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Experimental identification of a pneumatic valve-cylinder system for attitude control

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Abstract. Vertical vibrations on agricultural tractors, due to soil irregularities, represent a major cause of diseases of agricultural operators. The control of noise and vibration on the operator is of interest to the (Italian) National Institute for Insurance against Accidents at Work (INAIL). A prototype of an active suspension system of the operator seat has been developed in the laboratories of INAIL. The prototype can be configured with hydraulic or pneumatic actuation. This paper focuses on the pneumatic solution powered by a proportional pressure control electro-valve. Since several proportional pressure control valves are commercially available, three of them have been experimentally tested to determine the most suitable one for control purposes. Numerical modeling of the three valves has been carried out and described. Experimental tests on the selected valves allow for identifying the main dynamic parameters of the numerical models, providing a reliable simulator to be adopted for the development and optimization of the control system. The performances of the valves are compared and discussed.

Introduction

A major cause of diseases of agricultural operators is represented by the vertical vibration exposure on agricultural tractors due to soil irregularities. The effects of the whole-body vibrations on operators of industrial and agricultural machines are extensively investigated in [1]. To reduce the vibrations that are transmitted to the operator, the cabin and the seat are generally provided with a suspension system that can exploit either passive, semi-active, or active solutions.

The first ones, typically mechanical or pneumatic [2], allow for an attenuation of the vibrations beyond a given cut-off frequency. Better performances are guaranteed by the semi-active systems that adjust the damping coefficient according to the soil irregularities [3]. Active solutions provide the best performances by the adoption of an actuation system generally based on either hydraulic, electromagnetic, or electromechanical technology [4].

The (Italian) National Institute for Insurance against Accidents at Work (INAIL) is interested in active vibration control (AVC) and active noise control (ANC) technology to reduce the vibration and noise exposure of operators and founded several research programs in the last years. A laboratory prototype of an active suspension system for the operator seat is developed. It can employ either hydraulic or pneumatic actuation.

The present study is focused on the pneumatic actuation made of a double-effect pneumatic cylinder powered by a proportional pressure control electro-valve. The attitude control system generates the command signal for the electro-valve according to the soil irregularities. This kind of valve converts an electrical input signal into an output pressure.

In this work, the pneumatic actuation is evaluated for adoption in an active control system. This technology is, in fact, very versatile and is widely used for its cheapness, cleanness, and high power-to-weight ratio. In fact, it allows for force control, variable stiffness actuation (VSA) [5], position control in several industrial sectors [6], and also for non-traditional applications such as deformable pneumatic soft actuators [7, 8].

To properly develop an active control system, the dynamic properties of the main components of the system must be known, including that of the electro-valve. Among all the commercially available valves, three have been experimentally tested. The dynamic parameters of each valve are experimentally identified, and performance comparisons of the valves are reported and discussed to find the most suitable one for control purposes. Starting from the identified dynamic parameters, numerical modeling of each valve is carried out. The models provide a reliable simulator of the dynamics of the valves to be included in a more complex model of the whole active suspension system for the development and optimization of its control system.

The prototype of the seat suspension

The prototype of the seat suspension is made of the actuation system placed on a vibrating plate. As shown in Fig. 1, the actuation system is mounted over the vibrating plate (1), and it is composed of the pneumatic actuator (2), the sliders (3), the suspended body (4), a set of springs (5), a couple of electro-valves (6), a linear position transducer and a couple of accelerometers.

