ENERGETICS. HEAT ENGINEERING. ELECTRICAL ENGINEERING

ЕНЕРГЕТИКА. ТЕПЛОТЕХНІКА. ЕЛЕКТРОТЕХНІКА

UDC 534.1:62-752

O.V. Korolyov, DSc, Prof.,
Zhou Huiyu
Odessa National Polytechnic University, 1 Shevchenko Ave., 65044 Odessa, Ukraine; e-mail: korolyov118@gmail.com

DYNAMIC DAMPER PRESSURE FLUCTUATION IN THE PUMPING SYSTEMS

О.В. Корольов, *Чжоу Хуіюй*. **Динамічні погашувачі коливань тиску у насосних системах.** Інерційна частина будь-якого пристрою або машини (наприклад, насоса), підвішена або укріплена на пружному каркасі, що перебуває під дією збуджуючої сили, яка діє з постійною частотою, може бути схильна до коливань, особливо поблизу резонансної ділянки. Для усунення таких коливань можна вдатися до використання динамічного погашувача коливань. ***Мета:*** Метою роботи є аналітичне дослідження різних динамічних погашувачів для завдань зниження коливання тиску в насосних системах. ***Матеріали і методи:*** Порівняльний аналіз ефективності функціонування був проведений для динамічних погашувачів двох типів — гідравлічного і механічного. ***Результати:*** Представлено методику розрахунку динамічного погашувача коливань тиску рідини в насосах гідравлічного і механічного типу. Алгоритми розрахунків доведено до інженерних застосувань і впроваджено у виробничий процес. Проведені розрахунки показують, що застосування механічних динамічних погашувачів коливань доцільне на високочастотних насосах, разом з тим, при більшій частоти роботи насоса в 6 разів, виграємо в габаритах демпфера в 3,5 рази.
Ключові слова: динамічний гаситель коливань, демпфер, коливання, збуджуюча сила.

O.V. Korolyov, *Zhou Huiyu*. **Dynamic damper pressure fluctuation in the pumping systems.** Inertial part of any devices and equipment (e.g., pumps), hung or mounted on the resilient frame and being under the influence of the disturbing force that works at a constant frequency, may be subject to fluctuations, especially near of the resonance area. For elimination these fluctuations, you can resort to the use of a dynamic damper. ***Aim:*** The aim of the work is an analytical study of various dynamic dampers to reduce pressure fluctuation problems in pumping systems. ***Materials and Methods***: A comparative analysis of efficiency of functioning was carried out for two types of dynamic dampers — hydraulic and mechanical. ***Results:*** The technique for calculating of dynamic damper of fluid pressure fluctuations in the hydraulic and mechanical pumps is presented. Algorithms of calculations are reported to engineering applications and implemented in the production process. The calculations show that the use of dynamic mechanical dampers is expedient at high frequency pumps, and, with increasing frequency of the pump by 6 times, winning in the dimensions of the damper in 3.5 times.
Keywords: dynamic damper, damper, fluctuations, disturbance.

Introduction. Inertial part of any devices and equipment (e.g., pumps), hung or mounted on the resilient frame and being under the influence of the disturbing force that works at a constant frequency, may be subject to fluctuations, especially near of the resonance area. With regard to the pumping systems we can confidently assert that such fluctuations lead to increased vibration of pipelines, decrease the pumping system resource as a whole, as well as to a significant error in the measurement of flow rate, supplied in such pump. Elimination of such fluctuations can be done in one of two ways: either to stop the impact of the disturbing forces, that in the case when the source of disturbances is a pump system itself is impossible; either deduce the system from the resonance area, changing the inertia and resilient components of the system. However, the case with the suction conduit of the piston pump that is connected to a supply container of a large volume, this approach is unworkable. In this case, you can resort to the dynamic damper (DD), invented in 1909 by Frahm.

DOI 10.15276/opu.1.48.2016.07

 ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

The basic dynamic damper diagram is shown in Fig. 1. Thus, a periodic disturbing force $P_0 \sin \omega t$ acts on the inertia part of the mass *M* that contained in the mechanism. The resilient component of the system is summarized by a spring with stiffness *K*. The dynamic damper is an oscillating system with mass *m* relatively smaller then pump and by a spring with stiffness *k*. As can be seen from the figure, DD is related with the mass *M*. Conditions of DD work is the equality of natural frequency $\sqrt{K / m}$ of the associated damper and frequency ω of the disturbing force. In this case, the work of the whole system is realized in such way that the mass *M* does not fluctuate, and the oscillating system with mass *m* and the spring stiffness *k*, fluctuates so that elastic force of the spring is equal in magnitude and opposite in direction to the disturbing force $P_0 \sin \omega t$.

