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ACTIVE VIBRATION REDUCTION OPERATOR'S SEAT WITH THE USE OF CONTROLLED SINGLE-ACTING PNEUMATIC ACTUATOR

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Abstract

Mobile heavy machines produce vibrations with low natural frequencies. Because they tend to ride at low speeds, excitations due to road roughness excite low frequency vibrations, which can be reduced by active or semi active methods only. Under conditions of low frequency vibrations, energy dissipation in tires will reduce the vibration intensity in a minor degree only. In the unsprung mobile machine vibration isolation system are provided between a vibration source and the protected object (operator) along the path of vibration propagation. Most machines are now equipped with controlled suspension seats. In the case of active suspensions, the external energy source is required, for instance in the form of compressed air. The compressed air has the advantage that it is generally available in heavy machines as the working fluid and is environmentally friendly. Simulation tests were carried out both in the time domain in Matlab-Simulink and in the frequency domain in the program Mathcad. Simulation tests were performed to investigate effectiveness and stability of the proposed solution and the results were deemed satisfactory. The system was found to be feasible and implementable with respect to every parameter. The first purpose of this study is to develop a simulation model of the active suspension of operator's seat based on an adjustable pneumatic actuator. The other purpose of this study is to examine the effectiveness of different control strategies.

Keywords: *mobile heavy machines, unsprung vehicles, active reduction of vibration, controlled suspension seats, pneumatic actuator*

1. Physical model of the operator's seat in an unsprung mobile machine

In mobile machines with no shock absorbers the vibroacoustic emissions cannot be reduced at the very source, i.e. in wheel suspension elements, but the vibration reduction system can be provided between the cab floor and the object to be vibro-isolated (the operator), along the path the vibration propagate. In most cases, vibration reduction is effected through the use of controlled seat suspensions. Mobile machines during the ride induce intensive low-frequency vibrations, which cannot be effectively reduced by passive methods.

Energy dissipation in wheel tires reduces the vibration intensity in a minor degree only. Active reduction systems, on the other hand, require an external source of energy, for example compressed air. Fig. 1a shows the model of an active seat suspension in which a controlled pneumatic cylinder exerts the force $F(s)$ upon the seat platform accommodating the object to be vibro-isolated, with the weight G ($G = 981$ N).The absolute motion of the object represented by the coordinate – q is the resultant of the platform motion with respect to the vehicle floor – $y(s)$ and of the floor vibration – $z(t)$. That condition holds true as long as the operator does not break away from the seat, i.e. when: $\mathrm{d}^2q/\mathrm{d}t^2 \leq 9.81\mathrm{m/s}^2$. Active seat suspension systems incorporate relief springs to reduce the energy expenditure.

The structure of a kinematic system providing for vertical seat movement with respect to the cab floor is not included in the model shown in Fig 1a. Omission of the kinematic system is regarded as oversimplification when:

1) the cylinder is mounted obliquely to the direction of the seat motion,
2) the effects of friction in sliding pairs are neglected.

The kinematic system widely used in seat suspensions is that based on a shears mechanism including an oblique relief spring. Incorporating a controlled pneumatic cylinder into a parallel system with a spring, as shown in Fig. 1b, we obtain a model of the active system for the seat suspension.