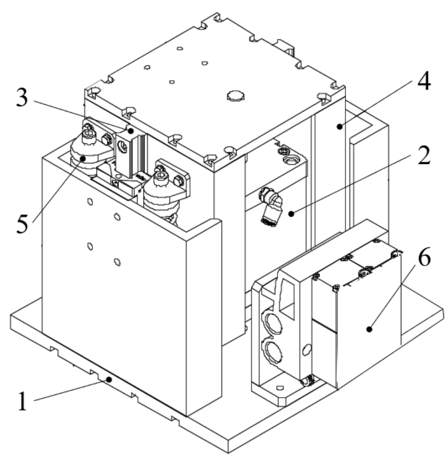

Fig. 1: Prototype of the seat suspension

The actuator (ISO 21287 double-acting, bore 80 mm, stroke 70 mm) is mounted along the vertical axis. The upper end of the rod is bolted to the suspended body and ensures its motion. The proportional pressure control valves are directly connected to the chambers of the cylinder. The outlet pressure of the valves must ensure the motion of the piston (rod) according to the soil irregularities. The dynamics of the suspended body is affected by the external disturbances in terms of platform vibrations and by the dynamics of the electro-valves.

Experimental activity

Three commercial pneumatic proportional pressure control valves are adopted and tested:
- Festo VPPN-6F-L-1-F-0L6H-A4N-S1 (hereinafter called Valve 1);
- MetalWork Regtronic 1/4” (hereinafter called Valve 2);
- SMC ITV2050-31F2BL3 (hereinafter called Valve 3)

The working principle of the three valves is the same: a pilot stage, made of two modulating pilot solenoid valves, ensures the pressure adjustment in a pilot chamber. Inside it, the pressure acts on

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the upper surface of a diaphragm connected to the air supply valve: changes in pressure cause the opening/closing adjustment of this valve, with a consequent adjustment of the outlet pressure.

Valve 1 provides multiple control loops instead of the conventional direct-acting controls. Although the outlet pressure is proportional to an input command current, which ranges from 4 to 20 mA, the valve is commanded by an equivalent tension command, which ranges from 1 to 5 Vdc. This valve can work according to three functional response modes (fast, universal and precise control behavior): in the present activity, the valve is set to the fast control behavior mode. No pressure full-scale adjustment is possible.

Valve 2 can work according to three functional modes: in the present activity, the fast regulation mode is set. In this case, the pressure full-scale can be adjusted.

Valve 3 can work in one only functional mode. No pressure full-scale adjustment is possible.

Regardless of the type, the dynamic behavior of these kinds of valves can be described, with a good approximation, by the following second-order equation [9]:

$$\ddot{p}_{out} + 2 \cdot \zeta \cdot \omega_n \cdot \dot{p}_{out} + \omega_n^2 \cdot p_{out} = \omega_n^2 \cdot V_{in} \cdot k_v \tag{1}$$

where p_{out} is the outlet pressure from the valve (measured in bar), V_{in} is the input command signal (measured in V), k_v is the static gain (measured in bar/V), and ζ and ω_n are the damping ratio and the natural frequency, respectively.

All the valves are tested in the pressure range 0.0 - 6.0 bar. The k_v value of each valve is identified in specific experimental tests where V_{in} is increased in quasi-static conditions from 0.0 to 6.0 bar. The dynamic parameters ζ and ω_n of each valve are identified by imposing an offset sine wave V_{in} with three different amplitudes and the same offset at different frequencies. The corresponding pressure amplitudes are 1.80 bar, 1.20 bar, and 0.60 bar, representing 20-80%, 30-70%, and 40-60% of the outlet pressure range, respectively. The corresponding offset pressure is 3.0 bar. For a given amplitude, tests are carried out at different frequencies in the range starting from 0.1 Hz up to a frequency that does not provide any variation of the outlet pressure.

A signal generator (GW Instek AFG-2125) is used for V_{in}. A data acquisition system (made of a USB NI6001 DAQ-board and software developed in the NI LabView environment) is adopted to acquire V_{in}, converted into the corresponding pressure set-point, and the outlet pressure p_{out}. The sampling frequency is set to 1 kHz.

