STUDY OF THE PRESSURE REGULATOR WORK WITH A SPRING-DAMPER SYSTEM APPLIED TO MILKING MACHINE
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ДОСЛІДЖЕННЯ РОБОТИ РЕГУЛЯТОРА ТИСКУ З ПРУЖИННО-ДЕМПФЕРНОЮ СИСТЕМОЮ СТОСОВНО ДО ДОЇЛЬНИХ УСТАНОВОК

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ABSTRACT
A mathematical model of a regulator for the vacuum gauge pressure with the dual mass valve-damper system was studied in the article. The differential equation was solved as well as eventual equations that simulate the valve and load moving depending on the following parameters: amplitude, oscillation of vacuum gauge pressure, load mass, valve diameter, springing of spring, damper environment description. The results of theoretical and experimental research of valve and load moving of vacuum gauge pressure regulator with the dual weight valve-damper was determined in conditions of different pressure and attenuation coefficient and also the characteristic oscillation frequency of the valve and load mass.

РЕЗЮМЕ
У роботі наведено математичну модель регулятора вакуумметричного тиску з двомасовою клапанно-демпферною системою. Розв'язано диференціальне рівняння та кінцеві рівняння, що моделюють переміщення клапана і вантажу залежно від амплітуди, частоти коливання вакуумметричного тиску, ваги вантажу, діаметра клапана, пружності пружини, характеристики демпферного середовища. Наведено результати теоретичних і експериментальних досліджень переміщення клапана і вантажу регулятора вакуумметричного тиску з двомасовою клапанно-демпферною системою за різного тиску і коефіцієнта затухання, а також частоти власних коливань клапана і ваги вантажу.

INTRODUCTION
The stability of the vacuum gauge pressure is one of the basic parameters that provide quality of the cow milk ejection process. This index depends on conditions of regulator operating and must exclude the possible vibrations and resonant phenomena in the vacuum system under valve operation of the regulator and during work of the milking machines. Allowable oscillation of the vacuum gauge pressure must not be more than 2.0 kPa (*ISO 6690:2006, 2007; ASAE EP445.1, 1996*). Stability of the vacuum gauge pressure is provided by both the regulator construction and its descriptions that are formed by the construction parameters. To ensure the technological parameters it is necessary to have a mathematical tool that does possible the simulation of the modes of the vacuum regulator operation.

The stability of the vacuum gauge pressure has been estimated by the researchers group (*Pazzona A. et al, 2003*) depending on the method of pressure regulation in the vacuum hose. A group of researchers note that stabilizing the vacuum by a gravitational type regulator is more dynamic, the time constant is twice smaller of the computer-aided system (CAS). In particular, the typical regulator spends on the average 1.69 seconds on proceeding in stable pressure, while the system of the VSD-controller spends 3.75 seconds (*Pazzona A. et al, 2003*).

The vacuum gauge pressure is regulated by the rotation frequency of the vacuum pump rotor using the PID control digital systems with the amplification factors parameters of the proportion, integral and differential links accordingly: $K_p = 20$, $K_i = 0.05$, $K_D = 0.5$ at the vacuum gauge pressure of 35, 40, 45 kPa. Amplitude of vibrations does not exceed 0.3 kPa, time constant is $\tau = 5$ s, maximal overcontrol is 2 kPa (*Radu R., Petru C., Ioan T., 2013*).

The research result of the vacuum-gauge pressure oscillations in a milking machine vacuum system did not show substantial differences between the regulators with the gravitational and digital control (*Pařilova M., Stadnik L., Ježkova A., Štolc L., 2011; Reinemann D. J., 2005*).

The vacuum-gauge pressure oscillations have been researched depending on: a) configuration of the vacuum and the milk duct systems (the length and diameter of pipelines, other parameters that influence on the pressure loss); b) milk flowrates in the milk pipelines; c) rates of air movement in vacuum pipelines (*Reinemann D.J., Schuring N., Badel R.D., 2007*).

