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THE STUDY ON MUFFLER PARAMETERS INFLUENCE ON THE PNEUMATIC PRESSURE REDUCING VALVE PERFORMANCE

Kirill Afanasev*, Evgeny Shakhmatov, Alexander Igolkin, Dmitry Stadnik, Victor Sverbilov

Samara National Research University, Moskovskoye Shosse 34, 443086, Samara, Russian Federation
email: Highlander-92@yandex.ru

The main noise source in the air or gas transmission systems is the air or gas pressure reduction in system shut-off-and-regulating elements, such as pressure reducing valve. The most common means of noise control in such systems is the installation of mufflers, that represent one or several orifices. However, the design of such devices requires a very careful approach, as mufflers can have a very negative impact on the parameters and performance of the whole system. This paper continues the studies on the pneumatic pressure reducing valve with the muffler installed in the outlet line. The modified mathematical model of this system, which was implemented in the Simulink software and described in the previous studies, was used to analyze the muffler impact on pressure relief valve static and dynamic characteristics. The program in Matlab software was also developed for determination of the noise, generated by the system. The influence of several muffler parameters (the flow area, the distance between the muffler and the regulator) on the quality of transient processes was analyzed during the simulation. The muffler flow area values and the muffler position in the system, which provide the lowest level of the noise, generated by the system, were determined. The experimental research, using the pneumatic test bench, located in a semi-anechoic chamber, was carried out for mathematical model verification.

Keywords: pressure reducing valve, muffler, feedback pipeline, stability, acoustic power

1. Introduction

Pressure regulators (pressure reducing valves) are one of the most popular types of valves used in air preparation units. They find application in many spheres: aerospace technologies, gas distribution stations, steam and heat supply systems, medicine (for example, anesthesiology), etc. The main difficulty in designing such units is the achievement of high control accuracy along with the stable system operation and low noise and vibration emission [1]. Self-oscillations of the regulator valve, which may occur due to cycling, chatter or flatter [2], are also the source of noise and vibration in the connected system. Special mufflers or dampeners are installed in the system to reduce noise and vibration. These mufflers represent a throttle plate pack installed downstream of the regulator's poppet valve. This type of muffler allows to implement step-by-step throttling: the pressure difference on the poppet valve decreases, and hence the noise level is reduced [3]. A lot of papers are devoted to the regulator-muffler systems research, at [4] an experimental study and CFD modeling of the system including regulator and perforated throttle dampeners (mufflers) was carried out. Next paper [5] is devoted to a valve operation theoretical study based on the created mathematical model and visualization of the muffler acoustic efficiency depending on the number of holes in the muffler and its total area. Mathematical model further development, as well as regulator dynamic characteristics study, taking into account the muffler installed in the system, was carried out in [6]. Issues of regulators stability, including regulators with pilot control, were considered in several works [7-9]. However, despite a large number of papers, none of them has a comprehensive study of the issue devoted to the stability of the system including pressure regulator and muffler, which also takes into account the optimal acoustic characteristics of the regulator and muffler. Work in this field was started in [10]. This paper continues the cycle of work described above and is aimed at a comprehensive study of the characteristics of a pneumatic pressure regulator with an installed muffler which represents an orifice and verification of the mathematical model.

2. Methods

The experiments were carried out using a test rig and a mathematical model. The mathematical model was described in [5, 6,10].

In this paper, the feedback pipeline mathematical model was modified on the basis of experimental studies. The losses due to the turbulent flow regime were taken into account in the pipeline mathematical model:

$$Z_p = f \cdot \frac{RT_1}{p_1} \cdot \frac{l_p}{d_p} \cdot \frac{G_p^2}{2 \cdot S_p^2}, \tag{1}$$

where Z_p – hydraulic losses in the pipeline; R – gas constant; T_1 – inlet air temperature; p_1 – inlet air pressure; l_p – pipeline length; d_p – pipeline cross-section diameter; G_p – air mass flow rate in the pipeline; S_p – pipeline cross-section area; f – the friction factor which approximates by Haalan function:

$$f = \left(-1.8 \log_{10} \left(\frac{6.9}{\mathrm{Re}} + \left(\frac{e}{3.7 d_p} \right)^{1.11} \right) \right)^{-2}. \tag{2}$$

where e – pipeline inner surface roughness.