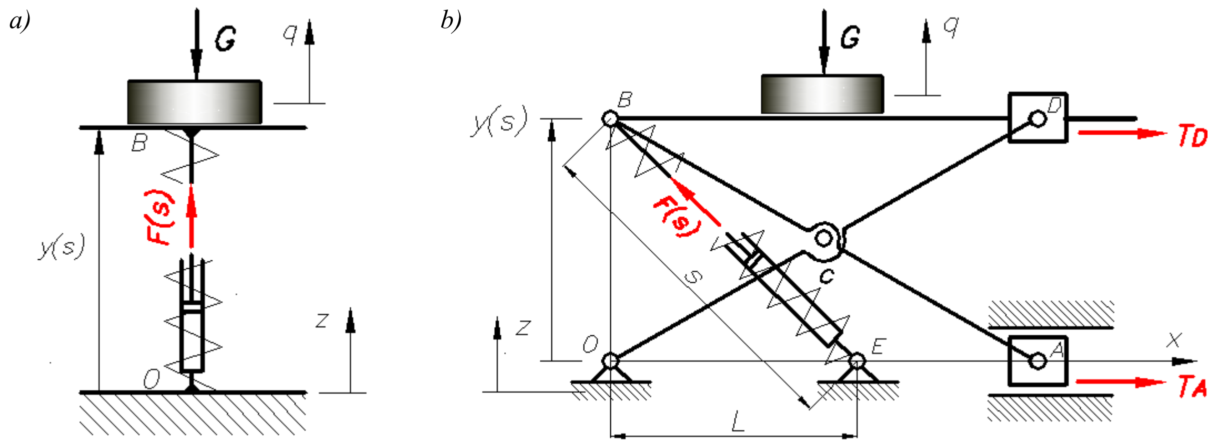

Fig. 1. Model of a dynamic system of the active seat suspension

The absolute movement of the seat with the operator is expressed by the equation:

$$q(t) = y[s(t)] + z(t)\,. \tag{1}$$

The relative seat movement – $y(s)$ is determined by the controllable length of the pneumatic cylinder – s. For the model shown in Fig. 1b, the relative seat movement is expressed as:

$$y(s) = \sqrt{s^2 - L^2}\,. \tag{2}$$

Taking into account friction in sliding pairs, vertical vibration of the operator sitting on the seat mounted as in Fig. 1b are governed by the equation:

$$m\frac{\mathrm{d}^2 q}{\mathrm{d}t^2} = F(s)\frac{\sqrt{s^2 - L^2}}{s}\cdot\frac{sL\mu\cdot\mathrm{sign}\left(\frac{\mathrm{d}s}{\mathrm{d}t}\right) + s^2 - 5L^2}{2\sqrt{s^2 - L^2}\cdot L\mu\cdot\mathrm{sign}\left(\frac{\mathrm{d}s}{\mathrm{d}t}\right) + s^2 - 5L^2} - G\,. \tag{3}$$

In the case of a vertically operated cylinder equation (3) simplifies to the form:

$$m\frac{\mathrm{d}^2 q}{\mathrm{d}t^2} = F(y)\cdot\frac{yL\mu\cdot\mathrm{sign}\left(\frac{\mathrm{d}y}{\mathrm{d}t}\right) + y^2 - 4L^2}{2y\cdot L\mu\cdot\mathrm{sign}\left(\frac{\mathrm{d}y}{\mathrm{d}t}\right) + y^2 - 4L^2} - G\,. \tag{4}$$

2. Model of a regulated single-acting pneumatic actuator

The total force $F(s)$ determining the seat mount mechanism's motion involves the active component due to air pressure, friction in the cylinder sealing, spring response and viscous friction:

$$F(s) = A(p - p_{\mathrm{A}}) + F_T - c(s - s_{\max}) - k\frac{\mathrm{d}s}{\mathrm{d}t}\,. \tag{5}$$

Nomenclature used in the description of the mass-free single-acting pneumatic cylinder:

A – active surface area of the piston (A = 7.854·10^3 m^2),
p – active pressure in the piston chamber in the cylinder,
p_A – atmospheric pressure (p_A = 1.01325·10^5 Pa),
p_0 – pressure supplied from the bottle (p_0 = 6.0·10^5 Pa),
$\dot{M}_d$ – mass flow rate of air supplied to the cylinder,
$\dot{M}_w$ – mass flow rate of air released from the cylinder,
c_p – specific heat under constant pressure,
c_v – specific heat under constant volume,
V – volume of space beneath the piston,
T – air temperature in the in the piston chamber in the cylinder,
T_0 – temperature of supplied air (ahead of the slit T_0 = 293 K),
R – gas constant when R = 287.9 J/(kg·K).

Recalling the laws of:
- mass conservation,
- energy conservation,

the continuity equation and the energy balance equations are derived for the mass flow rates of air supplied and released from the cylinder [1-3].