Full evidence of positions presented hereinafter given, for example, in [1, 2].

We introduce the following dimensionless parameters: $x_{st} = P_0 / K$ — static deformation of the main system, m; $\omega_d = \sqrt{k / m}$ — the natural frequency of the dynamic damper, 1/s; $\omega_s = \sqrt{K / M}$ — the natural frequency of the main system, 1/s; $\mu = m / M$ — the ratio of the mass of the dynamic damper to the mass of the main system.

Given these values can be obtained the dependence for the relative amplitudes of oscillations of the pump mass (1) and DD mass (2):

$$\frac{x_M}{x_{st}} = \frac{1-\omega^2/\omega_d^2}{(1-\omega^2/\omega_d^2)\left(1+\dfrac{k}{K}-\dfrac{\omega^2}{\omega_d^2}\right)+\dfrac{k}{K}}, \tag{1}$$

$$\frac{x_m}{x_{st}} = \frac{1}{(1-\omega^2/\omega_d^2)\left(1+\dfrac{k}{K}-\dfrac{\omega^2}{\omega_s^2}\right)-\dfrac{k}{K}}, \tag{2}$$

where ω — the frequency of the disturbing forces rad/s (1/s).

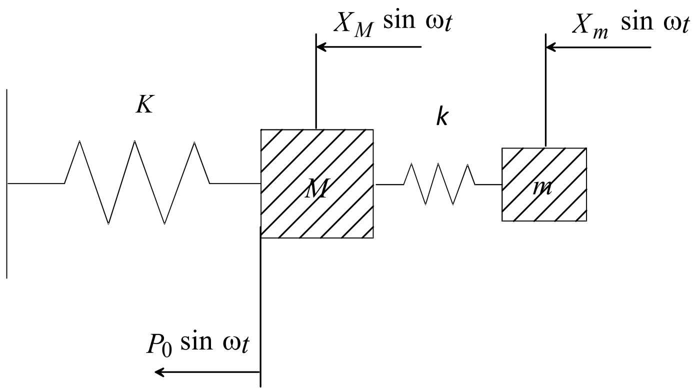

Fig. 1. Schematic diagram of the dynamic damper

As we can see from (1) and (2), when the frequency of the disturbing force ω and the DD natural frequency ω_d are coincided, the amplitude of the mass oscillation *M* goes to zero $x_M \to 0$, and the amplitude of oscillation DD, respectively, proportional to the ratio of springs elasticity coefficients of main and auxiliary mass

$$x_m = -x_{st}\frac{K}{k}.$$

The main disadvantage of dynamic dampers designs is their narrow range of work, which limits their use mainly in systems with disturbing force of constant frequency, such as synchronous machines. Recently, however, the range of DD applicability expanded by resolving the problems of

DD work in the systems with disturbing force of variable frequency. For this purpose DD is manufactured with possibilities to adjust its own frequency ω_d and adjustment it into resonance with disturbing force. Most often it achieved by changing the elasticity of the DD springs DD using various kinds of design tools. Range of the frequency change $\omega_d = \sqrt{k/m}$ is wide enough to ensure that the system had the ability to adjust and adjustment.

The aim of research is an analytical study of various dynamic dampers to reduce the problems of pressure fluctuations in pumping systems.

Materials and Methods. When applying DD to reduce pressure fluctuations in the system, such as a piston pump suction line, it is first necessary to determine what constitutes a mass *M*, and what is meant by a spring with stiffness *K*. Under the weight of *M* we mean the mass of liquid in the pipeline. Spring stiffness coefficient *K* corresponds to resilience of the compressible medium in the connected supply tank. Mounted on the pump inlet the gas cap-damper acts as DD, i.e., *k* — coefficient resilience of gas into the gas cap-connected damper and *m* — mass of the liquid contained between the pump inlet and place the damper settings. This disturbing force $P_0 \sin \omega t$ is applied to masses *M* and *m*, which, however, does not deprive of fairness the general considerations on the work of DD. The introduction of such an analogy simplifies greatly the analysis of the damper operation in these conditions and shows an important role not only of the resonance volumes included in the work of tanks, but also the mass of the liquid separated by installed damper in the pipeline.

As applied to the suction conduit masses of *M* and *m* can be written as follows:

$$M = F\rho_f (L - \ell),$$

where L, ℓ — the full length of the pipeline and the length of its section from the entrance to the pump to place the damper places, respectively;

F — square of the living section of the pipeline;

ρ_f — fluid density.