Fig. 2 shows the input pressure set-point and the associated output at the frequency 0.5 Hz. At the considered frequency, each valve exhibits a proper distortion pattern of the outlet pressure. In particular, Valve 1 exhibits a more visible distortion in the rising part of the curve. Such a behavior could be due to the multiple control loops performed by the valve controller. Valve 2 exhibits a high difference between the rising and falling parts of the curve. This might be due to the pneumatic resistances in the two airflow directions being very different. Valve 3 exhibits some distortions after the stationary points of the curve.

The magnitude and the phase shift of the outlet pressure with respect to the set-point, are achieved using the following criterion: at a given frequency, according to the nonlinear least-squares method, each of the two curves is described by an offset sine expressed by the equation:

$$y(t) = A + B \cdot \sin(2\pi f \cdot t + \varphi) \tag{2}$$

where A is the identified offset value of the input/output (i/o) signal (in bar), B is the identified amplitude of the signal (in bar), f is the frequency of the input signal (in Hz), t is the time (in s), and φ is the identified phase of the signal. Hence, for each frequency value, the magnitude $|Bo/Bi|$ and the phase shift $\varphi_f = \varphi_o - \varphi_i$ are computed. The whole procedure and the associated intermediate results are detailed in [10].

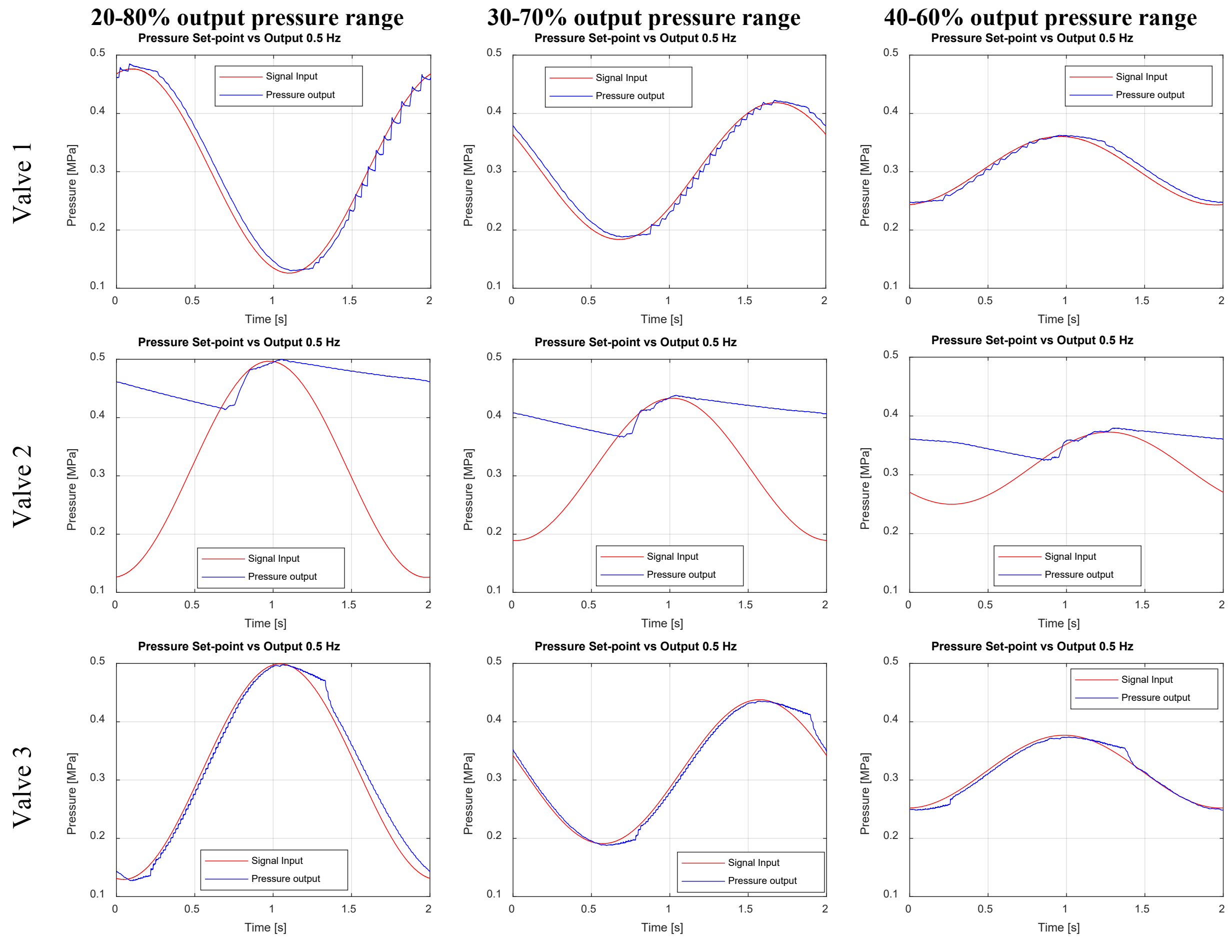

Fig. 2: Example of experimental input and output signals of the electro-valves

The identified magnitude and phase shift are gathered in Fig. 3, for the three valves. The identified k_v, ζ, and ω_n for each valve are reported in Table 1 together with the main characteristics. The identification aims to approximate the magnitude and the phase shift for the three amplitude values, representing a linear approximation of the non-linear behavior of the valves. Up to 5 Hz, Valve 1 behaves linearly and experimental data for different amplitudes provide the same values of magnitude and phase shift. On the contrary, for higher frequencies, different values are obtained. The high attenuation that is observed on the response of Valve 2 at 0.5 Hz (see Fig. 2) is confirmed by its magnitude plot. Valve 3 shows a non-linear behavior throughout the considered frequency range. In particular, also at low frequency, the experimental frequency response is amplitude-dependent.

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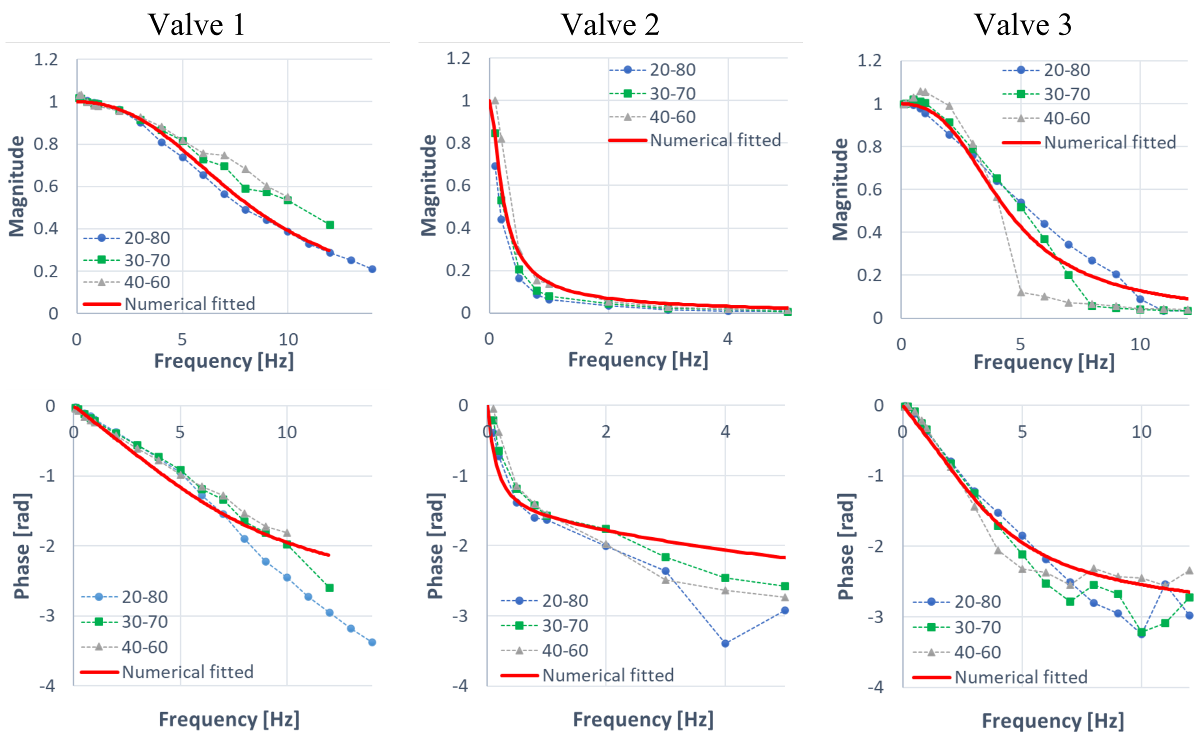