The construction variants of milking machine vacuum regulators (*Vagin Yu T. and other, 2012*) differ in the load mass that is counted on the set vacuum gauge pressure at the appropriate area of valve seat (*Dmytriv V.T., Dmytriv I.V., 2012; Dmytriv V.T., 2015, 2016*). However, the dynamic descriptions of the valve-damper system are uncoordinated with the vacuum power oscillations and speed parameters of the air entering in the milking machine vacuum system. The milking machine caused the pressure oscillation of the milker vacuum system (*Dmytriv V.T., Dmytriv I.V., 2017*). The oscillation amplitude depends on the probability of phase coincidence and frequency of pulsators work.

Dynamic descriptions of the regulator work must provide the smoothing of pressure oscillations.

Therefore, researches on the work of vacuum gauge pressure regulators for the milking systems are up-to-date.

MATERIAL AND METHODS

Development of the mathematical model of vacuum gauge pressure regulators with the dual mass valve-damper system and research of influence of the regulator construction descriptions and technological parameters of a milking machine vacuum system on the dynamic descriptions of pressure adjusting by a regulator were the aim of this research work.

Let us consider the work of the vacuum gauge pressure regulator of spring-gravitational type with a hydraulic damper, which was showed in fig. 1.

The equilibrium of the system is provided (fig.1) when the force is created by the pressure difference that added to the valve (1) mass m_1 and the spring elastic force (2) equals the load mass m_2 (3) and the damper plate (4). Let us consider the work of the vacuum regulator as dual mass system, when the additional shaking force was applied to the valve. This force appeared as a result of increasing the vacuum-gage pressure by the size of Δp_{vp}. The scheme of forces applied to the valve as a result of the shaking forces is shown in fig. 1. Let us name this mode by the dynamic mode of regulator. During the valve 1 upwards movement on the y_1 size and the load 3 on the size of y_2, the spring will get of y_1-y_2 additional deformation.
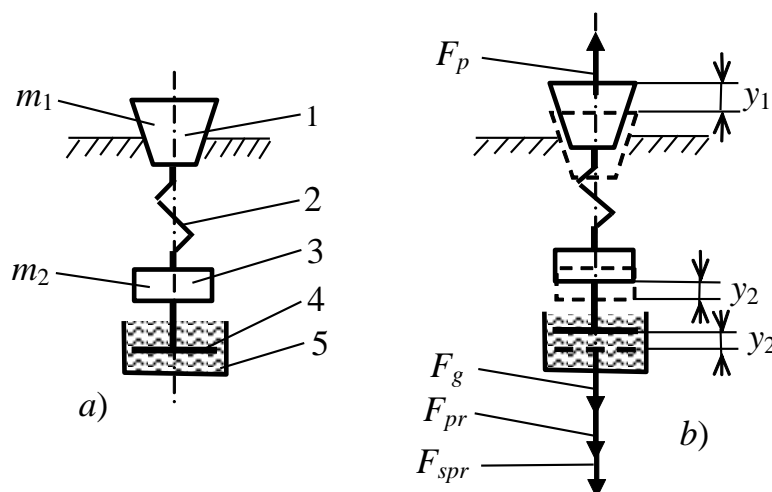
Fig. 1 - Scheme of vacuum gauge pressure regulator
a – equivalent scheme; b – functional scheme of action of forces;
1 - valve; 2 - spring; 3 - load; 4 – plate in a damper environment; 5 - damper environment;
F_p - force of the vacuum gauge pressure; F_{pr} – elasticity force; F_{spr} – the resistance force of damper environment; F_g - the force that is created by mass of regulator movable elements; m_1, m_2 – respectively the weight of the valve and the load with other elements

The system of differential equations of the motion and the load of valve should be written down:

$$\begin{cases} m_1 \cdot \ddot{y}_1 = -K_{pr} \cdot (y_1 - y_2) + \Delta p_{vp} \cdot S_{kl} \cdot f(t) \\ m_2 \cdot \ddot{y}_2 = K_{pr} \cdot (y_1 - y_2) - K_{spr} \cdot \dot{y}_2 \end{cases} , \tag{1}$$

where: K_{spr} – the integrated coefficient of resistance of the damping fluid, [N·s/m];

K_{pr} – the coefficient of elasticity of spring, [N/m];

S_{kl} – the sectional area of valve seat, [m²];

$f(t)$ – characteristic of the applied force changes;

The notation was proposed: $K_1^2 = \dfrac{K_{pr}}{m_1}$, $K_2^2 = \dfrac{K_{pr}}{m_2}$ – square of free oscillations frequency

accordingly the valve and load, [s⁻²]; $2n_2 = \dfrac{K_{spr}}{m_2}$ – coefficient of oscillation attenuation, [s⁻¹]; $h_1 = \dfrac{\Delta p_{vp} \cdot S_{kl}}{m_1}$ –

specific amplitude of the forced oscillation force, [m/s²].