Recalling the Clapeyrone's law for ideal gas in the differential form, we get a system of differential equations governing the pressure and temperature changes in the piston space:

$$\frac{\mathrm{d}p}{\mathrm{d}t}=\frac{R\,c_p}{V\,c_v}\left[(\dot{M}_d T_0-\dot{M}_w T)+\frac{1}{c_p}\frac{\mathrm{d}Q}{\mathrm{d}t}-\frac{p}{R}\frac{\mathrm{d}V}{\mathrm{d}t}\right], \tag{6}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t}=T\left[\frac{1}{V}\frac{\mathrm{d}V}{\mathrm{d}t}+\frac{1}{p}\frac{\mathrm{d}p}{\mathrm{d}t}-\frac{RT}{pV}(\dot{M}_d-\dot{M}_w)\right]. \tag{7}$$

When the effects of heat exchange between the cylinder walls and the ambience are neglected, the term dQ/dt can be removed from Eq. (6), which implicates the adiabatic process.

Flow velocity is determined by the ratio of pressure upstream and downstream the throttling slit – β. For the cylinder-filling phase, we write:

$$\beta=\frac{p}{p_0}\quad\rightarrow\quad v=\begin{cases}0.53^{1/\kappa}\sqrt{\dfrac{2\kappa RT_0}{\kappa-1}\left(1-0.53^{\frac{\kappa-1}{\kappa}}\right)} & \text{if } \beta\le 0.53,\\ \left(\dfrac{p}{p_0}\right)^{1/\kappa}\sqrt{\dfrac{2\kappa RT_0}{\kappa-1}\left[1-\left(\dfrac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}}\right]} & \text{if } \beta>0.53.\end{cases} \tag{8}$$

In the adiabatic process the exponent κ = 1.4. The mass flow rate is proportional to flow velocity – v, air density – ρ and the flow field – $A_f(u)$, being a function of the command (voltage) signal – u.

$$M=\lambda\cdot\varphi\cdot v\cdot\rho\cdot A_f(u), \tag{9}$$

where:

λ – contraction coefficient (λ = 1),
φ – velocity coefficient (φ = 0.975).

The dependence $A_f(u)$ is derived from the characteristics of the applied servo-valve.

Friction in the sealing in the piston and rod is another important issue. Experimental tests reveal that friction forces in sealings in the cylinder have a complex nature and are dependent on

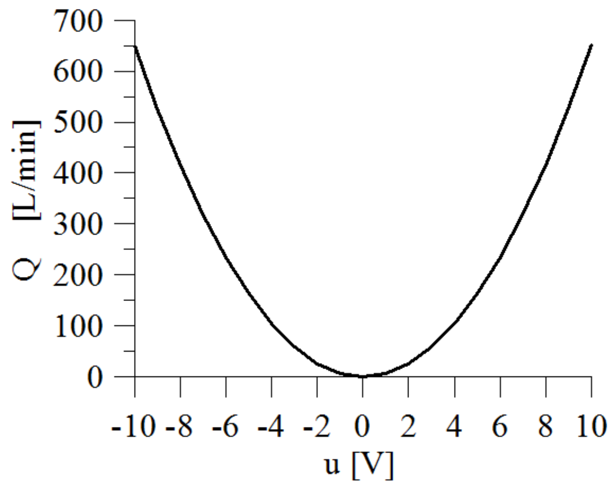

Fig 2. Flow characteristics of the servo-valve

the piston velocity relative to the cylinder. The „stick-slip" phenomenon, i.e. piston creeping with quasi-zero velocity under large loads, causes that only after the stick-slip velocity $-\Delta v_s$ is exceeded will the friction force decrease significantly [7]. Depending on the type of applied sealing and lubrication, this velocity falls in the range $\Delta v_s \in [0.001-0.01]$ m/s and from that instant the friction of motion (kinetic friction) begins [2, 3].

This effect is best captured by the Stribeck's curve and the Karnopp's method provides a concise mathematical formula expressing the static and kinetic friction:

$$F_T = -\begin{cases} \operatorname{sign}[A(p-p_\mathrm{A}) - c_{spr}(s-s_{\max})] \cdot \min\left[F_{FS}, |A(p-p_\mathrm{A}) - c_{spr}(s-s_{\max})|\right] & \text{if } \left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| < \Delta v_s, \\ \operatorname{sign}\left(\dfrac{\mathrm{d}s}{\mathrm{d}t}\right) \cdot \left[F_C + k\left(\left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| - \Delta v_s\right)\right] & \text{if } \left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| \geq \Delta v_s. \end{cases} \tag{10}$$

