REVERSIBLE OPERATION OF ROTARY PUMPS

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Abstract

The reversible operation of rotary pumps results from the use of programmable membrane valves. These valves were built for the first time by the American company CLA-VAL, and were later produced also in Bulgaria. Some theoretical and experimental investigations were performed on the reversible operation of valves by taking into account the elasticity, rotary mass and the loss. A numerical model with a mounted membrane valve was developed and compared with the experimental data.

The authors have studied membrane valves as means for protection because of their advantages as compared to back-pressure valves and other mechanical regulators.

One advantage is that they have hydraulic closing permitting reversible revolutions smaller than the rated revolutions, and smaller pressure.

In the present paper, an algorithm and numerical model are described in the case of reversible operation of rotary pumps. The final results show the value of the maximum reversible revolutions and the maximum pressure in the pump and the pipeline in the case of membrane valve protection. Moreover, a method is suggested for the development of four-quadrant characteristics of rotary pumps. The practical application of these characteristics in computer programs is possible by including correction factors depending on the dynamic equation of motion, similarity laws, experiments and spline functions. With a view to test the proposed algorithm, some experiments with analogue-digital equipment were carried out, and practical conclusions were drawn.

The authors are looking for co-operation for further development of software, the algorithm and for Internet publicity.

Keywords

Rotary pump, reversible operation, protection, back-pressure valve, membrane valve, brake regime, turbine regime, revolution, numerical method, algorithm, break-down disconnection, similarity laws, rated revolution, transient process.

Introduction

The reverse motion of water in rotary pumps can result in the absence of protection or by closing with a different type protection. In the present study, a programmable protection of the membrane valve type is described. The transitional process with membrane valve protection includes five major parts (See Fig. 3):

Part 1: Reduction of pump revolutions from the rated value to zero and hydraulic pressure drop to a minimum value.

Part 2: Increase of pressure and reversible revolution to maximum value - turbine mode of operation. The membrane valve chamber starts to fill up.

Part 3: Pressure drop and reduction of reversible revolutions to a practically steady motion.

Part 4: Further steady motion depending on the diaphragm diameter of the filling-and-evacuating tubule.

Part 5: Rapid pressure increase in the pipeline until the moment of total closing of the membrane valve.

DEVELOPMENT OF THE MODEL

The proposed algorithm is based on the above mentioned parts from one to five and will be considered in detail below.

PART 1. REDUCTION OF PUMP REVOLUTIONS FROM RATED TO ZERO AND HYDRAULIC PRESSURE DROP TO A MINIMUM VALUE

In the case of current breakdown in the rotary pump, a continuous decelerating motion arises in the direction of pumping. The pressure drops to the minimum and the revolutions are changed from rated to zero. In view of the characteristics of the transient process, i.e. $v \ll a$, where v is flow velocity, and a is the velocity of wave spreading, the principal equation of the unsteady motion are [1, 4]:

$$\frac{\partial}{\partial s}\left(\frac{p}{\gamma}+H\right)ds \pm \frac{a}{g}\frac{\partial v}{\partial s}ds = -S_f\,ds \qquad (1)$$
$$dx/dt = \pm a$$

where p denotes the pressure of the flow; v and H are the flow velocity and pipe elevation respectively; S_f is friction slope; $\frac{dx}{dt}$ denotes the characteristic direction; $a = \sqrt{\frac{g/\gamma}{1/\varepsilon + d/E\delta}}$ is the wave spreading velocity; g is the gravity acceleration; ε and E are the elastic module of the fluid and the pipe material; δ, d are respectively the pipe wall thickness and diameter.
Let us introduce a co-ordinate system as shown on Fig. 1, and the initial conditions are introduced in the following form: $p/\gamma + H = y$, $y\,(t{=}0) = y_0$, $v\,(t{=}0) = v_0$.