Then, the system of differential equations (1) will be:

$$\begin{cases} \dfrac{d^2 y_1}{dt^2} + K_1^2 \cdot (y_1 - y_2) = h_1 \cdot f(t) \\ \dfrac{d^2 y_2}{dt^2} + 2n_2 \cdot \dfrac{dy_2}{dt} + K_2^2 \cdot (y_2 - y_1) = 0 \end{cases} . \tag{2}$$

Let the character of change of the vacuum-gage pressure Δp_{vp} meet the dependence (fig. 2) that is analytically described by the next equations:

$$f(t) = \begin{cases} 1, & n \cdot T < t < n \cdot T + \tau \\ 0, & n \cdot T + \tau < t < (n+1) \cdot T \end{cases} , \tag{3}$$

where: $n = [\tau/T]$ – aliquot of number τ/T .
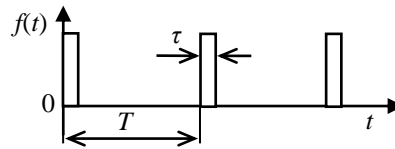
Fig. 2 - Impulse receiving character at vacuum gauge pressure changing Δp_{vp}
τ – duration of presence of vacuum-gage pressure impulse; T – impulse receiving period

After two differentiations of the second equation of the system (2), we shall get:

$$\dfrac{d^2 y_1}{dt^2} = \dfrac{1}{K_2^2} \cdot \left(\dfrac{d^4 y_2}{dt^4} + K_2^2 \cdot \dfrac{d^2 y_2}{dt^2} + 2n_2 \cdot \dfrac{d^3 y_2}{dt^3} \right) . \tag{4}$$

Next, we will put the obtained equation (4) in the first equation of the system (2):

$$\dfrac{d^4 y_2}{dt^4} + 2n_2 \cdot \dfrac{d^3 y_2}{dt^3} + \left(K_1^2 + K_2^2\right) \cdot \dfrac{d^2 y_2}{dt^2} + K_1^2 \cdot 2n_2 \cdot \dfrac{dy_2}{dt} = K_2^2 \cdot h_1 \cdot f(t) . \tag{5}$$

The characteristic equation that fits the homogeneous equation (5) looks like:

$$\lambda^4 + 2n_2 \cdot \lambda^3 + \left(K_1^2 + K_2^2\right) \cdot \lambda^2 + K_1^2 \cdot 2n_2 \cdot \lambda = 0 . \tag{6}$$

The equation (6) is rewritten in the following way:

$$\lambda \cdot \left(\lambda^3 + 2n_2 \cdot \lambda^2 + \left(K_1^2 + K_2^2\right) \cdot \lambda + K_1^2 \cdot 2n_2\right) = 0 . \tag{7}$$

One root of the equation (7) will be $\lambda_0 = 0$. Other roots will be obtained after solving the cube equation, with preliminary defining a discriminant:

$$D = \dfrac{4n_2^2}{9} - \dfrac{K_1^2 + K_2^2}{3} \tag{8}$$

A discriminant can take on two values, $D > 0$ and $D < 0$. To define the roots of the equation, additional determinant should be determined:

$$q = \left(\frac{2n_2}{3}\right)^3 - \frac{2n_2 \cdot \left(K_1^2 + K_2^2\right)}{6} + \frac{2n_2 \cdot K_1^2}{2} \qquad (9)$$

The analysis of previous calculations shows that the difference of values of the expressions (8) and (9) is $D - q < 0$. Then for $D > 0$ it is necessary that the squares sum values of valve and load free oscillation frequency be below the oscillation attenuation coefficient, but a value of q determinant will be always higher than the D discriminant.

