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REFINED MODEL OF PASSIVE BRANCH DAMPER OF PRESSURE FLUCTUATIONS

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This paper presents an analysis of the influence of the kind of a friction model on the dimensioning of a branch pressure fluctuation damper. The mathematical model of the branch damper is defined by determining the damper input impedance and finding its minimum corresponding to the maximum effectiveness in reducing pressure fluctuations. Three kinds of friction for the oscillatory flow in the damper, i.e. a lossless line, steady friction and a nonstationary friction model, are considered. Experimental studies confirmed that the use of the nonstationary friction model in the calculation of branch damper length ensures the highest effectiveness in reducing the amplitude of pressure fluctuations characterized by a given frequency.

Keywords: pressure fluctuations, damper, noise

1. Introduction

Hydrostatic drive systems have their well-known advantages, but their main disadvantage is that they generate much noise which may disqualify such kind of drive when the increasingly more stringent noise emission standards dictated by ergonomic considerations are exceeded. For this reason, a properly designed hydrostatic drive system should not only have the expected static and dynamic properties, but also emit as little noise as possible. Directive 98/37/WE includes a general recommendation that a machine should be designed in such a way that the hazard arising from the emission of the noise generated by it is reduced (preferably at the noise generation source) to the lowest level possible owing to the technological progress and the available means of noise reduction. The directive also requires that information about the noise at the operator workstation be included in the machine documentation. The following should be specified: the equivalent sound pressure level, the instantaneous peak acoustic pressure and the acoustic power level. Noise in a hydraulic system can be generated in two ways:

- directly – the noise source (e.g. the impeller of a fan in the electric motor driving the pump) produces changes in pressure in the surrounding air,
- indirectly – time-variable forces make the components of a hydraulic system vibrate. The vibration of the surfaces of the components results in noise emission.

Indirectly generated noise is the principal noise in hydraulic systems. Changeable forces acting on the hydraulic system components arise as a result of:

- pressure fluctuations (Tijsseling, 1996),

- mechanical connection of the system components through conduits and the common mounting. A single component (e.g. a valve) may vibrate as a result of the action of the fluid, causing the vibration of the components connected with it.

One of the major noise sources in a hydraulic system is the unstable operation of the pressure relief valves due to, among other things, external excitations produced by vibration of the machine frame or the feeder cover to which the pressure relief valves are often mounted. One should note that vibrations may arise in the resonant region of the element which controls the valve. Therefore, the coincidence of the frequency range of, e.g., the foundation and that of the hydraulic valve controlling element should be avoided (Stosiak, 2012).

Experimental studies aimed at locating and identifying vibration and noise sources must be carried out in order to effectively eliminate annoying noise. Energy measuring methods are particularly suitable for locating noise sourceswhen diagnosing the acoustic condition of hydraulic machines and equipment (Kollek *et al.*, 2001). For locating noise sources, Osiński and Kollek (2013) recommend using the acoustic intensity method (AIM) with a two-microphone acoustic probe whereby a map of noise intensity around the investigated equipment can be obtained and the loudest places can be indicated.

The causes of noisiness in the hydraulic system can be divided into mechanical causes and hydraulic causes. The group of mechanical causes includes workmanship and assembly faults, excessive clearances in all moving joints, unbalanced rotating parts and so on. The main hydraulic causes are: cavitation phenomena (Kollek *et al.*, 2007), forcing pressure fluctuations and working liquid pressure surges in the pump or displacement motor chambers. In a properly designed hydraulic system, cavitation should not occur while the occurrence of working liquid pressure surges largely depends on the type of the pump. Axial multiplunger pumps are the noisiest while vane pumps and internal gear pumps are the most silent-running. The research so far has identified pressure fluctuations and the resulting vibrations as the principal causes of noise generation in hydraulic systems, see Mikota and Manhartsgruber (2003), Kudźma (2001, 2006, 2012), Wacker (1985). Thus by reducing pressure fluctuations, one can reduce the noisiness of the individual system components and thereby prolong their service life. One of the effective ways of reducing pressure fluctuations, and so the hydrostatic noise of the drive system, is the use of pressure fluctuation dampers. This paper presents a refined passive branch damper model taking into account nonstationary flow resistance.