The forces of static friction F_{FS} and kinetic friction F_{FC} are determined by the difference of pressures acting upon the sealing elements. Recalling the empirical formulas [1, 2], we write:

$$F_{FS} = 2\sqrt{\pi A}\,[f_{FS} + k_{pS}(p-p_\mathrm{A})], \tag{11}$$

$$F_{FC} = 2\sqrt{\pi A}\,[f_C + k_{pC}(p-p_\mathrm{A})]. \tag{12}$$

Basing on [1, 2], it is assumed that: f_{FS} =18.5 N/m, k_{pS} = $7.65 \cdot 10^{-4}$ m, f_C = 16 N/m, k_{pC} = = $5.6 \cdot 10^{-4}$ m.

3. Structure of the regulator

As regards the sensitivity to vibration in the vertical direction, humans most rapidly respond to vibration in the frequency range 4 Hz ÷ 8 Hz. In consideration of the ride comfort, of particular importance is minimizing the acceleration of the driver's torso vibration.

Acceleration measurements can be taken with a disk – shaped acceleration meter placed beneath the seat upholstery. Signals from an acceleration meter integrated with a charge preamplifier can be directly fed to a digital flow controller or, in dispersed structures, to the input in an analogue-digital card. Schematic diagram of the pneumatic cylinder control is shown in Fig. 3.

The reference level for the acceleration signal is zero. Flow control is effected through throttling of the mass flow rate of air supplied and released from the cylinder. No matter what type of controllers, the criterion for control action is minimization of acceleration $\mathrm{d}^2q/\mathrm{d}t^2$.

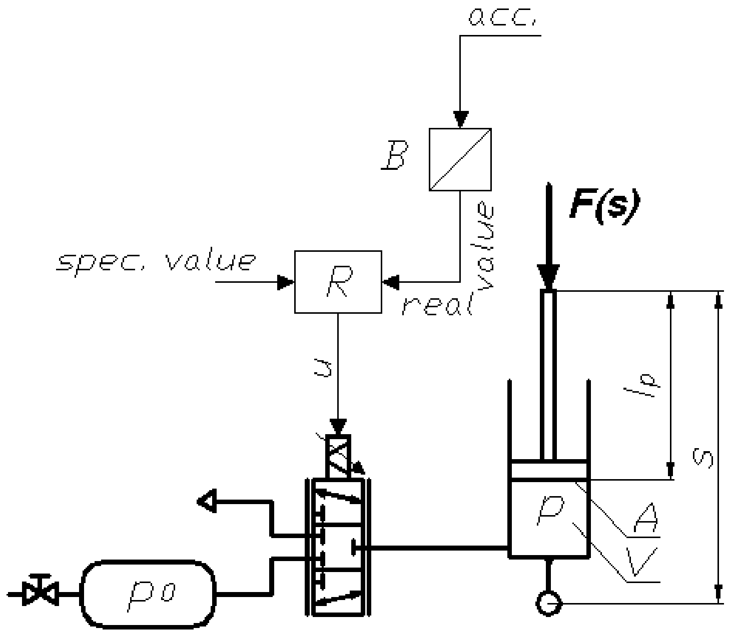

Fig. 3. Schematic diagram of control of the pneumatic cylinder

Development of a regulator's structure is a most complicated and time-consuming task, mostly because of the complexity of the mathematical model of active vibration control and the random nature of acting inputs due to road unevenness.

For low-frequency excitations with limited dispersion, the PDD^2 – type regulators are recommended [4]. The relationship between the command voltage in the regulator and the relative displacement can be written as:

$$u(y) = K_1 \frac{\mathrm{d}^2 y}{\mathrm{d}t^2} + K_2 \frac{\mathrm{d}y}{\mathrm{d}t} + K_3 . \tag{13}$$

The regulator is tuned through selecting the gains K_1, K_2, K_3, following a procedure outlined elsewhere [4], based on the spectral density function of the acting inputs and of the vibration isolation system.

The mathematical model of the vibration isolation system needs to be expressed in a linearised form. Further, for the relative velocity signal, Eq. (13) becomes the equation governing the behaviour of the *PID* regulator.