In this case the solution of equation (7) will be one actual and two complex roots:

$$\lambda_1 = -2 \cdot \text{sgn}(q) \cdot \sqrt{|D|} \cdot ch(\alpha) - \frac{2n_2}{3}$$

$$\lambda_{2,3} = \text{sgn}(q) \cdot \sqrt{|D|} \cdot ch(\alpha) - \frac{2n_2}{3} \pm j \cdot \sqrt{3} \cdot \sqrt{|D|} \cdot sh(\alpha) \qquad (10)$$

where: $\alpha = \frac{1}{3} \cdot Arch\left(\frac{|q|}{|D|^{3/2}}\right) = \frac{1}{3} \ln\left(\frac{q}{\sqrt{D^3}} + \sqrt{\left(\frac{q}{\sqrt{D^3}}\right)^2 - 1}\right) \cdot$

On condition of $D < 0$, the solution of equation (7) will be one actual and two complex roots:

$$\lambda_1 = -2 \cdot \text{sgn}(q) \cdot \sqrt{|D|} \cdot sh(\alpha) - \frac{2n_2}{3}$$

$$\lambda_{2,3} = \text{sgn}(q) \cdot \sqrt{|D|} \cdot sh(\alpha) - \frac{2n_2}{3} \pm j \cdot \sqrt{3} \cdot \sqrt{|D|} \cdot ch(\alpha) \qquad (11)$$

where:

$$\alpha = \frac{1}{3} \cdot Arsh\left(\frac{|q|}{|D|^{3/2}}\right) = \frac{1}{3} \ln\left(\frac{q}{\sqrt{D^3}} + \sqrt{\left(\frac{q}{\sqrt{D^3}}\right)^2 + 1}\right) \cdot$$

To better understand the physical process of pressure adjusting we will point the analytical solution of the homogeneous system of equations (2) in view of $y_1 = A \cdot e^{\lambda t}, y_2 = B \cdot e^{\lambda t}$ (*Samoilenko A.M., Kryvosheja S.A., Perestiuk N.A., 1989*).

The analytical solution was put in the first equation of (2) the system. Then we get:

$$e^{\lambda t} \cdot \left(A \cdot \left(\lambda^2 + K_1^2\right) - B \cdot K_1^2\right) = 0 \qquad (12)$$

Taking into account that $B = A \cdot (1 + \lambda^2 / K_1^2)$ from the (12) equation and the values of roots (10) and (11) the general solution of the homogeneous system of differential equations (2) is given bellow:

$$y_{10} = A_0 + A_1 \cdot e^{\lambda_1 t} + A_2 \cdot e^{\lambda_2 t} + A_3 \cdot e^{\lambda_3 t}$$

$$y_{20} = y_{10} + \frac{1}{K_1^2} \cdot \left(\lambda_1^2 \cdot A_1 \cdot e^{\lambda_1 t} + \lambda_2^2 \cdot A_2 \cdot e^{\lambda_2 t} + \lambda_3^2 \cdot A_3 \cdot e^{\lambda_3 t}\right) \qquad (13)$$

The partial decision that satisfies the beginning conditions has been found for $t = 0$, $y_{10}(0) = 0$, $\dot{y}_{10}(0) = 1$, $y_{20}(0) = 0$, $\dot{y}_{20}(0) = 0$. On the basis of expressions (13) the system of algebra equations is formed:

$$\begin{cases} A_0 + A_1 + A_2 + A_3 = 0 \\ \lambda_1 A_1 + \lambda_2 A_2 + \lambda_3 A_3 = 1 \\ \lambda_1^2 A_1 + \lambda_2^2 A_2 + \lambda_3^2 A_3 = 0 \\ \lambda_1^3 A_1 + \lambda_2^3 A_2 + \lambda_3^3 A_3 = -K_1^2 \end{cases} \qquad . \qquad (14)$$