2. The supply conduit as a long hydraulic line

Simplifying assumptions, in detail described by Kudźma (2012), are commonly made when deriving equations for the nonstationary flow of a liquid in closed conduits. On such assumptions, the (laminar and turbulent) motion of the liquid is described by the following equation, Kudźma (2012), Zarzycki (1994):

— the equation of motion towards the axis z

$$\frac{\partial v_z}{\partial t} = -\frac{1}{\rho_o}\frac{\partial p}{\partial z} + \nu\frac{1}{r}\frac{\partial}{\partial r}\Big(r\frac{\partial v_z}{\partial r}\Big) + \frac{1}{r}\frac{\partial}{\partial r}\Big(r\nu_t\frac{\partial v_z}{\partial r}\Big) \tag{2.1}$$

— the equation of continuity

$$\frac{\partial p}{\partial t} + \rho_o c_o^2\Big(\frac{\partial v_z}{\partial z} + \frac{\partial v_r}{\partial r} + \frac{v_r}{r}\Big) = 0 \tag{2.2}$$

where: v_z is the instantaneous velocity of the liquid in the conduit in the axial direction, v_r – instantaneous velocity of the liquid in the radial direction, p – instantaneous pressure of the liquid, ν – kinematic coefficient of molecular viscosity, ν_t – kinematic coefficient of turbulent viscosity, ρ_o – steady density of the liquid, z – axial coordinate of the conduit, r – radial coordinate of the conduit, c_o – velocity of pressure wave propagation, t – time.

In the case of turbulent motion, v_z and p are quantities averaged in accordance with the Reynolds rules. By integrating equations (2.1) and (2.2) over the conduit cross section, one gets a system of equations which can be presented as follows, see Kudźma (2012), Zarzycki (1994), Zarzycki *et al.* (2007)

$$\rho_o \frac{\partial v(z,t)}{\partial t} + \frac{\partial p(z,t)}{\partial z} + \frac{2}{R}\tau_w = 0 \qquad \frac{\partial p(z,t)}{\partial t} + \rho_o c_o^2 \frac{\partial v(z,t)}{\partial z} = 0 \tag{2.3}$$

where: $v = v(z,t)$ is the average in the conduit cross section velocity of the liquid, $p = p(z,t)$ – average pressure in the conduit cross section, R – inside radius of the conduit, τ_w – shear stress on the conduit wall. The expression $(2/R)\tau_w$ in equation $(2.3)_1$ represents pressure drop due to friction, per unit length.

In the case of a laminar flow, the most accurate model of hydraulic resistance is the model with variable resistance, which takes into account pressure losses as a function of frequency. Using this model one gets the following expression for impedance Z_0 of a long hydraulic line, see Zielke (1968)

$$Z_0(s) = \frac{\dfrac{\rho_o s}{\pi R^2}}{1 - \dfrac{2J_1\Big(\mathrm{j}R\sqrt{\dfrac{s}{\nu}}\Big)}{\mathrm{j}R\sqrt{\dfrac{s}{\nu}}J_0\Big(\mathrm{j}R\sqrt{\frac{s}{\nu}}\Big)}} \tag{2.4}$$

where: s is the Laplace transformation operator, J_0, J_1 are respectively zero-order and first-order first-type Bessel functions, j – imaginary unit.

By applying the inverse Laplace transformation, Zielke (1968) obtained the following relation for the instantaneous shear stress on the conduit wall

$$\tau_w(t) = \frac{4\mu}{R}v + \frac{2\mu}{R}\int\limits_0^t w(t-u)\frac{\partial v}{\partial t}(u)\,du \tag{2.5}$$

where: $w(t)$ is the weighting function and u is the time in the convolution integral.