Experimental research was carried out using a modernized test bench, refer to Fig. 1 and Fig. 2, and acoustic camera Norsonic Nor 848, operates on the beamforming principle. It has 256 microphones, video camera and allows to monitor and visualize the noise emanating from the object of the study.

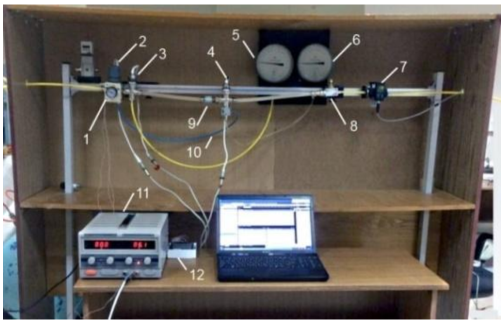

Figure 1: Modified test rig main view.

1 – pressure regulator with magnetoresistive sensor for popet travel measurement; 2 - dynamic pressure sensor (pressure in the feedback pipeline); 3 - dynamic pressure sensor (pressure after pressure regulator); 4 - dynamic pressure sensor (pressure after orifice plate); 5 - pressure gauge (pressure after pressure regulator); 6 - pressure gauge (pressure after orifice plate); 7 – flow meter; 8 – adjustable orifice; 9 – orifice plate; 10 – feedback pipeline; 11 – sensor power supply; 12 – analog input unit NI9239 with NI USB9162.

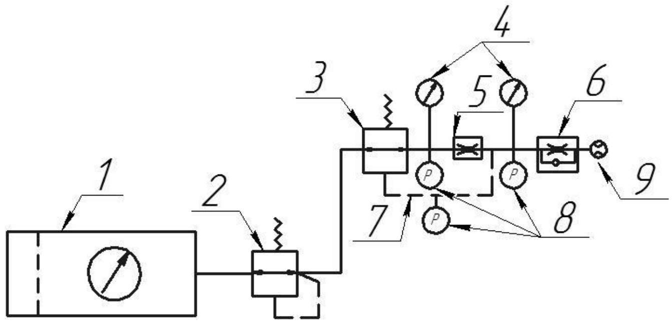

Figure 2: The test rig pneumatic scheme.
1 – pressure source; 2 – inlet pressure regulator; 3 – test pressure regulator; 4 – pressure gauges;
5 – orifice plate; 6 – outlet orifice; 7 – feedback pipeline; 8 - dynamic pressure sensors; 9 – flow meter.

The upgraded test rig is almost the same as described in [10] and it consists of two pressure regulator – one for a constant level of the inlet pressure, and the second was a subject of investigation. The feedback of the second pressure regulator is connected to the pipeline after the adjustable orifice which simulates the muffler flow area. This was made to keep the air pressure just behind the entire regulator-muffler system. The standard feedback hole in the regulator is muted. At the output of the system another load orifice and a flow meter are installed. The system also includes three dynamic pressure sensors VT-206 with a measuring range from 0 to 1.4 MPa and two standard pressure gauges with a measuring range from 0 to 0.6 MPa for monitoring the pressure level after the pressure regulator, the pressure level in the feedback line and the pressure level after the muffler. The experiments were carried out with an inlet pressure of 0.7 MPa and the pressure drop on the test pressure regulator of 0.3-0.5 MPa.

A flow meter Festo SFAB-200U-HQ10 with a measuring range from 0,05 to 320 Nl/min was installed to monitor the air flow at the outlet.

The position of the pressure regulator poppet is measured using a magnetoresistive sensor Honeywell SS495A1, which is fixed to the regulator using a special non-magnetic holder, refer to Fig. 3. The sensor was not attached directly to the regulator bottom cover in view of the fact that when the popet is moving, the bottom cover also has movement relative to the body and the popet, which introduces an error in the sensor data.