The coefficients of equations (13) were determined from the system of the equations (14) as follows:

$$A_0 = -\frac{\Delta_1 + \Delta_2 + \Delta_3}{\Delta}; \quad A_1 = \frac{\Delta_1}{\Delta}; \quad A_2 = \frac{\Delta_2}{\Delta}; \quad A_3 = \frac{\Delta_3}{\Delta}$$

where: $\Delta = \lambda_1 \cdot \lambda_2 \cdot \lambda_3 \cdot (\lambda_2 - \lambda_1) \cdot (\lambda_3 - \lambda_1) \cdot (\lambda_3 - \lambda_2); \quad \Delta_1 = \lambda_2 \cdot \lambda_3 \cdot (\lambda_3 - \lambda_2) \cdot (\lambda_2 \cdot \lambda_3 - K_1^2)$

$$\Delta_2 = \lambda_1 \cdot \lambda_3 \cdot (\lambda_1 - \lambda_3) \cdot (\lambda_1 \cdot \lambda_3 - K_1^2); \quad \Delta_3 = \lambda_1 \cdot \lambda_2 \cdot (\lambda_2 - \lambda_1) \cdot (\lambda_1 \cdot \lambda_2 - K_1^2).$$

The solution of the equation system (2) is the following:

$$y_1(t) = C_0 + C_1 \cdot e^{\lambda_1 t} + C_2 \cdot e^{\lambda_2 t} + C_3 \cdot e^{\lambda_3 t} + h_1 \cdot \int_0^t y_{10}(t-z) \cdot f(z)dz \tag{15}$$

$$y_2(t) = C_0 + \left(1 + \frac{\lambda_1^2}{K_1^2}\right) \cdot C_1 \cdot e^{\lambda_1 t} + \left(1 + \frac{\lambda_2^2}{K_1^2}\right) \cdot C_2 \cdot e^{\lambda_2 t} + \left(1 + \frac{\lambda_3^2}{K_1^2}\right) \cdot C_3 \cdot e^{\lambda_3 t} + h_1 \cdot \int_0^t y_{20}(t-z) \cdot f(z)dz$$

The constants of solutions (15) C_0, C_1, C_2, C_3 were determined from the initial conditions. If the $t = 0$, $y_1(0) = \dot{y}_1(0) = y_2(0) = \dot{y}_2(0) = 0$, the constants C_0, C_1, C_2, C_3 are zero as well.

The integral constituents of (16) equations according to the solution are the following:

$$y_1(z) = h_1 \int_0^t y_{10}(t-z) \cdot dz = h_1 \cdot \left(A_0 \cdot z - \frac{A_1}{\lambda_1} \cdot e^{\lambda_1(t-z)} - \frac{A_2}{\lambda_2} \cdot e^{\lambda_2(t-z)} - \frac{A_3}{\lambda_3} \cdot e^{\lambda_3(t-z)}\right). \tag{16}$$

$$y_2(z) = H_1(z) - \frac{h_1}{K_1^2} \cdot \left(A_1 \cdot \lambda_1 \cdot e^{\lambda_1(t-z)} + A_2 \cdot \lambda_2 \cdot e^{\lambda_2(t-z)} + A_3 \cdot \lambda_3 \cdot e^{\lambda_3(t-z)}\right)$$

Then, we will write down a (15) decision, taking into account the limitations of the system of (3) the function analytical expression and that in (16) the equations of the $z = t$ are:

$$y_1(t) = \begin{cases} y_1(t) - y_1(nT) + \sum_0^n (y_1(nT+\tau) - y_1(nT)), & nT \leq t \leq nT+\tau \\ \sum_0^n (y_1(nT+\tau) - y_1(nT)), & nT+\tau \leq t \leq (n+1)T \end{cases}, \tag{17}$$

$$y_2(t) = \begin{cases} y_2(t) - y_2(nT) + \sum_0^n (y_2(nT+\tau) - y_2(nT)), & nT \leq t \leq nT+\tau \\ \sum_0^n (y_2(nT+\tau) - y_2(nT)), & nT+\tau \leq t \leq (n+1)T \end{cases}. \tag{18}$$