The second term in equation (2.5) describes the influence of flow nonstationarity on the shear stress. It is a convolution integral of the instantaneous acceleration of the liquid and weighting function $w(t)$. The instantaneous conduit wall shear stress τ_w can be presented as the sum of quasi-steady quantity τ_{wq} and time-variable quantity τ_{wn}, see Vardy and Brown (2003)

$$\tau_w = \tau_{wq} + \tau_{wn} \tag{2.6}$$

If only the quasi-steady friction model is to be taken into account, the second term in equations (2.5) and (2.6) should be omitted, whereby only $\tau_w = \tau_{wq}$ remains. It should be noted that the quasi-steady frictional resistance models used in calculations of nonstationary states are valid only in the case of slow velocity changes, which applies to low excitation frequencies or small accelerations of the liquid.

2.1. Weighting function

The so-called weighting function, which depends on, among other things, the character of the flow, features significantly in above relation (2.5). For the laminar flow, the relation presented by Zielke (1968) is commonly used, whereas for the turbulent flow there are two main models (also dependent on the Reynolds number) proposed by Vardy and Brown (2003, 2004) and Zarzycki (1994), Zarzycki *et al.* (2007). Since the weighting functions presented by the above authors, especially the ones for the turbulent flow, are complicated and difficult to handle in numerical computations, their approximations are used in practice. From among the weighting function approximating relations proposed in the literature, the relation presented by Urbanowicz and Zarzycki (2012), which through the appropriate scaling of the coefficients can be used for both laminar and turbulent flows, deserves special attention

$$w(\widehat{t}) = \sum_{i=1}^{26} m_i \mathrm{e}^{-n_i \widehat{t}} \tag{2.7}$$

where: $\widehat{t} = \nu t/R^2$ is dimensionless time, and the coefficients m_i, n_i assume the following values (Urbanowicz and Zarzycki, 2012):
$m_1 = 1$; $m_2 = 1$; $m_3 = 1$; $m_4 = 1$; $m_5 = 1$; $m_6 = 2.141$; $m_7 = 4.544$; $m_8 = 7.566$; $m_9 = 11.299$; $m_{10} = 16.531$; $m_{11} = 24.794$; $m_{12} = 36.229$; $m_{13} = 52.576$; $m_{14} = 78.150$; $m_{15} = 113.873$; $m_{16} = 165.353$; $m_{17} = 247.915$; $m_{18} = 369.561$; $m_{19} = 546.456$; $m_{20} = 818.871$; $m_{21} = 1209.771$; $m_{22} = 1770.756$; $m_{23} = 2651.257$; $m_{24} = 3968.686$; $m_{25} = 5789.566$; $m_{26} = 8949.468$;
$n_1 = 26.3744$; $n_2 = 70.8493$; $n_3 = 135.0198$; $n_4 = 218.9216$; $n_5 = 322.5544$; $n_6 = 499.148$; $n_7 = 1072.543$; $n_8 = 2663.013$; $n_9 = 6566.001$; $n_{10} = 15410.459$; $n_{11} = 35414.779$; $n_{12} = 80188.189$; $n_{13} = 177078.960$; $n_{14} = 388697.936$; $n_{15} = 850530.325$; $n_{16} = 1835847.582$; $n_{17} = 3977177.832$; $n_{18} = 8721494.927$; $n_{19} = 19120835.527$; $n_{20} = 42098544.558$; $n_{21} = 92940512.285$; $n_{22} = 203458923.000$; $n_{23} = 445270063.893$; $n_{24} = 985067938.878$; $n_{25} = 2166385706.058$; $n_{26} = 4766167206.672$.

The function can be easily transformed to the Laplace variable domain. Then it assumes the form

$$L[w] = \sum_{i=1}^{26} \frac{m_i}{\widehat{s} + n_i} \tag{2.8}$$

where $\widehat{s}$ is dimensionless operator of the Laplace transformation $\widehat{s} = (R^2/\nu)s$.

