Theoretical and Experimental Research Regarding Amortisation of Presure Pulsation for Pumps with Plungers

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Abstract

This paper presents the results of the research work performed by the author regarding theoretical and experimental work focus on the pulsation amortisseurs of pressure. On practical experiments it was used a type of dampers of pulsation with inert gas and without membrane These kind of damper equipes the pumps with plungers used as hydraulic generators in the oil hydraulic extraction units.

Key words: *damper, pumps, hydraulic extraction unit*

Introduction

The hydraulic extraction units work in several conditions characterized by a rapid change of the functional parameters.The adjustement at the start conditions it's usualy do by regulation the debit of pumps by the help of the gearbox with 7 steps or by modifying the diameter of the pistons.The pulsations of debit represent a source of perturbations even they are reduced as amplitude when pumps with plungers single effect with 5 or 3 cilinders are used.It can appears vibrations in underground pipes system, in the pumping head, the succion pipes or in the evacuation ones. These aspects can be amplified if the gas presence in the extracted oil is situated at high level (above 5%) which imply a faulty functionation of the deep hydraulic unit.The used damper of pulsations without membrane and its symplifyed diagram, [1], is presented in figure 1.The body of the damper has a spheric shape an dis equiped with 2 lines (input and output) (1, 6).The laboratory testing unit consists of 2 emispheres (2, 5), catcing throught 2 flange (4), and a seal between them (8).This mode asure the posibility of changing the interior disks. At the interior end of the line is mounted a disk (3), which has 8 holes in L.

The report between the presure pulsation amplitude at the line with damper p_1 and the presure pulsation amplitude at the line without damper p_0 is do by the relation:

$$\frac{p_1}{p_0} = \left[\left(\omega^2 L_a C_a - 1 \right)^2 + \omega^2 R_a C_a{}^2 \right]^{-\frac{1}{2}}, \qquad (1)$$

where:
p_1 is the presure pulsation amplitude at the line with damper;
p_0 – presure pulsation amplitude at the line without damper;
ω – pulsation of debit swings;

L_a – acustic inductance,

$$L_a = \frac{\rho \cdot l}{A},\tag{2}$$

where:

ρ is density of the fluid;

l – lenth of shock tube;

A –surface aria of shock tube;

C_a – acustic capacity.

$$C_a = V c^{-1} \rho^{-1},\tag{3}$$

where:

V is volume of the damper;

c – propagation speed of sound through the liquid

$$c = \left(\frac{E}{\rho}\right)^{\frac{1}{2}},\tag{4}$$

where:

E is liquid eqivalent elasticity module;

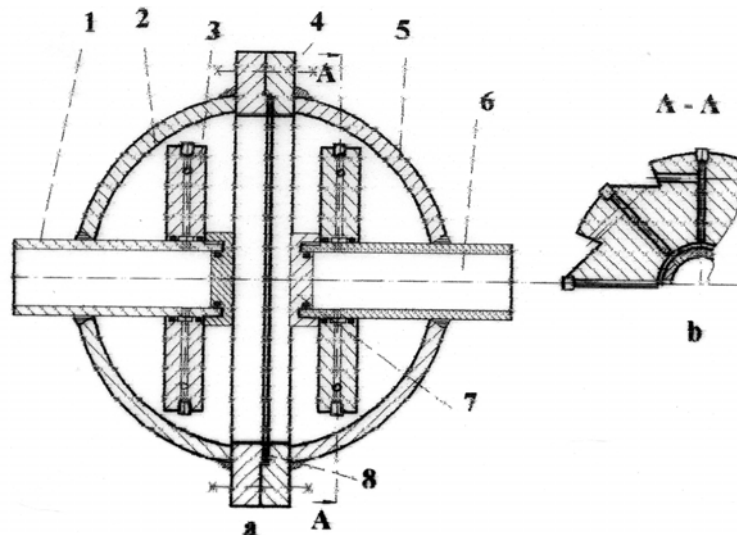R_a – acustic resistance.

$$R_a = \frac{\rho \cdot c}{A}\tag{5}$$

Fig. 1. Hydraulic damper: 1 - input pipe; 2 - halfbody; 3 - input disk;
4 - colar; 5 - halfbody; 6 - output pipe; 7 - bonnet; 8 - seal of ring.