RESULTS

We studied the regulator valve-damping systems of both the valve and load movement depending on the amplitude and frequency of the vacuum pressure oscillation. Initial data for calculating the square of characteristic oscillation frequency in accordance with the K_1^2 valve and K_2^2 load, $2n_2$ oscillation damper factor (attenuation coefficient), K_{op} resistance coefficient of the damper environment, K_{pr} spring elasticity coefficient were the following: the wire diameter of the regulator springs d_{dr} = 0.0018 [m]; the outer diameter of the spring D_{pr} = 0.021 [m]; the number of spring turns n = 10; elastic shear modulus for steel G_{pr} = 80.5 [GPa]; the diameter of the damper plate D_{pl} = 0.0874 [m]; plate shift in the damper environment x_{pl} = 0.01 [m]; dynamic viscosity of the damper environment μ_{dm} = 0.065 [Pa·s]; weight of the regulator load m_2 = 1.4 [kg]; weight of the regulator valve m_1 = 0.17 [kg].

The results of the calculation were the following: resistance coefficient of the damper environment K_{op} = 0.039 [N·s/m]; the spring elasticity coefficient K_{np} = 1134 [N/m]; specific amplitude of the forced oscillation force h_1 = 1,011 [m/s²]; the square of characteristic oscillation frequency of the valve and the load is K_1^2 = 666.781 [s⁻²] and K_2^2 = 80.966 [s⁻²] respectively; oscillations attenuation coefficient $2n_2$ = 2.786 ·10⁻³ [s⁻¹]. The roots of the solution and equation coefficients: λ_1 = - 2.461·10⁻³; λ_2 = - 1.494·10⁻⁴ + j·27.345; λ_3 = - 1.494·10⁻⁴ - j·27.345; A_0 = 43.995; A_1 = - 43.995; A_2 = - 3.564·10⁻⁷ - j·0.016; A_3 = - 3.564·10⁻⁷ + j·0.016. The numerical values of the discriminant at the given factors is D = - 249.249.

Taking into account the coefficients and roots of the $z = t$ and $t = \tau$ (fig. 2), the values of the (16) equations will be:

$$y_1(t) = 1.011 \cdot \left(43.995 \cdot t - \frac{43.995}{2.461 \cdot 10^{-3}} \cdot e^{-2.461 \cdot 10^{-3} \cdot (\tau - t)} - \frac{(-3.564 \cdot 10^{-7} - j0.016)}{(-1.494 \cdot 10^{-4} + j27.345)} \times \right.$$

$$\left. \times e^{(-1.494 \cdot 10^{-4} + j27.345) \cdot (\tau - t)} - \frac{(-3.564 \cdot 10^{-7} + j0.016)}{(-1.494 \cdot 10^{-4} - j27.345)} \cdot e^{(-1.494 \cdot 10^{-4} - j27.345) \cdot (\tau - t)} \right),$$

(19)

$$y_2(t) = y_1(t) - \frac{1.011}{666.781} \cdot \left(43.995 \cdot 2.461 \cdot 10^{-3} \cdot e^{-2.461 \cdot 10^{-3} \cdot (\tau - t)} - (3.564 \cdot 10^{-7} + j0.016) \times \right.$$

$$\times (-1.494 \cdot 10^{-4} + j27.345) \cdot e^{(-1.494 \cdot 10^{-4} + j27.345) \cdot (\tau - t)} +$$

$$\left. + (-3.564 \cdot 10^{-7} + j0.016) \cdot (-1.494 \cdot 10^{-4} - j27.345) \cdot e^{(-1.494 \cdot 10^{-4} - j27.345) \cdot (\tau - t)} \right)$$

Example of regulator pressure valve and load oscillation for the above-mentioned construction and technological parameters of 2.5 [kPa] pressure oscillation and its duration of 0.25 [sec] is shown in fig. 3. The maximum movement of the regulator valve for the 45-50 [kPa] of vacuum pressure and 1.0-1.4 [kg] load weight is shown in fig. 4.

