DYNAMICS OF THE ELECTROHYDRAULIC SERVOPUMPS FOR INDUSTRIAL OPEN CIRCUITS

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The paper presents the modelling, simulation, and experimental validation of the mathematical model created for a high performance electrohydraulic servopump. The validation test was performed by the authors in the Electrohydraulic Amplifier Performance Certification Laboratory associated to the Romanian Accreditation Association - RENAR. The experimental results were found in good agreement with the theoretical ones. The simulation model was included in the AMESIM super components library developed by the authors in the frame of some PhD thesis.

Keywords: modeling, simulation, experimental identification, servopumps, open fluid power systems.

1. Introduction

The modern servopumps are controlled by hydro mechanic or electrohydraulic servo systems, according to the application performance demands. Special applications like military ones require hybrid systems, including electric control devices and mechanical feedback, rigid or elastic ones. Another structural matter regards the pressure supply of the control system. The classic servo systems are powered through auxiliary pumps with gears, put into motion by the controlled pumps, and are protected against overpressure by piloted valves. In the case of an open circuit, powering the servo system through the controlled pump is possible, if the minimum back-pressure of the latter is greater than the minimum pressure necessary for control. A typical example for this principle is the PV Plus servopump family (Figs. 1 and 2). The tilting angle of the swash plate is controlled by a three-way flow valve (Fig. 3), with a small overlap, actuated by a proportional force solenoid which has a nominal force of about 90 N. The spool control edges have different geometry: one edge is very sharp and is controlling the displacement increase. The other control edge has a small chamber, increasing the displacement recovery time. The position control loop feedback from the hydraulic cylinder rod

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is given by an inductive transducer, whose sensing shaft is permanently in contact with a conical surface of the piston due to a spring. The transducer is waterproofed (Fig. 4). The pump flow is set up by a PID compensator which is followed by the PWM voltage interface of the proportional force solenoid.

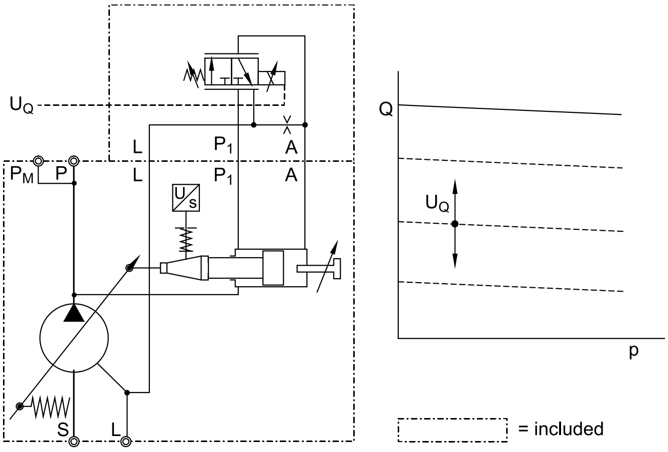

Fig. 1. Hydraulic diagram of a PARKER PV Plus electrohydraulic servopump.
Source: PARKER HANNIFIN CORPORATION [12]

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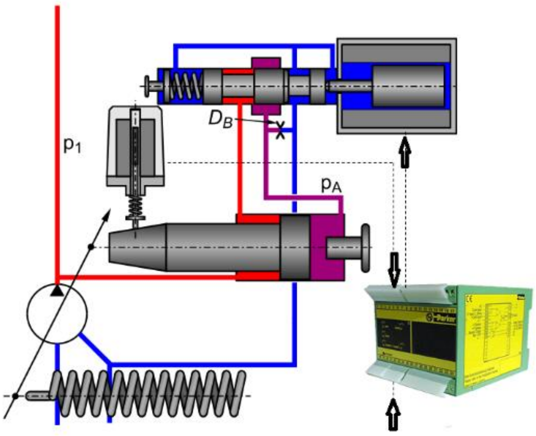

Fig. 2 Schematic diagram of a PARKER PV Plus electrohydraulic servopump.
Source: PARKER HANNIFIN CORPORATION [12]