Liquid elasticity influence

The small compresability of the liquids allows the quikly transmition of the movement grant to hydrostatic transmitions a high rigidity. On normal conditions the elasticity module of the oils used at the hydrostatic transmitions is in the range $(1,3...1,7) \cdot 10^3$ MN/m², [2].

The dinamick liquids based on water generally have the elasticity module aproximated as water $2,1 \cdot 10^3$ MN/m². Generally the ellasticity module (E) of the petroleum products presents important variations being influenced by pressure, temperature, and disolved gases. In the work, [3] is said that in the range of pressure $2,5...50$ MN/m², E depends only by temperature and density, which is not total corectly. In the same work presents values of E in the range

$(0,55…1,66)·10^3$ MN/m^2. In the pressure domain $0…50$ MN/m^2, E has a liniar grow with the pressure as the realation:

$$E(p) = E(p_{at}) + pK_e, \qquad (6)$$

where:
$E(p_{at})$ is ellasticity module at athmospheric pressure;
p – working pressure;
K_e–corection coeficient depending on temperature, pressure and undisolved gas, $K_e \in (10…15)$.

Experimental on pulsation flow damper

The experiments do on the dumper in fig.2 located on bench releved the mode of reduction of the amplitudes of pressure drops (because of flow pump oscilations).
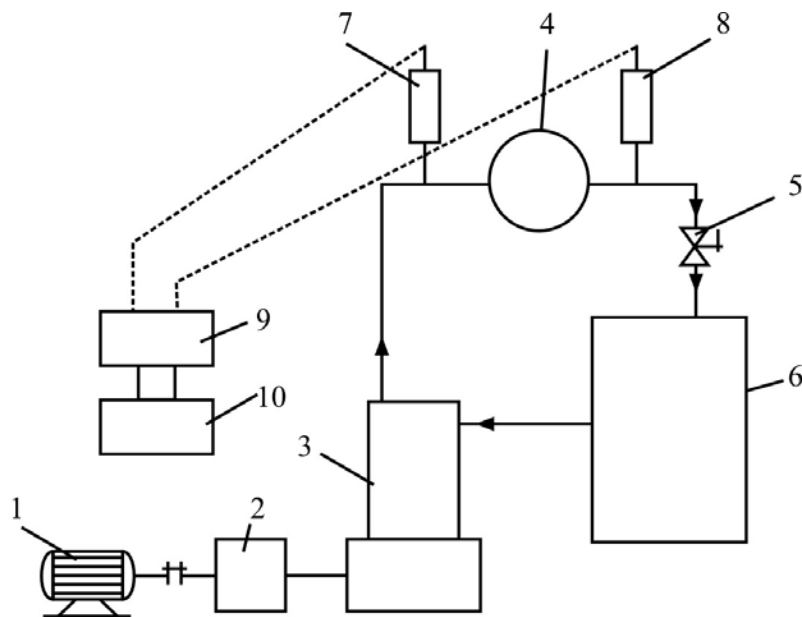

Fig.2. Scheme of experimental bench of hydraulic damper:
1- electric engine; 2 - gear box; 3 - triplex pump with single action;
4 - damper of pulsations; 5 - cock robinet; 6 - tank; 7 - pressure traducer
on inlet of damper; 8 - pressure trasducer on outlet of damper;
9 - strain gridge; 10 - osscilograph loop system.

The volumic pump is 3PN–15 type and has 3 gears. For vibration pressure law enlistment were used 2 tensiometric transducers were installed downstream and upstream of damper, a tensiometric punte N2321 and a osscilograph galvanometric loop system H–117.

The box of gear with 3 steps permited the experimental determinations for diferent work waves and to releve the efficiency of the damper for diferent flows.