The values of the universal coefficients are determined in the following way, see Urbanowicz and Zarzycki (2012)

$$\begin{aligned} &n_{1u} = n_1 - B^*;\ n_{2u} = n_2 - B^*;\ \ldots;\ n_{26u} = n_{26} - B^* \\ &m_{1u} = \frac{m_1}{A^*};\ m_{2u} = \frac{m_2}{A^*};\ \ldots;\ m_{26u} = \frac{m_{26}}{A^*} \end{aligned}$$

where

$$A^* = \sqrt{\frac{1}{4\pi}} \qquad B^* = \frac{\mathrm{Re}^\kappa}{12.86} = \frac{2320^\kappa}{12.86} \qquad \kappa = \log\frac{15.29}{\mathrm{Re}^{0.0567}} = \log\frac{15.29}{2320^{0.0567}}$$

The values of the universal coefficients of the laminar-turbulent weighting function are necessary to determine the current shape of the weighting function used in numerical computations. Thus the function defined by the universal coefficients

$$L[w] = \sum_{i=1}^{26} \frac{m_{iu}}{\widehat{s} + n_{iu}} \tag{2.9}$$

sufficiently well represents the nonstationary frictional loss model for both the laminar flow and the turbulent flow, provided that the Reynolds number is determined earlier.

Performing the Laplace transformation on equations (2.1) and (2.2) for zero initial conditions and then integrating the equations relative to the variable z with the limits of 0-L (L is the length of the hydraulic line) one gets a matrix transition function for a long hydraulic line. Assuming a harmonic excitation, the function can be presented in the following form, see Kudźma *et al.* (2002), Zarzycki (1994), Zarzycki *et al.* (2007)

$$\begin{bmatrix} p_1 \\ q_1 \end{bmatrix} = H(\mathrm{j}\omega) \begin{bmatrix} p_2 \\ q_2 \end{bmatrix} \tag{2.10}$$

where: $H(\mathrm{j}\omega)$ is the matrix transition function, p_1, p_2, q_1, q_2 are harmonically variable deviations from the mean value of the pressure and the rate of flow respectively. When at the harmonic excitation a quasi-steady state is considered using the model with distributed parameters, the transmittance matrix assumes the following form, see Kudźma (2012) and Zarzycki (1994)

$$H(\mathrm{j}\omega) = \begin{bmatrix} h_{11} & h_{12} \\ h_{21} & h_{22} \end{bmatrix} \tag{2.11}$$

where the particular matrix terms are expressed by the relations

$$\begin{aligned} h_{11} &= \cosh(T\psi_z \mathrm{j}\omega) & h_{12} &= Z_c\psi_z \sinh(T\psi_z \mathrm{j}\omega) \\ h_{21} &= \frac{1}{Z_c\psi_z}\sinh(T\psi_z \mathrm{j}\omega) & h_{22} &= \cosh(T\psi_z \mathrm{j}\omega) \end{aligned} \tag{2.12}$$

where: $Z_c = \rho c_o/(\pi R^2)$ is the characteristic impedance of the conduit, $T = L/c_o$ – time constant. ψ_z – operator defining the influence of viscosity (a viscosity function) expressed by

$$\psi_z = \frac{\psi}{\mathrm{j}\Omega} \qquad \psi = \varepsilon + \mathrm{j}\delta \tag{2.13}$$

ε – coefficient of sinusoidal pressure wave amplitude damping, δ relates to wave phase velocity, j is the imaginary unit

$$\begin{aligned} \varepsilon &= \sqrt{\frac{-(\Omega^2 + 2b_2\Omega) + \sqrt{(\Omega^2 + 2b_2\Omega)^2 + (2b_1\Omega)^2}}{2}} \\ \delta &= \sqrt{\frac{(\Omega^2 + 2b_2\Omega) + \sqrt{(\Omega^2 + 2b_2\Omega)^2 + (2b_1\Omega)^2}}{2}} \end{aligned} \tag{2.14}$$

and

$$b_1 = \Re\Big(\frac{1}{2}R_o\frac{\pi R^4}{\mu} + 2\mathrm{j}\Omega L[w]\Big) \qquad b_2 = \Im\Big(\frac{1}{2}R_o\frac{\pi R^4}{\mu} + 2\mathrm{j}\Omega L[w]\Big) \tag{2.15}$$

where: $L[w]$ is simple Laplace transformation of the weighting function, $\Omega = \omega R^2/\nu$ – dimensionless frequency, ω – angular frequency of excitations, R_o – constant resistance calculated from the Darcy-Weisbach formula