4. Simulation testing of the seat suspension active model

Due to complexity of the nonlinear mathematical model of the seat suspension active system, the linearization procedure was abandoned. The *PID* regulator was chosen instead, where the input signal was taken as the error signal of pressure value in the actuator's cylinder – p in relation to the atmospheric pressure p_A. Pressure measurements are implemented easily and the system behaviour was stable when compared to other types of input signals. It is so because pressure in the actuator directly controls the active component of the force $F(s)$. The regulator's performance was investigated for the applied sine inputs.

Figure 4 illustrates the effectiveness of the regulator action when it is switched on after a 5 seconds' delay. Accelerations of the operator's vibration are reduced nearly 2.5 – fold in the near-resonance conditions (around 2 Hz).

Control of mass flow rate of air supplying the actuator with voltage signals from servo-valve (see Fig. 2) is shown in Fig. 5. When the regulator is on, temperature and pressure fluctuations in the actuator's cylinder are minimised, which is shown in Fig. 6.

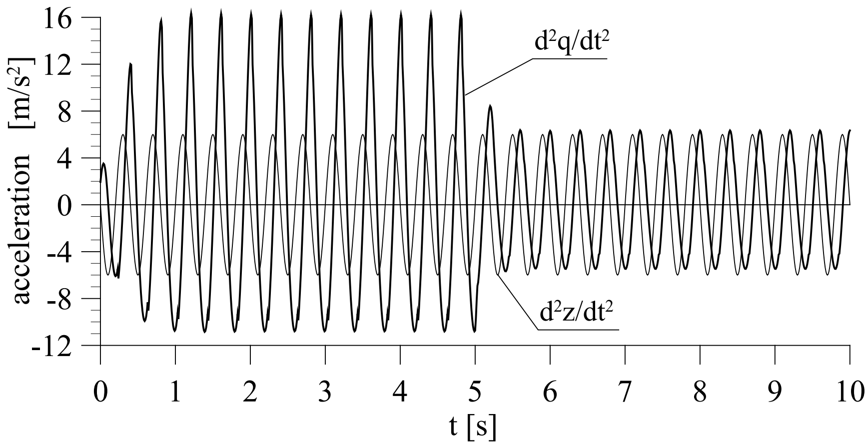

Fig. 4. Acceleration of operator's vibrations and excited vibrations

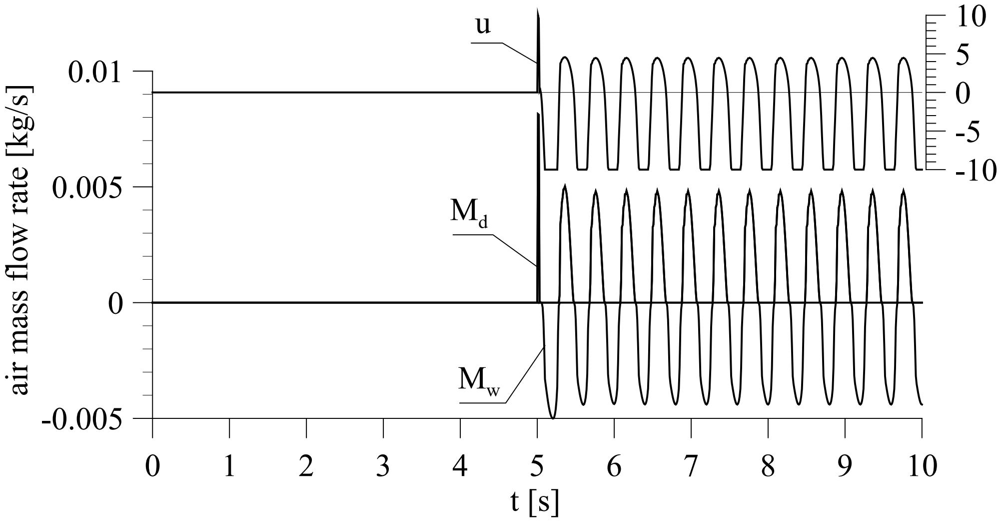

Fig. 5. Controlling the mass flow rate of air supplying the actuator using the voltage signal from a servo-valve