Analysis of the regulator movement of the valve-damping system (fig. 3) by the p = 48 [kPa] vacuum pressure in the milking machines vacuum pipeline and the Δp_{vp} = 2.5 [kPa] permissible oscillations in vacuum pressure and the total weight of the load $m = m_1 + m_2 = 1.57$ [kg] shows that the maximum movement of the valve is $y_1(0.112$ [s]) = 3.093 [mm] and the total duration of the open state of the valve is t_{sum} = 0.225 [s]. Re-raising of the valve to a height $y_1(0.315$ [s]) = 0.5292 [mm] lasts $\sum t$ = 0.084 [s]. The τ = 0,225 [s] total duration of the pressure impulse is Δp_{vp} = 2.5 [kPa]; that exceeds the working vacuum gauge pressure of p = 48 kPa. Load has a single movement for $y_2(0.08-0.13$ [s]) = 0.394 [mm] height lasting t_{sum} = 0.225 [s]. Load is in a static state within the damping environment until next pressure impulse.
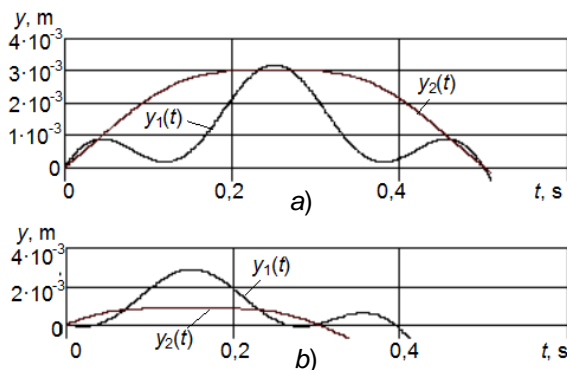
Fig. 3 - Oscillation of the regulator valve-damping system by impulse of vacuum pressure of Δp_{vp} = 2.5 kPa and τ of its duration:

$a - \tau$ = 0.25 s; $b - \tau$ = 0.15 s; $y_1(t)$ – valve oscillations;
$y_2(t)$ – load oscillations in damping environment
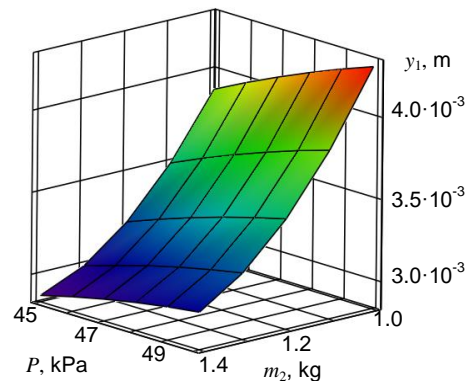
Fig. 4 - The maximum valve movement y_1 of the pressure regulator with spring-damper system depending on the p vacuum gauge pressure and the load mass m_2 for the impulse duration of τ = 0.25 [s]

For the duration of the impulse vacuum pressure of τ = 0.112 [s] the movement character of the valve and load has a single oscillation which is equal to the impulse duration. The maximum rise of the valve is $y_1(0.056$ [s]) = 1.329 [mm], the maximum movement of load in the damping environment is insignificant – $y_2(0.056$ [s]) = 3.101·10^{-2} [mm].

For the load weight of m_2 = 10 [N] and impulse pressure duration of τ = 0.15 [s], the maximum valve movement is up to $y_1(t)$ = 2.783 [mm].

To confirm the results of theoretical studies the planned experiment was made considering the following factors: the K_2^2 square of characteristic oscillation frequency of the load mass and the oscillation attenuation coefficient $2n_2$. The square of characteristic oscillation frequency of the load mass was changed within K_2^2 = 80.966...174.927 [s^{-2}] according to the limits of the m_2 = 14…10.8 [N] of the weight load change. The oscillation attenuation coefficient was within $2n_2$ = 1.38·10^{-3}…1.789·10^{-3} [s^{-1}].

General view of the laboratory setup for the study of vacuum-gage pressure regulators is shown in fig. 5.