$$R_o = \frac{\lambda \mathrm{Re}\mu}{8\pi R^4} \tag{2.16}$$

λ – dimensionless coefficient of linear frictional losses, Re – Reynolds number, μ – dynamic viscosity of the liquid. When only the quasi-steadyfrictional losses are taken into account, relations (2.14) are reduced to the form, see Zarzycki (1994)

$$\varepsilon = \sqrt{\frac{1}{2}}\Omega\sqrt{-1 + \sqrt{1 + \Big(\frac{R_o}{\Omega}\Big)^2}} \qquad \delta = \sqrt{\frac{1}{2}}\Omega\sqrt{1 + \sqrt{1 + \Big(\frac{R_o}{\Omega}\Big)^2}} \tag{2.17}$$

3. Branch damper dimensioning based on long hydraulic line equations

A branch damper is a conduit of proper length inserted at the right angle into the main conduit and stoppered at its end. The principle of operation of the branch damper is based on the interference of the pressure wave generated by an excitation with the pressure wave bounced off the damper and propagating in the opposite direction. Thus the branch damper dimensioning problem comes down to determining its length L_0 depending on the frequency of the excitations which are to be damped. There is a prevailing view that this damper is a narrow-band damper and its effectiveness in reducing pressure fluctuations is limited to one frequency – the damper resonance frequency, see Kudźma (2001, 2006), Wacker (1985), Kollek and Kudźma (1997), Mikota and Manhartsgruber (2003). It is assumed that the damping effectiveness sharply diminishes already at slight deviations from the resonance frequency. However, the above analyses were based on the ideal liquid model (not representative of the real conditions) and their conclusions have not always been corroborated in operational practice (Kudźma, 2001, 2006). A schematic of the branch damper hydraulic system is shown in Fig. 1.

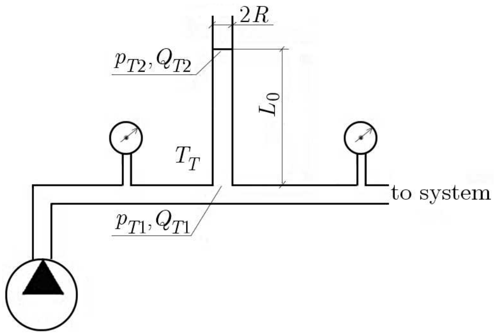

Fig. 1. Schematic of a branch damper hydraulic system

A branch damper mathematical model is defined by determining the damper input impedance and finding its minimum corresponding to the maximum pressure fluctuation reduction effectiveness. Three cases: lossless oscillatory flow, flow with quasi-steady losses and the case with the nonstationary friction model are considered. In order to carry out an analysis of the hydraulic system incorporating a branch damper one must first determine the operational impedance $Z_T(s) = p_{T1}(s)/Q_{T1}(s)$ in the supply station T_T, where $p_T(s)$ and $Q_T(s)$ are the Laplace transforms of the deviations of the pressure p_T and flow rate Q_T. Thus one should select an impedance value which ensures the minimum variation of pressure p_T. Treating the branch damper with length L_0 as a long hydraulic line, for a harmonic excitation one can write (consistently with relations (2.12) and (2.13)) the following

$$\begin{bmatrix} p_{T1} \\ Q_{T1} \end{bmatrix} = \begin{bmatrix} \cosh(T\varPsi_z \mathrm{j}\omega) & Z_c\varPsi_z \sinh(T\varPsi_z \mathrm{j}\omega) \\ \dfrac{1}{Z_c\varPsi_z}\sinh(T\varPsi_z \mathrm{j}\omega) & \cosh(T\varPsi_z \mathrm{j}\omega) \end{bmatrix} \begin{bmatrix} p_{T2} \\ Q_{T2} \end{bmatrix} \tag{3.1}$$

Since the flow is blocked at the damper end, $Q_{T2} = 0$, the impedance Z_d at the place where the branch is connected, according to equation (3.1), has the form

$$Z_d = \frac{\rho_o c_o \varPsi_z}{\pi R^2 \tanh \frac{L_0 \varPsi_z \mathrm{j}\omega}{c_o}} \tag{3.2}$$

When the lossless model is adopted, the viscosity function $\varPsi_z = 1$ should be used in equation (3.2), whereas for the quasi-steady frictional losses one should use relations (2.17). The nonstationary friction model is taken into account through equations (2.13)-(2.17) and substituting $\mathrm{j}\varOmega$ for $\widehat{s}$ (the dimensionless operator of the Laplace transformation) in relation (2.9).