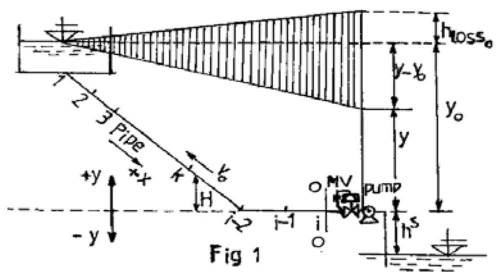

Figure 1. General scheme of the pump with membrane valve

The integration of (1) is by the Euler method along characteristic direction with a step in length $\Delta l = l/n$ and a step in time $\Delta t = \Delta l/a$, i.e. the time indices are: $t_i = i\Delta t$. For the head in cross section 0-0 (Fig. 1) the result is:

$$y_i = y_o + \frac{a}{g}\left(v_i - v_{i-1}\right) + \frac{a}{g}\sum_{j=2}^{i-1}\left(v_j - v_{j-1}\right) + \frac{\Delta l}{C^2 R}\,\mathrm{sign}(v_i)\,\frac{v_i^2 + v_{i-1}^2}{2} + \\ + \frac{\Delta l}{C^2 R}\,\mathrm{sign}(v_j)\sum_{j=2}^{i-1}\frac{v_j^2 + v_{j-1}^2}{2} \qquad (2)$$

The equation (2) is true when time indices "i" satisfy the inequality $i \le n$ i.e. $t_i = i\Delta t \le T_{prop}$, where $T_{prop} = l/a$. If the inequality for time indices "i" is $i > n$, the equation (2) should be extended by Euler-McLoren formula for all indices "r".
In such case "r" is integer number that is a counter of the entire phases $T_{prop} = l/a$.

BOUNDARY CONDITION

The boundary condition in cross section at the reservoir is y_0=const.The boundary

condition in cross section 0 at the pump is described most accurately in [1] by taking into account the 4-quadrant pump characteristics. Here, the method is applied with characteristics approximated with cubic splines in the form of Hermit.

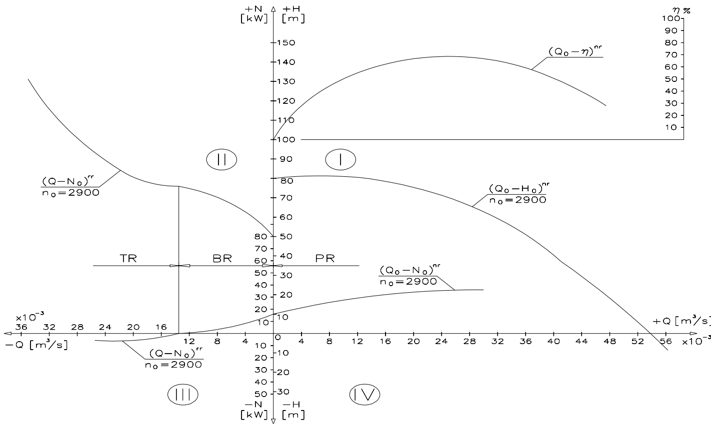

Figure 2. Four quadrant characteristic of rotary pump.
Abbreviation: BR – brake regime; TR – turbine regime; PR – pumping regime;
rr – reverse revolution ; nr – normal revolution; n_o – rated revolution

The capacity N is calculated in a moment $t_i= i\Delta t$ by means of the following equations:

$H_{oi}\geq 0$, $$N_{oi} = \frac{9.81 Q_{oi} H_{oi}}{\eta_p \eta_m^{mech}} \qquad (3)$$

$H_{oi}<0$, $$N_{oi} = \frac{9.81 Q_{oi} (H_{oi} + \Delta H_i)}{\eta_p \eta_m^{mech}} \qquad (4)$$

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

$$\Delta H_i = A n_o^2 + B n_o Q_{oi} + H_{oi} \qquad (5)$$

If $\Delta H_i > H_{oi}$, it is accepted $H_{oi} = \Delta H_i$. The coefficients A, B are determined by means of two points from Q-H of the highest efficiencies η_i. After calculating the capacity N_i at moment $t_i = i\Delta t$ it is necessary to find the revolutions n_i.The following relationships are used to this end:

$n_i \geq 0.3 n_o$ $\quad n_i = \dfrac{k_i}{k_i / n_{i-1} + \Delta t}$ $\qquad$ $n_i < 0.3 n_o$ $\quad n_i = n_{i-1} - \dfrac{\Delta t}{k_i}(0.3 n_o)^2 \qquad (6)$

$$k_i = \frac{n_o^3 [GD^2]}{365000 \overline{N_{oi}}}, \overline{N_{oi}} = \frac{N_{oi-1} + N_{oi}}{2}, [GD^2]$$ -is the rotary moment of the pump aggregate.