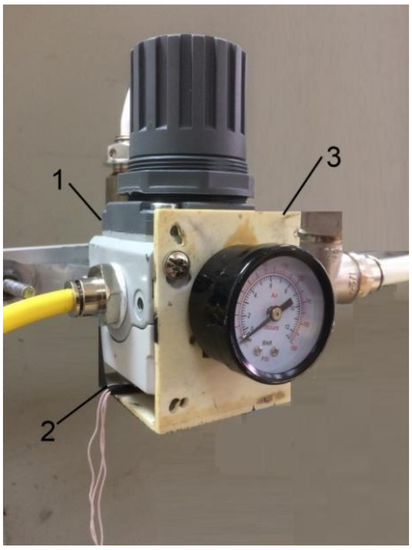

Figure 3: Sensor for popet travel measurement installation
1 – regulator; 2 – sensor for popet travel measurement; 3 – sensor holder.

A magnet in the form of a ring was mounted to the regulator popet. Preliminary experiments showed that the considered spring-mass system is an aperiodic link of the second order, therefore the installation of a small mass ring (2 g) does not affect the transient characteristics of the system due to the high spring stiffness. The sensor calibration for determining the correspondence between the signal and the popet travel was carried using a special calibration unit, refer to Fig. 4.

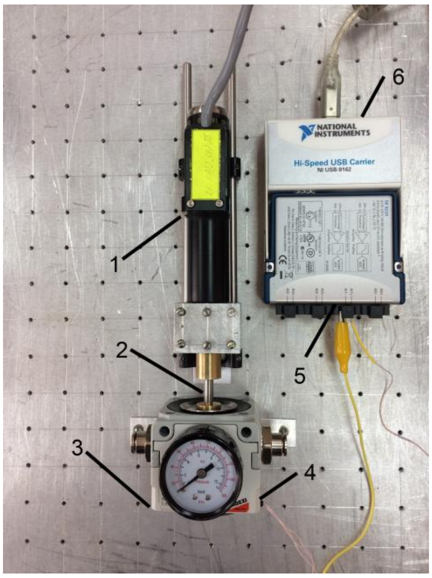

Figure 4: Popet travel sensor calibration unit
1 – linear actuator; 2 – regulator rod;
3 – pressure regulator; 4 – magnetoresistive sensor for popet travel measurement;
5 – analog input unit National Instruments NI9239; 6 – unit NI USB9162.

The pressure regulator rod was connected to a linear actuator which set a single displacement. The sensor was connected to a testing PC via National Instruments NI9239 and NI USB9162 units.

3. Results

3.1 Research on dynamic characteristics

The feedback channel of the gas pressure regulator in the factory configuration is made as a hole connecting the output cavity with the sub-membrane (control) cavity, which does not lead to significant delays in the feedback signal transmission.

The muffler installation at the regulator output inevitably leads to the appearance of a feedback pipeline. The appearance of a delay in signal transmission from the output to the sub-membrane (control) cavity can be accompanied by the appearance of self-oscillations, refer to Fig. 5.

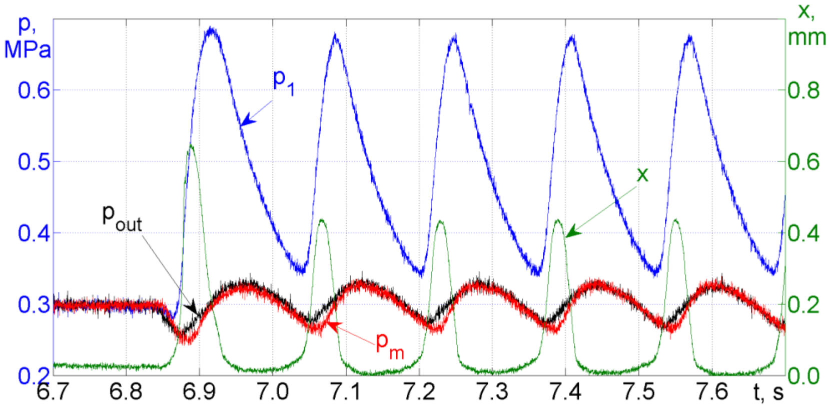

Figure 5: Regulator unstable work for feedback pipeline length of 2950 mm and orifice diameter of 2 mm ($f = 6.25\,Hz$).