Fig. 3: Magnitude and phase of the three valves responses

Table 1 – Main characteristics of the tested proportional electro-valves

Valve	P_{out} *range [bar]*	V_{in} *range [Vdc]*	k_v *[bar/V]*	ζ	ω_n *[rad/s]*
1	0.00 – 6.0	1 – 5	1.43	0.85	45
2	0.05 – 6.0	0 – 10	0.60	3.5	6.28
3	0.05 – 9.0	0 – 10	0.90	0.8	23

Numerical modeling and experimental validation

To validate the model of the valves, experimental data are compared in Fig. 4 to numerical results obtained by imposing on the model the same experimental offset sine V_{in}. The model of Valve 1 fits the experimental data with a good approximation at 7 Hz and at the highest amplitude condition. As previously noted, the model of Valve 2 is not able to retrace the experimental data because the adopted model does not consider the nonlinear behavior of the valve, which produces a different offset of the response with respect to the expected one, in addition to different behavior with rising and falling pressures. Finally, also the model of Valve 3 does not properly retrace the experimental: some slight nonlinearities occur.

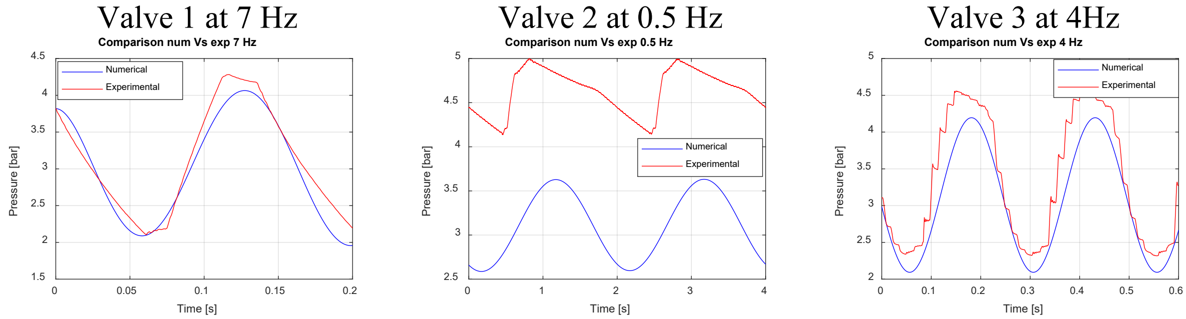

Fig. 4: Comparison between experimental and numerical output pressures of the three valves

Conclusions

In this paper, three commercial proportional pressure control pneumatic electrovalves are experimentally tested to identify their dynamic properties and to evaluate their use in a pneumatic vibration control system. The tests performed on the selected electrovalves allowed to identify the resonance frequencies and the damping factors of the second-order system adopted to model the valve behavior. However, for two of the three electrovalves, significant nonlinearities are present that introduce a relevant distortion in the pressure response of the valve, limiting their use for a control application. Although the model of Valve 1 fits the experimental data with a good approximation up to 7 Hz, the attenuation of the response can be considered suitable for control purposes only below 2 Hz.

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