The head H_i and the velocity v_i at the moment $t_i = i\Delta t$ are found by means of similarity laws. The head in the rotary pump h_i^{rp} with suction is calculated by formula:

$$h_i^{rp} = H_i \mp h_i^s \qquad (7)$$

where h_i^s is suction height of the pump (the sign minus is valid for unsubmerged pump, plus-for submerged pump). The head in the pump calculated by formula (7) must be equal to the head caused by the transient process (2). An iteration procedure related to discharge Q_{oi} is

necessary in order to achieve the required condition for precision. The iteration procedure is based on Newton's method of solving non-linear systems.

PART 2. INCREASE OF PRESSURE AND rr TO MAXIMUM VALUE - TR

The head should be calculated along the pipe by means of formula (2).

The boundary condition was established on the basis of experimental study of the reversible operation of two types of pumps: type 1 with high-speed n_o = 2900 rpm and type 2 with slow-speed n_o=1000 rpm [1,2].

The equation for (Q - H) characteristic in II quadrant with reversible revolutions (rr) in TR is

$$H_{oi}^{rr} = H_{Q_{max}^{BR}}^{rr} + 6.9\left[\frac{Q_i^{rr} - Q_{max}^{BR}}{Q_{H=0}^{nr}}\right] H_{Q=0}^{rr} \tag{8}$$

where H_{oi}^{rr} - head with rated rr in moment t_i; $H_{Q_{max}^{BR}}^{rr}$ - head for rated revolution n_o;

Q_{max}^{BR} is maximum discharge for BR which is calculated as follows, $Q_{max}^{BR} = \psi Q_{H=0}^{nr}$ where ψ=0.253 for a high-speed pump; ψ=0.295 for a slow-speed pump in all other cases this coefficient ψ is determined by interpolation or extrapolation of the given values.

The head in BR is:

$$H_{oi}^{rr} = H_{Q=0}^{rr} + \left|\frac{Q_{oi}^{BR}}{Q_{H=0}^{nr}}\right|^{0.75} \times H_{Q=0}^{rr} \times 1.453 \tag{9}$$

The head for the value of discharge equal to zero is:

$$H_{Q=0}^{rr} = \beta H_{Q=0}^{nr} \tag{10}$$

The values of the coefficient β are determined as follows: β =0.625 for a high-speed pump and β =0.390 for a slow-speed pump, in all other cases the β is determined by interpolation or extrapolation; $Q_{H=0}^{nr}$ is a discharge with nr for H=0; Q_i^{rr} is a discharge with rr at the moment t_i.

The equation for (Q-N) characteristic in III quadrant with rr in TR is:

$$N_{oi} = 9.81\eta_{pj}\eta_m^{mesh}\eta_{Rj}Q_j H_{oj}^{rr} \tag{11}$$

Q_j in equation(11) is composed by means of four spline functions between interpolated points. The increased pressure and maximum rr can be determined by means of the formulas below. When the minimum value of the head of the rotary pump has been reached a reverse motion of water-flow begins in the pipeline to the pump with velocity v_o and losses:

$$h_{loss} = \left(\frac{2gl}{C^2R} + \sum\xi_{cur} + \xi_{en}\right)\frac{v_o^2}{2g} \quad \text{or} \quad v_o = \sqrt{\frac{2gh_{loss}}{\sum\xi_{loss}}} \tag{12}$$

where l is length of the pipeline, $\sum\xi_{cur}$ - sum of local disturbance by curve, ξ_{en} is a disturbance by entrance, h_{loss} is determined by equation (7) for h_{min}^{rp}.