Figure 2 shows an example of how the geometric parameters of the branch pressure fluctuation damper are determined for the basic harmonic of pump 2110 (this type of pump was used for experimental verification) manufactured by the Warsaw Waryński Construction Machinery Plant. The dominant frequency in the pump delivery fluctuation spectrum follows from the relation

$$f_i = \frac{n_p z_t K}{60} \tag{3.3}$$

where: n_p is the rotational speed of the pump shaft [rpm], z_t – number of teeth, K – next number of the harmonic component, $f_1 = 250\,\mathrm{Hz}$, $z_t = 10$ teeth and pump shaft rotational speed $n_p = 1500\,\mathrm{rpm}^{-1}$. From formula (3.3), after transformations, it follows that the initial damper impedance modulus $|Z_d|$ for the lossless model will be minimal when the following condition is satisfied

$$\frac{\omega_w}{c_o} L_0 = K\pi + \frac{\pi}{2} \tag{3.4}$$

where $K = 0, 1, 2, \ldots$. Using the dependence between the angular frequency ω_w and frequency f_w, and the following expression for pressure wavelength λ_f (Kudźma, 2012)

$$\lambda f = \frac{c_o}{f_w} \tag{3.5}$$

one can determine (from condition (3.5)) length L_0 of the branch damper ensuring the maximum pressure fluctuation amplitude damping for a given frequency f_w as a function of length λ_f of the pressure wave in the pipeline

$$L_0 = \frac{\lambda_f}{4} \tag{3.6}$$

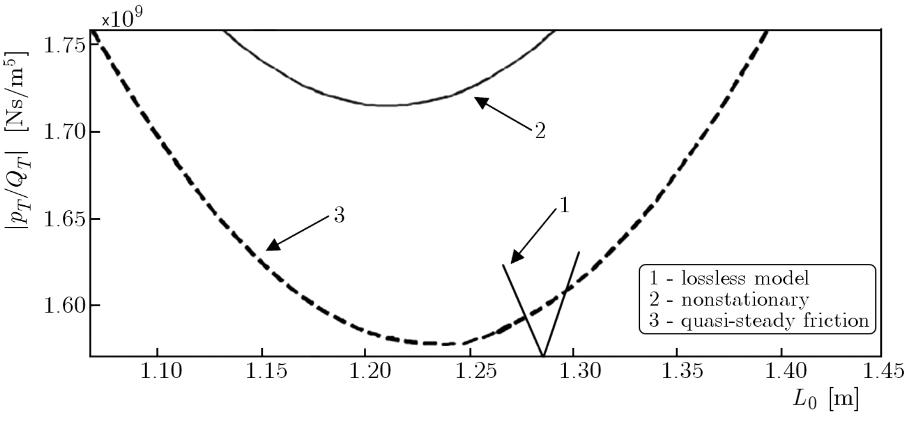

Fig. 2. Modulus of the initial impedance $|Z_d|$ of the branch damper as a function of its length L_0 for different friction models at viscosity $\mu = 30 \cdot 10^{-3}\,\mathrm{Ns/m^2}$, pressure wave propagation velocity $c_{oszt} = 1288\,\mathrm{m/s}$ (acc. to Kudźma (2012)) and $R = 4.5\,\mathrm{mm}$