Fig. 3 presents one of the registering which was done. The medium pressure downstream of damper, was of ,58 MN/m^2 and upstream of hydrophor 0,44 MN/m^2.The angular speed of the pump is 15,7 rad/s. By graph results the pressure downstream the damper has a oscilation between a minimum value of 0,46 MN/m^2 and a maximum value of 0,73 MN/m^2, and upstream of hydrophor is around of a constant value by 0,44 MN/m^2; above this a high frecvence oscilation is superimpose. The pressure drop 0,58–0,44=0,14 MN/m^2 represents the loss of energy because of the presence of damper in discharge of pipe.
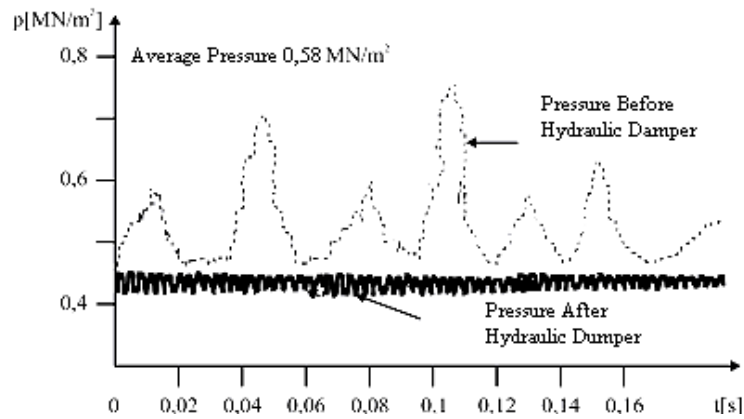
Fig. 3. Osscilation diagrams on behind of damper (p_1)

and back of damper (p_2).

The main dimensional elements of pump 3PN–16 are: piston diameter = 60 mm; stroke
S = 56 mm. Angular speed for experiments are: ω_1 = 5,23 rad/s; ω_2 = 6,28 rad/s; ω_3 = 15,7 rad/s.
The medium corespondent flows are: Q_1 = 0,395 l/s; Q_2 = 0,475 l/s; Q_3 = 1,186 l/s.

The yelds of damper is

$$\eta_h = \frac{p_{2n}}{p_{1n}} \cong 0,758.$$

The calculus of coeficients of iregularity for pressure goes to:
– input the damper, $\varepsilon_1 = 0,465$;
– output the damper, $\varepsilon_2 = 0,045$;

it results a satisfied value of amortisation degree of oscilations:

$k = \dfrac{\varepsilon_1}{\varepsilon_2} = 10,33$ which can be considered adequately.

For the second working regim at which the angular speed of cranckshaft it was 15,7 rad/s, the
average value of pressure upstream the damper it is 1,02 MN/m^2, and downstream the damper is
0,9 MN/m^2. The difference 1,02-0,9=0,12 MN/m^2 represents the loss of energy at when the
liquid is moving throug the experimental damper. The comparison of thr registered signals can
confirme that the hidrophor has an important effect on oscilations and expecially on those of
low frecvency.

If the experiment is repeated at a higher grade of obturation of the van on the discharged pipe it
results an average value of pressure upstream the dumper , 1,72 MN/m^2, and downstream of 1,5
MN/m^2. The diference of 1,72–1,5 = 0,22 MN/m^2 represents the loss of energy at when the
liquid is moving throug the experimental damper.

For this regim in spite of the large oscilations of pressure upstream the damper (minim 1,4
MN/m^2, maxim 1,92 MN/m^2) at the outlet of dumper pressure remains approximate constant
(1,5 MN/m^2) if high frecvency oscilations are neglected.

The same observations can be done when the pump is working at a revolution of 5,23 rad/s. At
the one of the recorder [3], the average value of pressure upstream of dumper was by 0,6
MN/m^2, and downstream was by 0,3 MN/m^2.The difference 0,6–0,3 = 0,3 MN/m^2 represents
the pressure loss at the movement of liquid throug the dumper.

There was done 10 recorders which are presented in the paper [3]. It may be mentioned that the vehiculated liquid on laboratory banch was water.