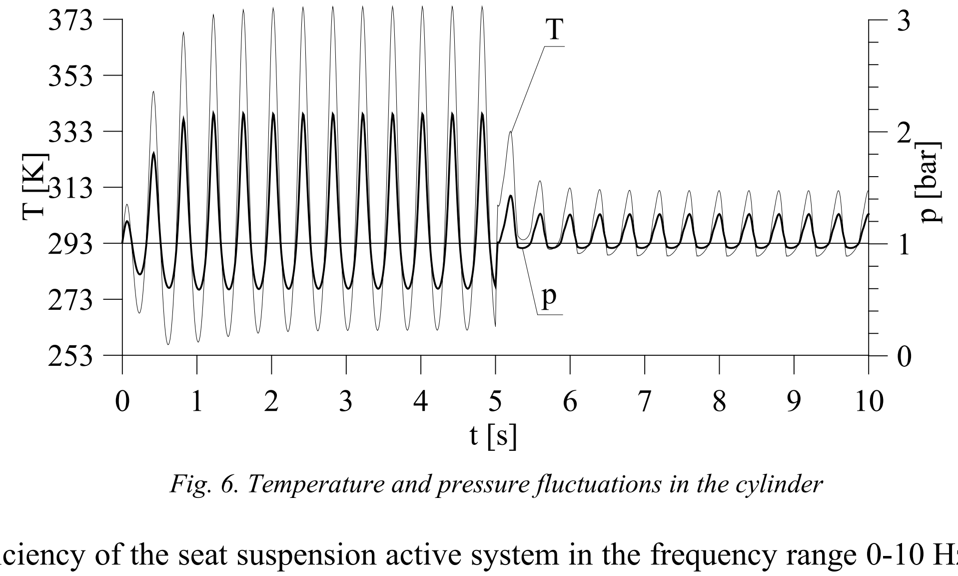

Fig. 6. Temperature and pressure fluctuations in the cylinder

The efficiency of the seat suspension active system in the frequency range 0-10 Hz is evaluated basing on the transmissibility function for vibration acceleration – $\alpha(f)$, defined as the ratio of acceleration amplitudes of the operator's vibration and of the acting excitation.

Transmissibility function for the active system (*PID* regulator in the on state) is designated by – α_{PID} and graphed with a thick line. For comparison, the transmissibility function for the passive system (*PID* regulator in the off state) is designated by – α and graphed with a thin line.

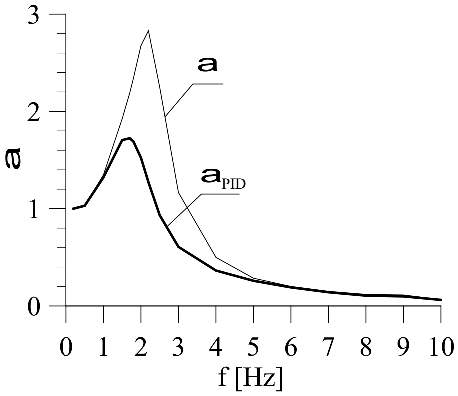

Fig. 7. Transmissibility function

The active system is most effective in the neighbourhood of resonance frequency. Its effectiveness is nearly twice as high as that of the passive system. In the super-resonance range (in excess of 6 Hz) vibration isolation performance of the active and passive systems is nearly identical. The active system behaves like a real vibration isolator ($\alpha_{PID} < 1$) for frequencies $f > 2.5$ Hz, the passive system – for $f > 3.5$ Hz.

The model of the seat suspension active system used in the calculation procedure was developed in Mathcad and Matlab-Simulink. The remaining parameters of the system are: length of links in the shears mechanisms – $L_{AB} = L_{OD} = 2L = 0.6$ m, coefficient of friction in sliding pairs in the shears mechanism – $\mu = 0.1$, stiffness of the relief spring – $c = 1.0284 \cdot 10^4$ N/m

4. Conclusion

The model of the seat suspension active system takes into account thermodynamic processes in the pneumatic actuator as well as friction in the piston and sliding pairs sealings. The structure of the regulator system was chosen and the regulator was tuned accordingly. Simulation tests were performed to investigate effectiveness and stability of the proposed solution and the results were deemed satisfactory. The system was found to be feasible and implementable with respect to every parameter. Because of small pressure pulsation and low flow rates, the power demand should be insignificant, which encourages further tests and, finally, implementation of the solution. Future research efforts will concentrate on development of more complex control strategies, such as self-tuning regulators.

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