The above graph is obtained by stepwise opening of the output adjustable orifice by an amount corresponding to the average flow rate 200 Nl/min. The analysis of the graph shows the presence of a delay between the output pressure p_{out} and pressure in the sub-membrane (control) cavity p_m. Reducing the feedback pipeline length leads to a decrease in the delay, however, the self-oscillation parameters change insignificantly: an increase in the oscillations frequency is observed, while their amplitude decreases, refer to Fig. 6.

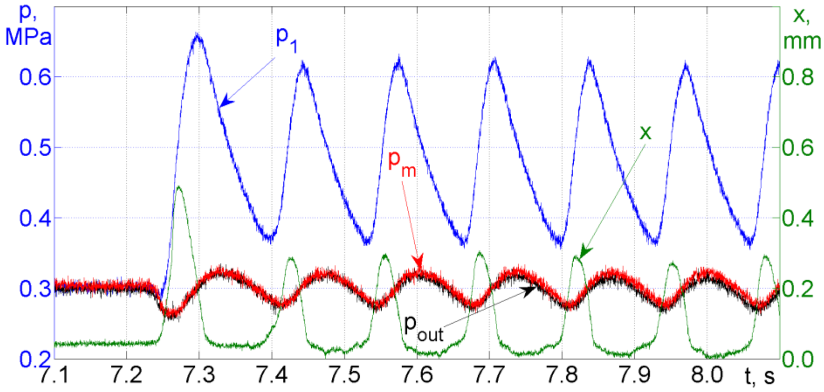

Figure 6: Regulator unstable work for feedback pipeline length of 800 mm and orifice diameter of 2 mm ($f = 7.63 Hz$).

The orifice plate removing while keeping the feedback pipeline length the same leads to a significant increase in the oscillations frequency and a decrease in their amplitude, refer to Fig. 7. The results presented at the Fig. 7, are obtained by smooth opening of the output orifice. It can be seen from the graph that the amplitude of the pressure oscillations in the sub-membrane (control) regulator cavity exceeds the amplitude of pressure oscillations in the output cavity. Thus, there is an oscillation increase in the regulator control cavity in the presence of a feedback pipeline and the absence of orifice plate.

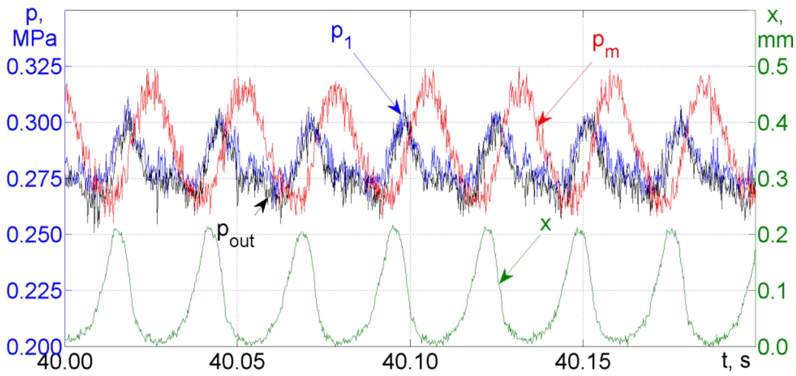

Figure 7: Regulator unstable work for feedback pipeline length of 1000 mm and without orifice ($f = 37.5 Hz$).

It can be seen from all obtained graphical dependencies that the time at which the rate of increase of pressure before the orifice plate dp_1 / dt has the maximum value, coincides with the time when the pressure in the sub-membrane (control) cavity is minimal. Thus, the phase shift of the above signals by 90 degrees indicates the presence of an auto oscillatory system, the dissipation in which is compensated by the energy input from the source of constant pressure at the inlet (volume).

The additional experiments with the change of length of the pipeline between the regulator and the muffler were carried out, they confirmed the assumption that this parameter does not affect the auto-oscillations nature in the system.