Numerous numerical studies, corroborated by experiments, indicate that in order to obtain the maximum pressure fluctuation damping for a given excitation frequency f_w, the damper length calculated for the ideal liquid should be shortened by the value of correction ΔL_0 determined assuming the nonstationary friction model and introducing the notion of relative change χ in damper length

$$\chi = \frac{\Delta L_0}{L_0} \tag{3.7}$$

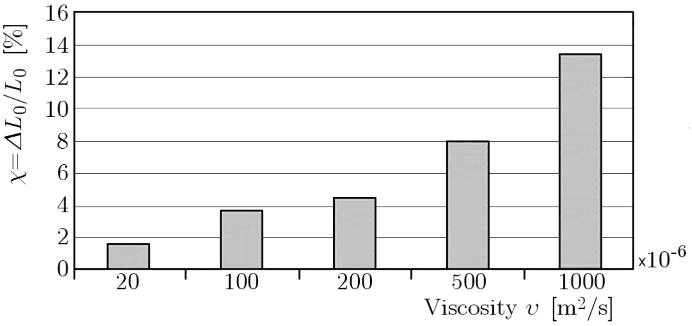

Fig. 3. Relative change χ in the branch damper length versus viscosity υ of the working medium

The numerically determined value of coefficient χ depending on the kinematic viscosity of the oil is shown in Fig. 3.

In real conditions, the optimal length of the branch damper should be calculated from relation

$$L_{0opt} = \frac{\lambda}{4}(1 - \chi) \tag{3.8}$$

4. Experimental verification

Branch damper effectiveness tests and acoustic tests were carried out in the real loader Ł-200 boom lifting gear system incorporating P2C2110C5B26A gear pump made by the Warsaw Waryński Construction Machinery Plant (the manufacturer-installed pump model). Figure 4 shows a schematic of the hydraulic system of the Ł-200 loader boom lifting gear which was placed in a sound chamber (the drive motor and the supply system were outside the chamber) with an insulating power of 50 dB.

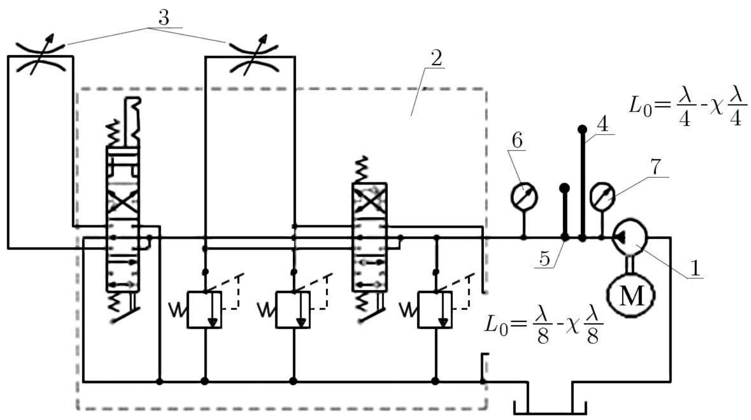

Fig. 4. Schematic of Ł-200 loader boom lifting gear hydraulic system with throttle valves and dampers: 1 – pump, 2 – distributor R1011VF1V, 3 – throttle valve, 4 – branch damper, 5 – branch damper $\lambda/8$, 6, 7 – pressure sensors

A retunable damper, with adjustable length Lo (and so with adjustable natural frequency) was designed in order to experimentally verify the method of determining (selecting a friction model) the optimal length. Figure 5 shows an axial cross section of the investigated damper. The pressure fluctuation amplitude levels p_{T1} in the hydraulic system versus branch damper length L_0 are shown in Fig. 6. The damper whose length $L_{0opt} = 1.24$ m was determined on the basis of the nonstationary friction model was found to be most effective. For the lossless model the damper length was 1.28 m, but the effectiveness of the damper was by 4 dB lower than that of the damper with the nonstationary friction model.