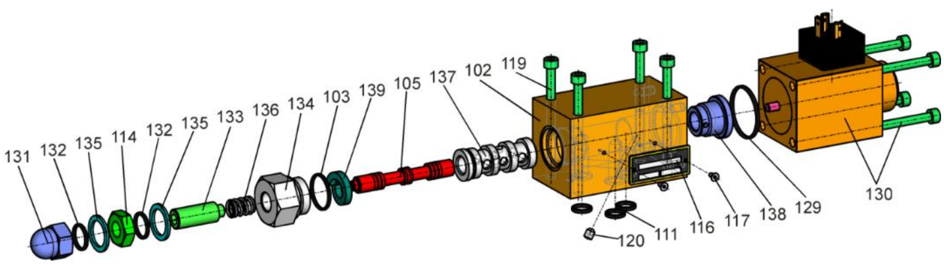

Fig. Expanded view of a proportional three-way flow valve: 102-valve case. 105-spool; 130-proportional force solenoid; 136-reference spring; 137-sleeve.
Source: PARKER HANNIFIN CORPORATION

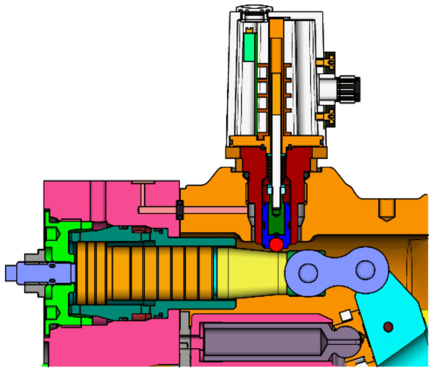

Fig. 4 Measurement system for the tilting angle of the swash plate.
Source: PARKER HANNIFIN CORPORATION

The analogue or digital servo controller allows the voltage control (0…10V), the current control (0…20 mA), or by a PROFIBUS CAN. The reference of the transducer core can be adjusted with a potentiometer. The PID error amplifier generates a control signal for a PWM generator. which has a Dither signal with a frequency of 100 Hz, and an amplitude of 0.2 V. The servo controller is DC powered (22…36 V) by the internal power supply (±15 V); it is galvanically isolated to prevent accidental overvoltage.
In cyclic applications, the servomechanism is combined with a direct-action mechanical pressure compensator, or with an electrohydraulic one controlled by the flow proportional servovalve, using the information supplied by a pressure transducer, U_P. The pressure limiting system is a proportional one, not needing a PID error amplifier, but only a time modulator of the command voltage of the flow proportional servovalve (Fig. 5).
Other manufacturers prefer the closed loop pressure control, after filtering the signal supplied by a piezo ceramic pressure transducer and the precise control of the spool position of the proportional servovalve, by a position feedback obtained with an inductive low voltage displacement transducer (LVDT). From the point of view of unitary information processing and pump capacity control, this solution is optimal, but its cost is bigger and the safety level is reduced. The mechanical-

hydraulic pressure compensator offers a higher level of safety than the electronic one in the case of accidental working fluid degradation.

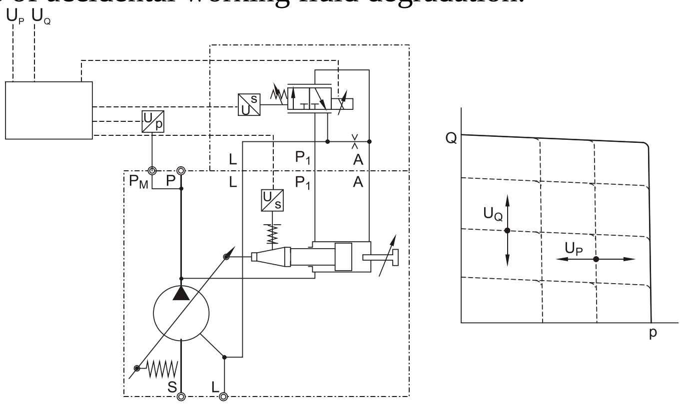

Fig. 5 Open circuits' electrohydraulic servopumps hydraulic scheme and steady state characteristics obtained by a dedicated digital controller from the family P-Q

Source: PARKER HANNIFIN CORPORATION

This paper presents the modelling, simulation, and experimental validation of the model of the servopump PV046R1K1T1NUPG performed by the authors in the Electrohydraulic Amplifier Performance Certification Laboratory included in the Romanian Accreditation Association - RENAR [17]. The experimental results are found in good agreement with the theoretical ones. The simulation model was included in the AMESIM super components library [7], [8].