The graphical representation of the experimental results in a three-dimensional model view is described by the regression equation (20) (see fig.6).
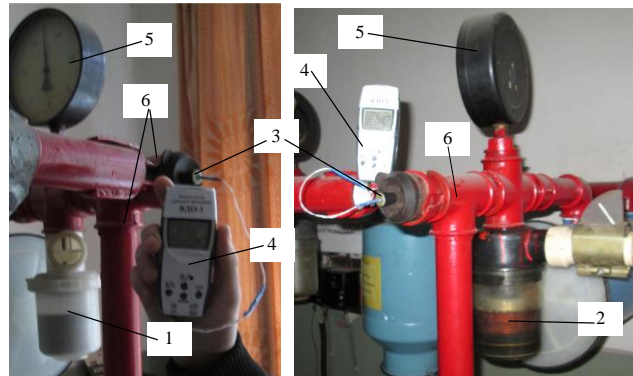
Fig. 5 - General view of the laboratory setup for the study of vacuum-gage pressure regulators
1, 2 – vacuum pressure regulators; 3 – vacuum-gage pressure sensor; 4 – vacuum analyser; 5 – vacuum gauge; 6 – vacuum pipeline

$$y_1 = 1.1147 + 0.0351 \cdot K_2^2 + 4350.5645 \cdot 2n_2 + 6.484 \cdot 10^{-5} \cdot K_2^4 - 0.293 \cdot K_2^2 \cdot 2n_2 - 8.0338 \cdot 10^5 \cdot 2n_2^2. \quad (20)$$
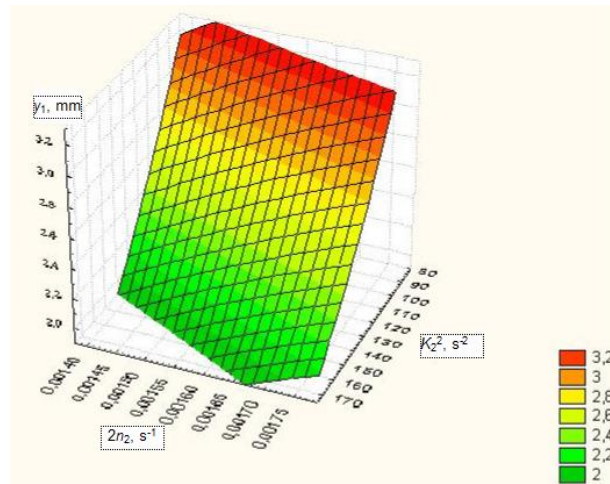
Fig. 6 – The maximum valve movement y_1 of the vacuum regulator with the spring-damper system depending on the $2n_2$ oscillation attenuation coefficient and the K_2^2 square of the natural oscillations frequency of the regulator load

For the square of characteristic oscillation frequency of K_2^2 = 104.956 [s^{-2}] valve movement amplitude is y_1 = 3.3 [mm] for the vacuum-gage pressure of p = 50 [kPa]. For the p = 45 [kPa] the oscillation parameters are K_2^2 = 131.195 [s^{-2}], y_1 = 2.8 [mm]. The reduction of the elasticity coefficient causes the lessening of the square of characteristic oscillation frequency. For the square of characteristic oscillation frequency of K_2^2 = 80.966 [s^{-2}] the amplitude of the valve movement is y_1 = 3.1 [mm] for the vacuum pressure of p = 50 [kPa]. For the p = 45 [kPa] the oscillation parameters are the following K_2^2 = 101.208 [s^{-2}], y_1 = 2.7 [mm].

CONCLUSIONS

The analysis of study results shows that the reduction of the square of characteristic oscillation frequency of the regulator load mass increases the amplitude of the regulator valve oscillation. The increase of the spring elasticity coefficient leads to increasing the square of characteristic oscillation frequency of the regulator load weight, and that increases the valve movement. Increase of the load mass causes the reducing of the valve movement and the lifting repeatability (oscillation) of the regulator valve was also increased. The lifting repeatability (oscillation) of the regulator valve was increased with the increase of the duration of vacuum-gage pressure impulse.

Increase of the load mass reduces the height of valve lifting and the air supply into the vacuum system of the milking machine. Rising the vacuum pressure up to 50 [kPa] increases the valve movement and the repeatability of its opening.

The oscillation of the regulator valve with spring-damper system of the valve is damping out. The amplitude of the single oscillation depends on the load mass and duration of the vacuum-gage pressure impulse.

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