Calculation of vibration damper frequency in such conditions will look like:

$$\omega_d = \sqrt{k/m}\left(\frac{P_0 nF}{\rho_f \ell V_\mathrm{d}}\right)^{1/2},$$

where P_0 — the pressure in the damper;

n — adiabatic index;

F — square of the living section of the pipeline;

V_d — volume of gas in the damper (compressible volume).

Let us consider two types of pumps: the single-piston and three-piston ones. Since the frequency of the disturbance under these conditions is $\omega_\mathrm{I} = 2\pi f^1 = 10\pi$ for single-piston pump and $\omega_\mathrm{III} = 60\pi$ — for three-piston pump, the conditions for the calculation of vibration damper frequency will take the form:

$$\omega^i = \left(\frac{P_0 nF}{\rho_f \ell V_d}\right)^{1/2}.$$

Hence, the structural characteristics of the "gas cap" are defined, respectively, for the two cases:

$$(\ell V_d)_\mathrm{I} = \frac{P_0 nF}{\omega_\mathrm{I}^2 \rho_f} = \frac{P_0 nF}{100\pi\rho_f},$$

$$(\ell V_d)_\mathrm{III} = \frac{P_0 nF}{3600\pi^2 \rho_f}.$$

It should be noted that the choice of the place of installation of the damper, i.e. length ℓ is limited to condition $m << M$, and therefore the damper must be installed as close as possible to the

 ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

entrance to the pump (usually $\ell \le D_p$, where D_p — diameter of the pipe). Accordingly, the parameters of the gas chamber of the damper, made of pipes of ∅70×7.0 and installed on the pipeline of $\varnothing_p$45×1.5 will be equal to

$$(V_d)_{\mathrm{I}} = \frac{P_0 n d_p^2}{4 \cdot 100 \pi \rho_f} = 2.43 \cdot 10^{-4} \frac{1}{\ell}, \ (V_d)_{\mathrm{III}} = 6.76 \cdot 10^{-6} \frac{1}{\ell},$$

and "gas cap" length, respectively, is equal to

$$(H_d)_{\mathrm{I}} = \frac{(V_d)_{\mathrm{I}}}{F_d} = \frac{0.0987}{\ell}, \ (H_d)_{\mathrm{III}} = \frac{2.74 \cdot 10^{-3}}{\ell}.$$

As a method of struggle with significant pressure fluctuation in the suction line can also be the installation of DD. Diagram is shown in Fig. 2.

DD that presented in Fig. 2 is a resonance absorber, i.e., it only works in the field of resonant frequencies and frequencies close to them. In the case when the disturbance frequency becomes significantly different from the resonant, the liquid oscillations in a pumping system become similar to oscillation in a system without installed DD. To avoid such effects, spring elements must be installed in such a way as to be able to change their elastic properties by changing the degree of tension (which changes the resonant frequency of the damper).

The best design for this purpose is consistent installation of elastic-inertial elements (piston-spring) in one housing. This vibrational chain in the case of a large number of components is a high-pass filter, i.e., it does not pass a frequency disturbance $\omega \ge 2\sqrt{k/m}$.

As a model system we consider DD comprising three oscillators with masses m, $2m$, $3m$ DD oscillation frequency in this case can be calculated as $\omega_s = 0.283\sqrt{k/m}$. According to [2] it will reduce the frequency of natural oscillations of DD in comparison with the structure shown in Fig. 2 approximately 3 times that essential to quench the low-frequency oscillations.

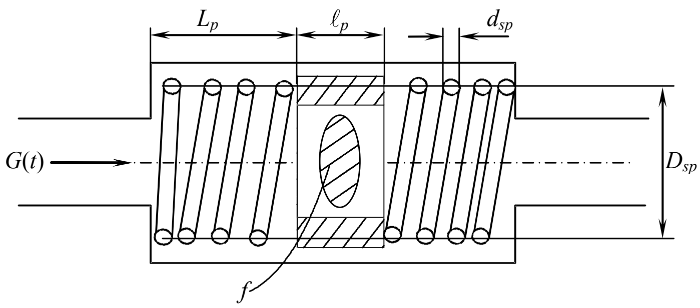

Fig. 2. Schematic diagram of the DD with mechanical elastic elements

We calculate the parameters of the DD that is installed in a horizontal pipeline of $\varnothing_p$45×1.5.