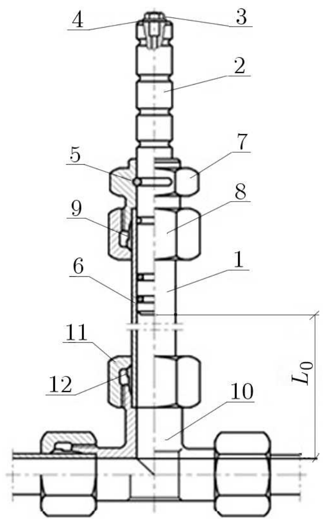

Fig. 5. Branch damper with retunable natural frequency: 1 – branch damper, 2 – piston, 3 – bleeder screw, 4 – copper washer, 5 – cotter pin, 6 – sealing ring, 7 – connector shell, 8 – connecting nut, 9 – cutting ring, 10 – coupling shell, 11 – nut, 12 – cutting ring

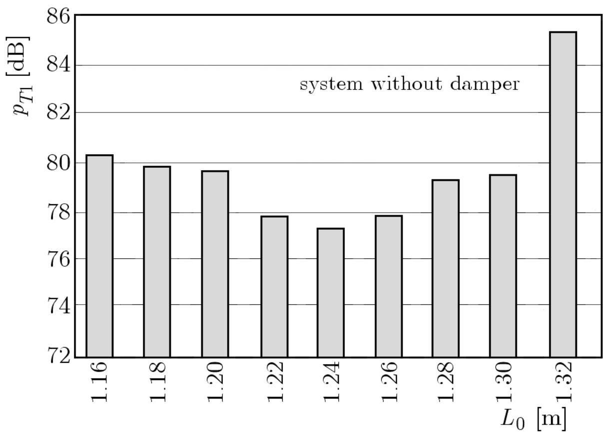

Fig. 6. Pressure fluctuation amplitude levels p_{T1} in the hydraulic system versus the branch damper length L_0 – pump delivery fluctuation first harmonic $f_1 = 250\,\text{Hz}$, average pressure $p_T = 10\,\text{MPa}$

5. Conclusion

The branch damper whose dimensioning comes down to determining its length is effective in reducing amplitudes only for specific frequencies. If in the first approximation the optimal length is assumed in accordance with $L_0 = \lambda_{f1}/4$ (leaving out oscillatory flow resistances in the damper), the pressure fluctuation amplitude damping is obtained for the basic harmonic f1and harmonic $3f_1$, i.e. generally $f_w = 2K - 1$, $K = 1, 2, 3, \ldots$. In order to suppress even harmonics, one should assume damper length $L_0 = \lambda_{f1}/8$ (λ_{f1} – the wavelength for the basic harmonic). In terms of pressure fluctuation effectiveness, the most advantageous solution is to use a double branch damper. When the flow resistances are left out and the optimum length is adopted, the particular pressure fluctuation components are suppressed completely. In real conditions, when the nonstationary friction model is assumed, the optimal length of the branch damper for a given excitation frequency is calculated from relation (3.2). One can determine the optimal branch damper length using simplified relation (3.8) and the data shown in Fig. 3. In this case, the following regularity is observed: the higher the coefficient of viscosity of the working liquid, the shorter (by 2-15%) the optimal length in comparison with the length defined by formula (3.6) for the ideal liquid. When the double branch damper is installed in the outlet port of the pump in the Ł-200 loader boom lifting system, the pressure fluctuation amplitude is reduced several times in the whole range of the excitation frequencies, whereby the total noise (the measure of which is the sound pressure level subject to correction) is reduced by a few to about twenty $dB(A)$, depending on the system load, see Fig. 7.

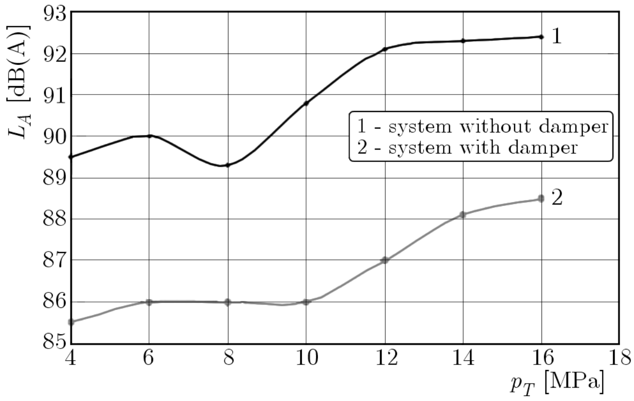

Fig. 7. Corrected sound pressure level $L_A dB(A)$ of Ł-200 loader boom lifting gear hydraulic system with the double branched damper and without damper versus the forcing pressure p_T

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