2. Numerical simulation of the servopump dynamics

The numeric simulation model from Figure 6 corresponds to the servopump which is mounted on the hydraulic amplifier test bench from the laboratory designed by the authors. The proposed simulation model was included in the AMESim Model library under the name *spehd.ame* [8]. The modelling problems were solved taking into account some technical bulletins published by SOCIETE IMAGINE [2] … [6].

Figures 7…12 show the evolution of the main parameters which define the dynamics of the servopump for a step voltage input signal. It is noted that for small signals, the servomechanism responds like a second order system, while for greater amplitudes its behavior is aperiodic. The time constant is less than 50 ms, quality which allows the use of the servopump in fast positioning systems.

The next Figs. (13 to 22) are presenting the effect of the solenoid force variation (Fig. 13) on the control valve spool on different servopump parameters. The response of the solenoid for a step input current generated by the servo controller is a first order one with a constant time of about 0.035 s [1].

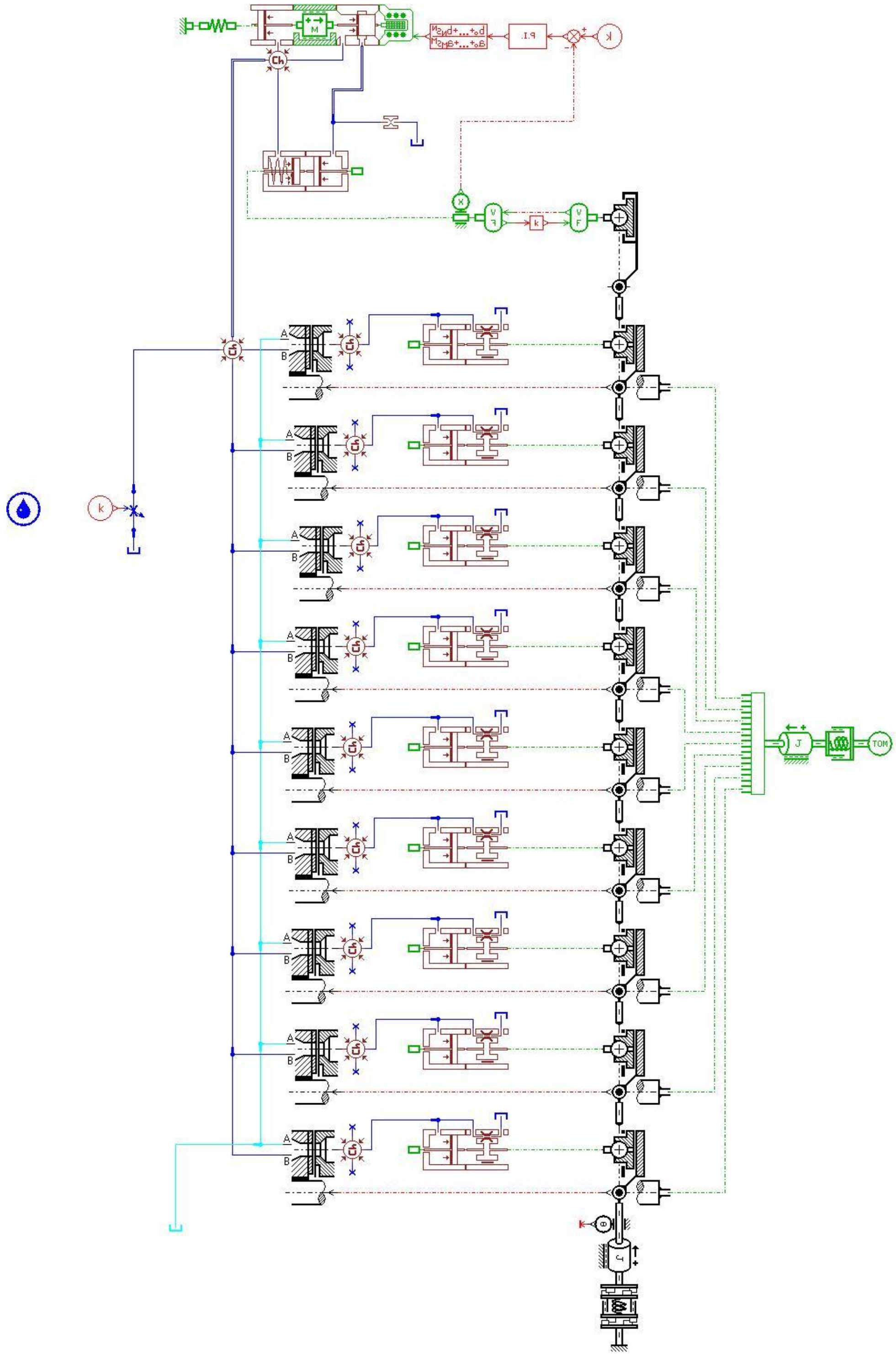

Fig. 6 The simulation model of a swash plate servopump built in AMESIM [11].

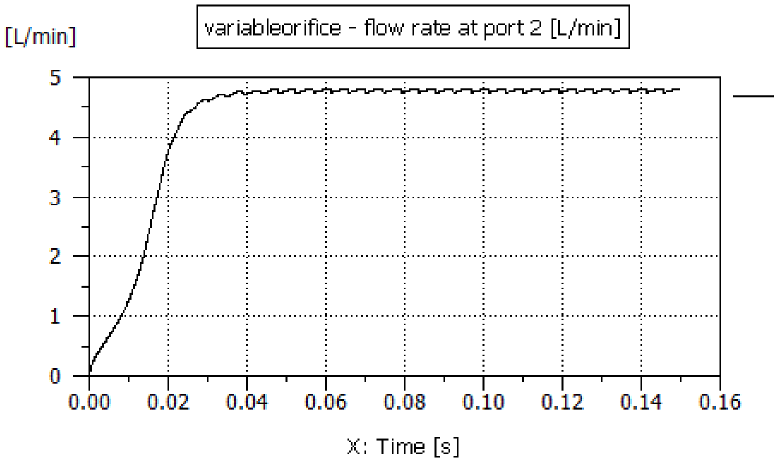

Fig. 7 Servopump flow rate delivered to a small variable orifice for a small input voltage (0.5 V)

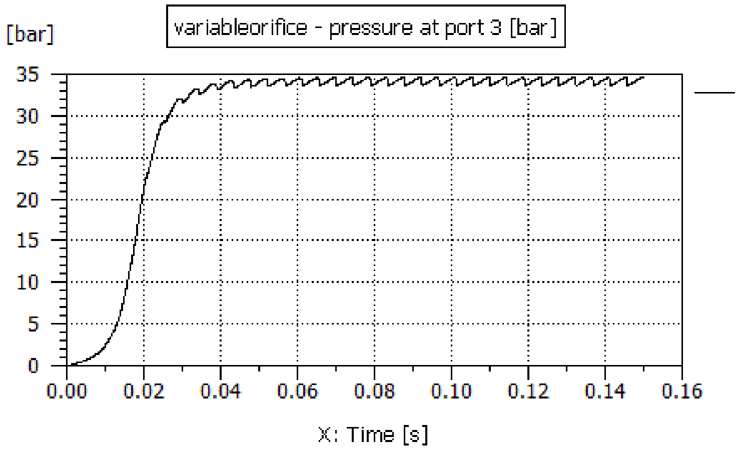

Fig. 8 Servopump output pressure for a small input voltage of 0.5 V (the flow is supplied to a small variable orifice)