3.2 Research on acoustic characteristics

In addition to the dynamic characteristics the system optimal acoustic performance (the muffler efficiency) was examined for new measurements conditions in the semi-anechoic chamber.

The Fig. 8 shows the acoustic visualization of working system, where we can see two sources – the muffler and the regulator, since the muffler installation in the system invariably leads to the formation of a second source, but an effective muffler, meanwhile, reduces the overall noise level emitted by the system.

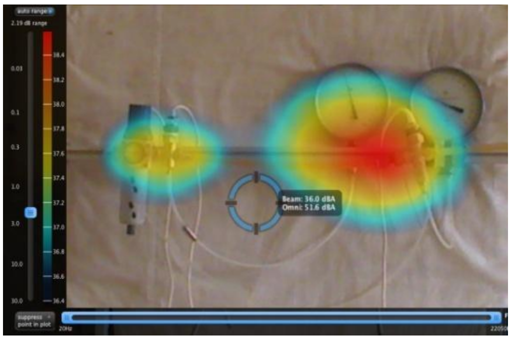

Figure 8: Acoustic visualization of muffler and regulator noise.

The sound power levels for the muffler and regulator were analyzed separately with the help of the acoustic camera. The total value was calculated using Eq. (3) and it was used as a factor in assessing the muffler effectiveness:

$$L_{RS} = 10 \cdot \lg(10^{0.1L_R} + 10^{0.1Ls}), \qquad (3)$$

where L_R and L_S - sound power levels in dB for regulator and muffler respectively.

The theoretical relationship between the sound pressure level and the muffler cross-sectional area for various flow rates was considered earlier in [10], where the system stability areas were also considered in dependence on the muffler cross-section diameter. According to the obtained data, the orifices with a cross-sectional diameter of 1.5 and 2 mm were examined. The effect of the length of the pipeline between the regulator and the muffler on the sound power levels of the system was also examined. The results are shown in Table 1.

Table 1: Measurement results

Pipe length, mm	Orifice diameter, mm	$L_{w\ regulator}$, dB	$L_{w\ muffler}$, dB	$L_{w\ sum}$, dB	Flowrate, NL/min
90	1.5	28.7	33.9	35.05	143.0
225	1.5	27.7	34.4	35.24	144.4
270	1.5	28.0	35.4	36.13	147.5
90	2.0	33.5	37.3	38.81	298.5
225	2.0	32.5	37.7	38.85	304.9
270	2.0	32.1	38.0	38.99	309.7
90	without muffler	33.5	35.0	37.32	>320
225		33.4	35.4	37.52	>320
270		34.8	36.2	38.57	->320

As can be seen from the Table 1, the lowest system sound power level is achieved for a pipe length of 90 mm and an orifice with a diameter of 1.5 mm. The length of the pipe between the regulator and the muffler does not have a significant effect on the overall noise level, there is a tendency for an insignificant increase in the noise level with an increase in the length of the pipe. The values of the theoretical and experimental optimum values of the muffler cross section are still different. In paper [10] it was calculated that for a flow rate of about 140 Nl / min, the optimum cross-sectional area is 3.19 mm^2, the value obtained here - 1.76 mm^2, such discrepancy can be explained by the impossibility of taking into account all factors of sound propagation in a mathematical model.

4. Conclusion and future work

Thus, dynamic characteristics of pneumatic pressure regulator with the installation of the muffler were examined. The feedback pipeline length influence on the system dynamic characteristics is analyzed, and the length influence of the pipeline between the regulator and the muffler on the dynamic characteristics and sound power levels of the system is considered.

The proposed modification of the regulator, in which the pressure feedback signal comes from downstream of the muffler, proves its efficiency. However, the feedback pipeline length has critical impact on the regulator dynamics. The muffler effect on transient process quality is analyzed. It is shown that the increase in the muffler resistance tends to the settling time increase and further to the unstable behaviour.

The analysis carried out in the paper would be helpful to a proper muffler design for the given pressure reducing valve.

Future research will be focused on the implementation of the results, obtained for an orifice as a muffler simulator, for studying the real construction - a package of orifice plates, and also on the mathematical model subsequent verification by taking into account the obtained experimental data.

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