of unsteady flow to the corresponding feeder tank the same begins to supply the water. Two cases are observed:

Case 1. When for two consecutive time step the head is decreasing $H_{oi} < H_{oi-1}$ and $h_i < h_{i-1}$, the velocity is growing up by modulus $v_i > v_{i-1}$. Then the closing of BPV follows. In that case the velocity in the pipe is negative ($v_i < 0$) and the flow discharge is equal of the discharge of MV. Further the head y_i is reducing to the minimum one.

Case 2. The closing of the BPV may be happening if the discharge of the pump is less than the discharge of MV: $Q_i^p < Q_i^{MV}$. This case is a little bit unstable and after some oscillations the BPV is shut off. The problem is solved by iterations according the following four stages:

a) An iterative procedure related to the discharge of the pump is performed in order to achieve the required condition for accuracy. The system includes the pump capacity - equation (4) or (5), the revolutions' equation (7) or (8) and equations (9) by similarity laws.

b) Assessment of the discharge of MV.

The following equations are valued:

$$Q_i^{MCh} = \mu_i F_{ev}\sqrt{2gh_i} \text{ and } \mu_i = 1 \Big/ \sqrt{\xi_{ex} + 2\xi_{en} + \xi_{SGV} + \xi_f^{ev}} \tag{12}$$

where ξ_{ex} the local loss coefficient of exit, ξ_{en} the local coefficients for entrances, $\xi_f^{ev} = \lambda l / d_{ev}$ the friction coefficient of evacuation tube, F_{ev} the section of evacuating tube.

The coefficients μ_{SGV}, ξ_{SGV} of SGV are defined in Table 1 depending on the relative strain of the spring and the size of MV, namely

$$\Delta p_{SGV} = (h_{SGV} - h_i)/h_{SGV} \le 1 \tag{13}$$

Table 1. Coefficients of losses for SGV with size: $F_{SGV}/F_{ev} = 1.0133$, $MV\ Ø150$

Δp_{SGV}	μ_{SGV}	ξ_{SGV}	Δp_{SGV}	μ_{SGV}	ξ_{SGV}
0.01	0.0097	10659	0.28	0.2216	20.37
0.02	0.0054	1861	0.32	0.2427	16.97
0.03	0.0322	964	0.36	0.2613	14.65
0.04	0.0375	712	0.44	0.3041	10.81
0.08	0.0798	156.9	0.548	0.3518	8.08
0.12	0.1056	89.2	0.628	0.3772	7.03
0.16	0.1406	50.4	0.748	0.3863	6.70
0.20	0.1664	36.1	0.868	0.3965	6.36
0.24	0.1915	27.27	1.0	0.4082	6.00

The coefficients of losses for specified MV are given in Table 2.

Table 2. Coefficients of losses for MV$\varnothing$150, $\Delta = 70.6 ml/mm$, $\delta_{\max} = 38mm$

Δh	μ_{MV}	ξ_{MV}	Δh	μ_{MV}	ξ_{MV}
0.0066	0.0075	17777	0.2105	0.1733	33.3
0.0132	0,0152	4339	0.2368	0.1892	27.94
0.0197	0.0229	1899	0.2895	0.2230	20.10
0.0263	0.0308	1051	0.3605	0.2615	14.62
0.0526	0.0501	398	0.4132	0.2917	11.75
0.0789	0.0794	159	0.4921	0.3358	8.87
0.1316	0.1200	69.5	0.5711	0.3585	7.78
0.1579	0.1396	51.3	0.6157	0.3764	7.06
0.1842	0.1575	40.3	1.0000	0.4082	6.00

The two tables shown above are extended further for MV with various sizes: Ø50, Ø65, Ø100, Ø150, Ø200, Ø250 mm. By means of experiments given in Table 1, Table 2 the size of inside opening of MV, the volume W_i, the relative steps δ_w and Δh of MV can be determined:

$$W_i = \frac{Q_i^{MCh} + Q_{i-1}^{MCh}}{2} \Delta t,\ \sum W = \sum W_{i-1} + W_i,\ \delta_w = \frac{\sum W_i}{\Delta} \frac{ml}{ml/mm} \tag{14}$$

$$\Delta h = \delta_w / \delta_{\max} \to \mu_{MV} \to Q_i^{MV} = \mu_{MV} F_{MV} \sqrt{2gh_i} \tag{15}$$

c) Calculation of the pipe discharge Q_i from the equation of continuity :

$$Q_i = Q_i^p - Q_i^{MV} \tag{16}$$

If the values of the head from (1), (10) or (11) are equal and (16) is satisfied, the next time step can be executed. The calculations stop when the minimum head is achieved.

d) Calculation of the necessary volume of the feeder tanks.