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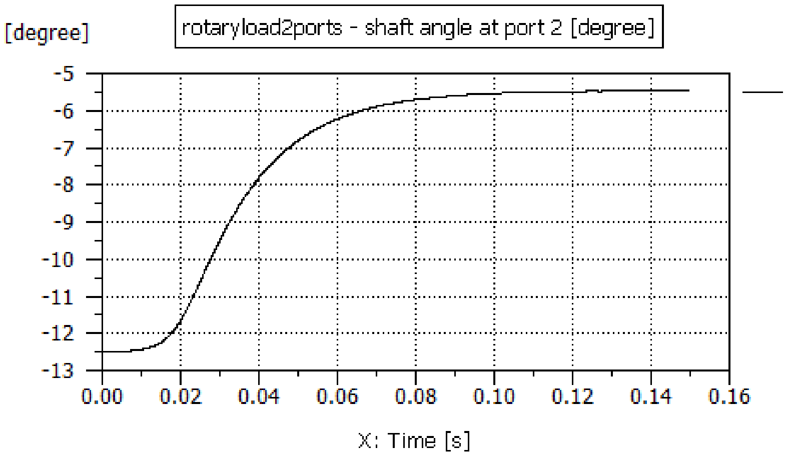

Fig. 9 The swash plate angle variation during the transient

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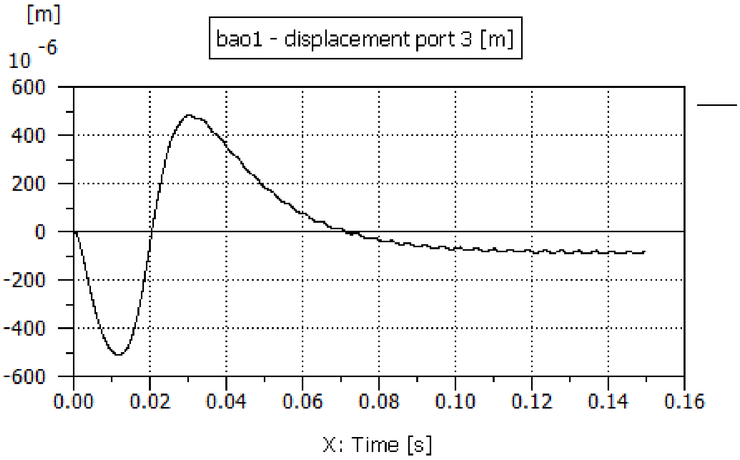

Fig. 10 Control valve spool displacement during the transient

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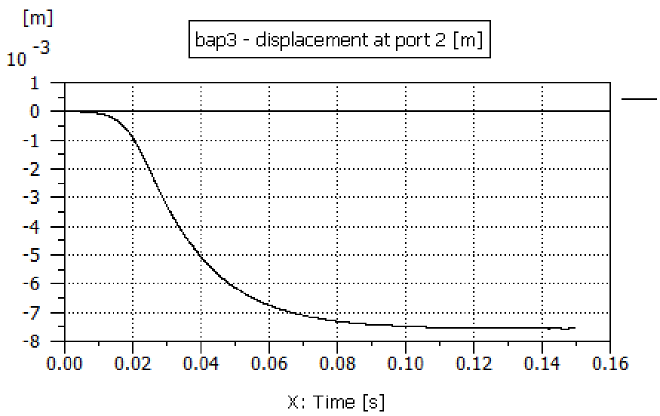

Fig. 11 Piston displacement during the transient

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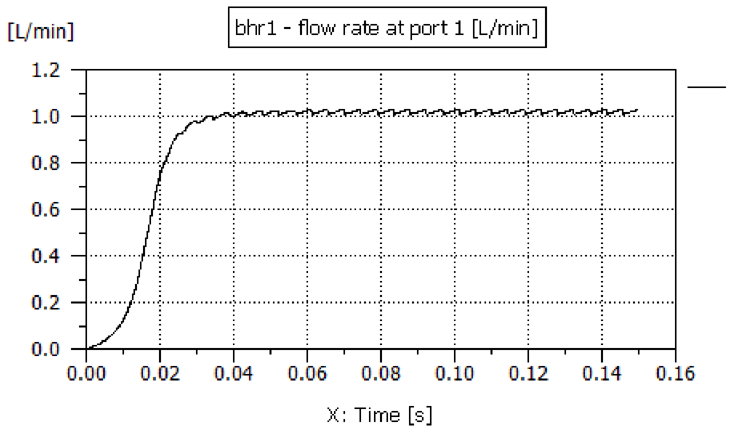

Fig.12 Flow rate passing through the flow control valve during the transient

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The effect of the solenoid force variation on the control valve spool, shown in figure 13, on different servopump parameters is presented in the next figures (14 to 22). The response of the solenoid for a step input current generated by the servo controller is a first order one with a constant time of about 0.035 s [1].