The value of velocity v_o calls forth initial revolution n_{ho} according to similarity low for h_o:

$$\frac{H_o}{h_o} = \left(\frac{n_o}{n_{ho}}\right)^2 , \quad n_{ho} = n_o\sqrt{\frac{h_o}{H_o}} , \; h_o = h_{min}^{rp} + \frac{1}{4}h_{loss}^{mv} \tag{13}$$

Dynamic equation of the pump in IIquadrant with TR taken into account the rotary masses is:

$$M = -J\frac{d\omega}{dt} \tag{14}$$

where M is a dynamic moment of the aggregate, ω is an angular speed of the rotation, J is a rotary moment. Having in mind the similarity low for II quadrant and after transformation equation (14) leads directly to the formula for revolutions:

$$\ln n_i = \ln n_{i-1} + \frac{\Delta t}{k_{1i}} \tag{15}$$

where $k_{1i} = \dfrac{n_o^2\left\lfloor GD^2\right\rfloor}{365000\overline{N}_{oi}}$, $\overline{N_{oi}} = \dfrac{N_{oi-1} + N_{oi}}{2}$.

By means of equations (7) and (8) the head in the rotary pumps is:

$$h_i^p = H_{oi}^{rr}\left(\frac{n_i}{n_o}\right)^2 + 0.5\xi_i^{mv}\frac{v_i^2}{2g} \mp h_i^s \tag{16}$$

Here the coefficient of ξ_i^{mv} is calculated according to experimental data for ∅100 v=24ml/mm The problem is solved consecutively for the separate intervals by iteration using (16) and (2). The final iteration is reached for accepted v_i when it gets equality of h_i^p (16) with y_i (2). The maximum rise in the pressure and maximum value of rr are get with achievement of Q_{max}^{BR}.

PART3.PRESSURE DROP AND REDUCTION OF rr PRACTICALLY STEADY MOTION

The formula for (Q-H) is given by (9). (Q-N) characteristic in II quadrant in BR is:

$$N_{oi}^{BR} = \left(\frac{Q_{max}^{BR} - Q_i}{Q_{max}^{BR}}\right)^{1.5} N_{Q=0}^{BR} \tag{17}$$

where $N_{Q=0}^{BR}$ is a given extrapolated point of N_o^{nr} according to $N_{Q=0}^{nr} \approx N_{Q=0}^{BR}$.

When the point Q_{max}^{BR} is obtained the TR is exhausted and BR begins with fall of the head and rr (see Fig.3 and Fig.4).

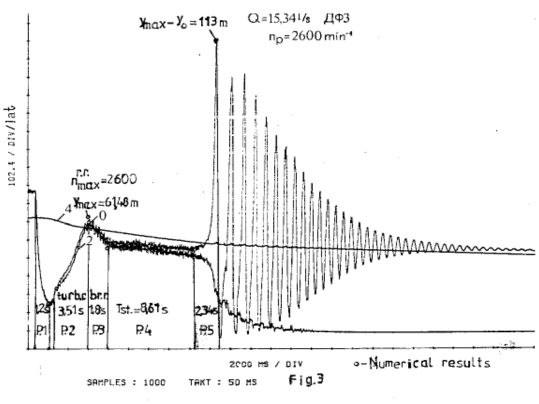

Figure 3. Experiment 18 rotary pump 25E80a membrane valve ∅100mm diaphragm ∅3, Q=15.34.10^{-3} m^3/s n_o^{nr}=2900, n_{max}^{rr}=2600rpm