Because the experimental data are obtained from a periodic process with a pulsation ω or period T ($\omega T = 2\pi$), is possible like the best pressure express as a function ot time [p_1 (t)], to be a thrigonometric polinom by form,

$$y = \frac{a_0}{2} + \sum_{j=1}^{m} \left(a_j \cos j\omega t + b_j \sin j\omega t \right), \tag{7}$$

It's consider the particular case when the n values y_k where determined for equidistanced values of the independent variable as

$$t_k = \frac{T}{n}k \text{ , for } k = 0,1,2,\ldots, \text{n–1}$$

Because the system of functions 1, cosωt, sinωt,…, cosmωt, sinmωt (for $m \leq \frac{n-1}{2}$) is octogonal on the set values t_0, t_1, …., t_{n-1}, the parameters in expression (7) are calculated by relations:

$$a_j = \frac{2}{n}\sum_{k=0}^{n-1} y_k \cos j\omega t_k, \text{ for } j = 0,1,\ldots,m$$
$$b_j = \frac{2}{n}\sum_{k=0}^{n-1} y_k \sin j\omega t_k, \text{ for } j = 1,2,\ldots,m \tag{8}$$

where $\omega t_k = \frac{2\pi k}{n}$.

For the analyse it's considered a recorder [3], the working of pump 3PN–15, at a angular speed ω = 6,28 rad/s, a n average flow of pump by 0,475 l/s. This is presentes in figure 4.
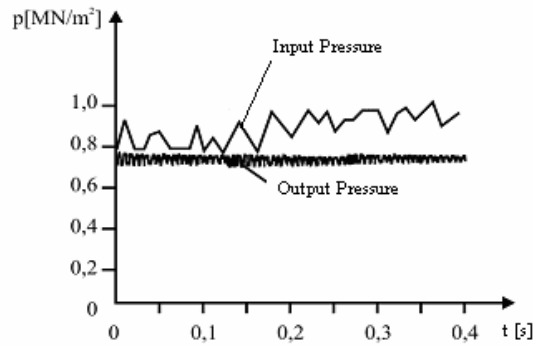

Fig. 4. Amortisation effect by reducing pressure oscilations.

From diagram it can be observed that the medium pressure upstream is 0,85 MN/m^2, and downstream 0,74 MN/m^2 resulting a pressure losses on damper of 0,11 MN/m^2. Based on recorders in forty points are calculated the coeficiencies of the thrigonometric polinom with relation (7). The input pressure can be express by the relation:

$$p_1(t) = 0,85 + \sum_{j=1}^{19} \left(a_j \cos j\omega t + b_j \sin j\omega t \right). \tag{9}$$

If it considers the uper observations regarding the presence of gases in extracted oil products (even the oil is reoperated to a separation step in the deeper hydraulic agregat), a small percent of gases it exist that it can induced more dinamic effects.The armonic analyse of the signal presents in fig. 4 releve the causes of changing of the flow for the used pump by side of oscilations caused by the movement of crankshaft it apears 8 armonics of high frecvences comparative with the fundamental one: f_8, f_9, f_{10}, f_{13}, f_{14}, f_{15}, f_{16}, f_{19}. These perturbations which generat „noise" have swings in the range 0,049 and 0,011 of the base signal (because of the movement of the crankshaft) and can be explained by the usage of elements of the used pump. Regarding this aspect, the eficiency of the proposed type damper is proven, by the initial calculus of the amortisation system, including the situation of uncontrolable perturbations (causing by usage), too.

Conclusions

The conclusions of the work:

*   the pressence of the petroleum gases have a positive effect from the point of view of the presure vibrations reducing from the pumps with plunger to deep hydraulic unit;

*   the pressure losses through the damper of pulsations it's express through a parabolic law by the form;

*   the amortisation effect of oscilations is more powerfull at small angular speeds of crankshaft of the pump;

*   the efective calculus for the dumper of pulsations which was studied goes to a the value 0.758 for the hydraulic effective power (η_h) and an amortisation grade k=10,33;

*   the armonic analysse of the inlet pressure in pulsatios dumper results an aproximation of it by a thrigonomertical polinom presented by relation (9).

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Cercetări teoretice şi experimentale privind amortizarea pulsaţiilor de presiune de la pompele cu piston

Rezumat

În lucrare, autorul prezintă câteva din cercetările teoretice şi experimentale efectuate asupra amortizoarelor de pulsaţii ale presiunii. Aceste amortizoare de pulsaţii sunt de tipul fără membrană şi gaz inert. Acestea echipează pompele cu piston folosite ca generatoare hidraulice la instalaţiile de extracţie hidraulică a petrolului.