Firstly the head in the places of the feeder tanks is defined by (3). After that the head is compared with the elevation of the corresponding feeder tank and if $y_i^k \langle \Delta H_m^k$ is satisfied the vacuum would appear. Then the feeder tank will supply the necessary water volume by AV:

$$W_i^k = \Delta v_i^k . \Delta t . \overline{F_m} . \sigma \tag{17}$$

$$\Delta v_i^k = \frac{g\left|y_i^k - \Delta H_k^m\right|}{\overline{a_m}},\ \overline{a_m} = \frac{a_m + a_{m+1}}{2},\ \sum W_i^k = \sum W_{i-1}^k + W_i^k,\overline{F_m} = \left(F_m + F_{m+1}\right)/2 \tag{18}$$

$$\sigma = \frac{\left|y_i^k - \Delta H_k^m\right|}{\Delta t}\left(\frac{1}{a_m tg\alpha_m} + \frac{1}{a_{m+1} tg\alpha_{m+1}}\right) \tag{19}$$

where $tg\alpha_m, tg\alpha_{m+1}$ the slope gradient of the peak bend points.

The AV with sizes Ø80,150,250 mm could push in and let off air with capacity 140, 600 and 2000 m³/hr in the feeder tanks. The inertial head in a given k section at i- time step is calculated by:

$$h_{in_i}^k = \frac{1}{g}\left(\Delta p_m^e \frac{d_m^2}{d_{m+1}^2} a_m + \Delta p_{m+1}^0 \frac{d_{m+1}^2}{d_m^2} a_{m+1}\right)\left(\frac{d_1^2}{d_m^2} v_{i-\sum_{m=1}^{m-2} \mathrm{int}_m + 1} - \frac{d_1^2}{d_{m+1}^2} v_{i-\sum_{m=1}^{m-1} \mathrm{int}_m}\right) \tag{20}$$

2.3 INCREASE OF THE HEAD FROM $h_{\min}$ TO THE TOTAL SPREADING OF UNSTEADY FLOW ALONG THE PIPE

The operation of the set is controlled by the following conditions:

- Condition for raising of the head
- Condition for elimination of the closed BPV
- Condition for closing of the SGV

After this control the calculation continues according to the main directions marked in S*tage3* .

2.4 INCREASE OF THE HEAD TO THE TOTAL CLOSING OF SGV

The calculation continues by analogy with *Stage 4* till the total closing of SGV.

2.5 SLOW INCREASE OF THE PRESSURE TO THE FULL CLOSING OF THE MV

The problem is solved by iterations according Eq. (1) and equations for filling of MV-(14), (15).

3. NUMERICAL RESULTS

The following data are accepted. For a rotary pump and the motor: two pumps of $Q_0 = 0.450$ m³/s; and velocity of $v_0 = NQ_0 / F_1 = 2.0.450/0.636 = 1.415\, m/s$,el.motorA-122/4, with

capacity of $N_0 = 630kW$, [GD²]=100 $kg_f.m$,η_m^{mech} =0.987. A pipe length and wave velocities are: $l_1 = 1950m, l_2 = 1475m, l_3 = 1784m, \sum l_m = 5209m, a_1 = 908m/s, a_2 = 908m/s$, $a_3 = 852m/s$. For diameters and boundary conditions the data are: $d_1 = 0.9m, d_2 = 0.9m, d_3 = 1.1m, y_0 = 80m$. A submerged pump is accepted with size: $h_s = 2m, d_s = 0.7m$. The coefficient of the suction pipe and the size of MV are:

$$\sum \xi^s = \frac{\lambda l_s}{d_s} + 5 = 5.11, d_{fi} = 6mm, D_{di} = 2mm, l_{fi} = 0.6m, \sum \xi_{fi} = \xi_f + (\xi_{en} + \xi_{ex} + \xi_{cu}) +$$

$$+ \xi_{di} = 190.6, d_{ev} = 14mm, l_{ev} = 0.7m, d_{start-givingtube} = 6mm.$$

Operation of the set is performed according to the conditions given in section 2.3. The final results are shown in Fig. 3 for the consecutive stages.

4. CONCLUSIONS

The protection of pipeline with diameters Ø900 mm and 1100 mm is realized by two numbers MV Ø150 and diaphragms d_{di} Ø2. The numerical procedure for solving the pressure and flow distribution includes a variety of components along the pipeline and uses OOT. The maximum result of 92 m is equal practically to the steady head of 91.02m.By means of the feeder tanks with air-valves in nodes k=7,15 the additional water volume is respectively 1.275 m^3 and 3.506 m^3. This protection enables to trace the pipelines closed to the ground without thick pipes wall and deep tunnels. Thus the solution allows an effective design of one pumping station without appearance of cavitation zones.

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