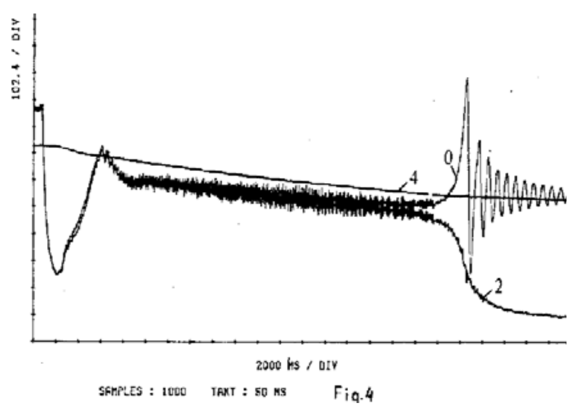

Figure 4. Experiment 22 rotary pump 25E80a membrane valve ∅100mm diaphragm ∅2, Q=15.69.10^{-3} m^3/s n_o^{nr}=2900, n_{max}^{rr}=2600rpm

PART 4. FUTHER STEADY MOTION DEPENDING ON THE DIAPHRAGM DIAMETER

Practical steady motion in pipeline is observed. For the performed experiment with diaphragm ∅3mm of the filling–evacuating tubule - Fig.5 the calculated time of steady motion is 8.61s.

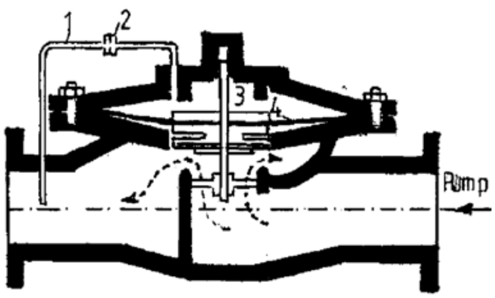

Figure 5. Membrane valve –
1. Filling-evacuating tubule; 2. Diaphragm;
3. Membrane chamber; 4. Membrane

PART 5. RAPID PRESSURE INCREASE IN THE PIPELINE UNTIL THE MOMENT OF TOTAL CLOSING OF THE MEMBRANE VALVE

Transient process in the pipeline to full closing of membrane valve is established.

It is accepted Δ=0.195 with ξ^{mv}=42. After relatively fast closing of the membrane valve the discharge Q=9.896 l/s decreases to 0 and a sharp increase of the pressure in the pipeline begins. The problem is solved with iteration through the method of finite differences.

NUMERICAL RESULTS

Example is according to experiment 18 in Fig.3. The data are with a rotary pump $Q=0.01534m^3/s$, n_o^{nr}=2900rpm, y_o=62m, l=223m, d=100mm, a=1235m/s, Δt=0.18s(step), $\sum \xi_{pipe} = 79.2$, h^s=1m(submerged), $[GD^2]=0.82kgm^2$ for AO2-72/2, 30kw, η_m^{mech}=0.936

The numerical results are compared with experiment 18. They practically coincide.

Part 1: y_{min}=15m, T=1.2s

Part 2: After n^{nr}=0 return of the water-flow begins. After the time for spreading and transmission t= 0.36s the velocity v_2=3.244m/s is created according to (12). Then the initial drive rr are n_{ho}=1193 rpm. Further more according to the method in Part 2 the revolutions and head are $n_{max}^{rr} = 2600rpm$, y_{max}=61.48m, T_{TR}=3.51s.

Part 3: After achievement of $n_{max}^{rr} = 2600rpm$ BR begins with decrease of the revolution and fall of the pressure to 2332rpm. The final results are y_{min}=46.86m and T_{BR}=10x0.18=1.8s.

Part 4: It is accepted that the steady motion with Q=9.896 l/s practically finishes for ξ^{mv}=42. Fast increase of ξ^{mv} begins after ξ^{mv}>42. By the data of chamber volume (see Fig. 5), the final result for steady time is T_{st} =8.61s.

Part 5: The following results are achieved: y_{max} -y_o=113m, T=2.34s.

CONCLUSION

Obviously the size of diaphragm ∅ is determinative membrane parameter for the maximum of hydrodynamic pressure which is 113m for ∅3mm and 42m for ∅2mm. The following conclusion could be obtained: whenever there is no protection of pumping aggregates the algorithm described in part 1, 2, 3 is valid but without the filling of membrane chamber and the term with ξ_i^{mv}. The authors look for further creating of software and algorithm with membrane valve mounted on branch of the pipeline, pressure relief valve “NEYRTEC” and for popularity by INTERNET.

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