Initial data for the calculation: $\sigma = 7 \cdot 10^{10}$ Pa — shift modulus of elasticity of a spring made of bronze wire; $\rho_p = 7800$ kg/m^3 — density of the pistons (pistons are made of stainless steel 12Kh18N9T); $\rho_f = 808$ kg/m^3 — liquid density (N); $G = 0.833$ kg/s — nitrogen consumption (average); $\omega_{\mathrm{I}} = 5$ rad/s — the frequencies of oscillations of piston pump groups for single-piston system; $\omega_{\mathrm{III}} = 30$ rad/s — the frequencies of oscillations piston pump group for three-piston system; $D_d^{in} = 42$ mm — the internal diameter of DD; $D_p^{out} = 41.6$ mm — the outer diameter of the piston; $R_{sp} = D_{sp}/2 \cong 19$ mm — radius of spring coils (at its diameter at its $D_{sp} = 38$ mm); $d_{sp} = 2$ mm — spring diameter (wire).

The stiffness of the springs, in the system in accordance with [4]

$$k = \frac{\sigma d_{sp}^4}{8nD_{sp}^3},$$

where n — the number of spring coils.

The mass of the piston according to [4], is determined as

$$m = \ell_p (F_p - G_p)\rho_p,$$

where, taking the hole square for the passage f from the condition $d_p / D_p = 0.7$, we obtain an expression for the mass of the piston:

$$m = \ell_p (F_p - G_p)\rho_p = 5.407 \cdot \ell_p,$$

at the value of the piston diameter $d_p = 29$ mm.

Then the expression for the natural frequency of the DD oscillations takes the form:

$$\omega_s = 0.283\sqrt{\frac{k}{m}} = 0.283\sqrt{\frac{\sigma d_{sp}^4}{8nD_{sp}^3 \cdot 5.407\ell_p}} = 5.37\sqrt{\frac{1}{n\ell_p}}.$$

Taking into account that the frequencies of perturbing forces for two types of pumps are 5π and 30π rad/s, respectively, define the dimensions of the springs and pistons included in DD. For single piston pump —

$n \cdot \ell_p = \left(\frac{5.37}{5\pi}\right)^2 = 0.1169$ m; for three piston pump — $n \cdot \ell_p = \left(\frac{5.37}{30\pi}\right)^2 = 3.25 \cdot 10^{-3}$ m.

By varying the number of coils in the spring, we determine the necessary length of the pistons. Number of coils of wire for single piston pump will take to 10, for three piston pump — 2. We obtain the following values:

1) $n_{\mathrm{I}} = 10$ $\quad \ell_p = 0.0117$ mm, $2\ell_p = 23.4$ mm, $3\ell_p = 35.1$ mm.

2) $n_{\mathrm{III}} = 2$ $\quad \ell_p = 1.6$ mm, $2\ell_p = 3.25$ mm, $3\ell_p = 4.9$ mm.

The calculations make it possible to construct a dynamic vibration damper as for single piston pump, as well as three piston pump.

Conclusions. Here are the methods of calculation of dynamic damper of fluid pressure oscillations in hydraulic and mechanical pumps. Algorithms of calculations brought to engineering applications and implemented in the production process in Kislorodmash and Krioprom plants (Odessa, Ukraine).

It is shown that the pressure variations in the suction pipe can be reduced not only by setting the “gas cap”-damper, but also by dynamic damper of mechanical type. Calculations show that the use of mechanical DD suitably in high frequency pumps, and with increasing frequency of the pump in 6 times, winning in the dimensions of the damper in 3.5 times. The presented method of calculation will allow expand the range of applicability of DD in hydraulic systems.

Література

1. Den Hartog, J.P. Mechanical vibrations / J.P. Den Hartog. — New York: Dover Publications, 1985. — 436 p.
2. Коренев, Б.Г. Динамические гасители колебаний. Теория и технические приложения / Б.Г. Коренев. — М.: Наука, 1988. — 304 с.
3. Королев, А.В. Скорость звука в трубопроводе с поверхностным кипением на стенках / А.В. Королев // Тр. Одес. политехн. ун-та. — 2006. — Вып. 1(25). — С. 184 — 186.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

4. Кочетов, В.Т. Сопротивление материалов / В.Т. Кочетов, М.В. Кочетов, А.Д. Павленко. — 3-е изд., перераб. и доп. — СПб.: БХВ-Петербург, 2004. — 533 с.

References

1. Den Hartog, J.P. (1985). *Mechanical Vibrations*. New York: Dover Publications.
2. Korenev, B.G., & Reznikov, L.M. (1993). *Dynamic Vibration Absorbers: Theory and Technical Applications*. New York: Wiley.
3. Korolyov, A.V. (2006). Sound speed in conduit with superficial boiling on walls. *Odes'kyi Politechnichnyi Universytet. Pratsi*, 1, 184 — 186.
4. Kochetov, V.T., Pavlenko, A.D., & Kochetov, M.V. (2004). *Materials Strength* (3rd Ed.). Saint Petersburg: BHV-Petersburg.

Received January 12, 2016
Accepted March 28, 2016