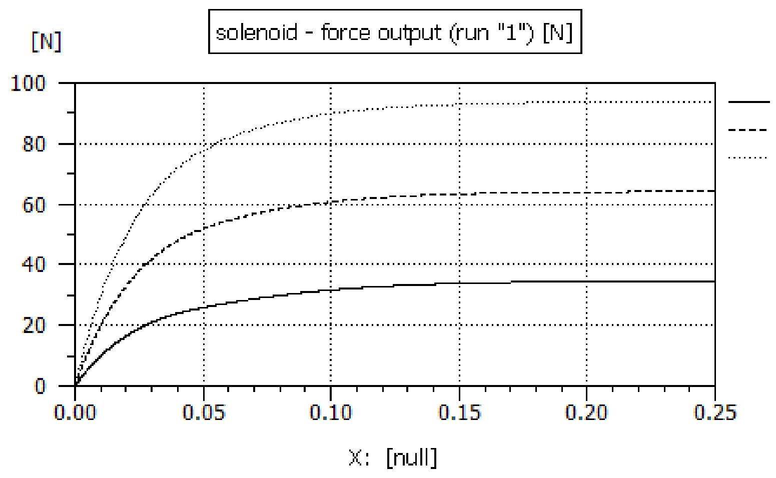

Fig. 13 Response of the solenoid force for a step current input

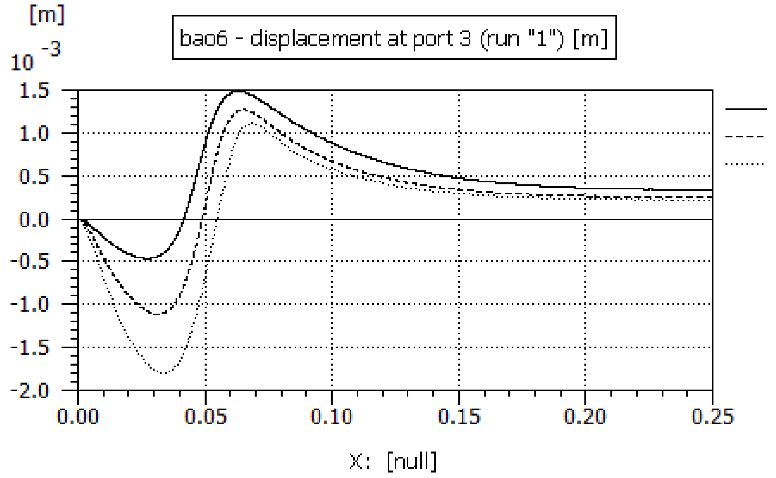

Fig. 14 Control valve spool displacement during the transients generated by the solenoid step force variation

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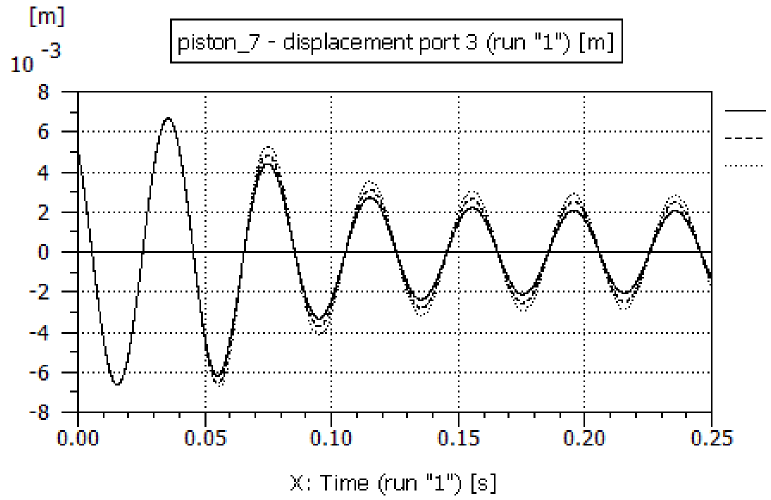

Fig. 15 Stroking piston displacement during the transients

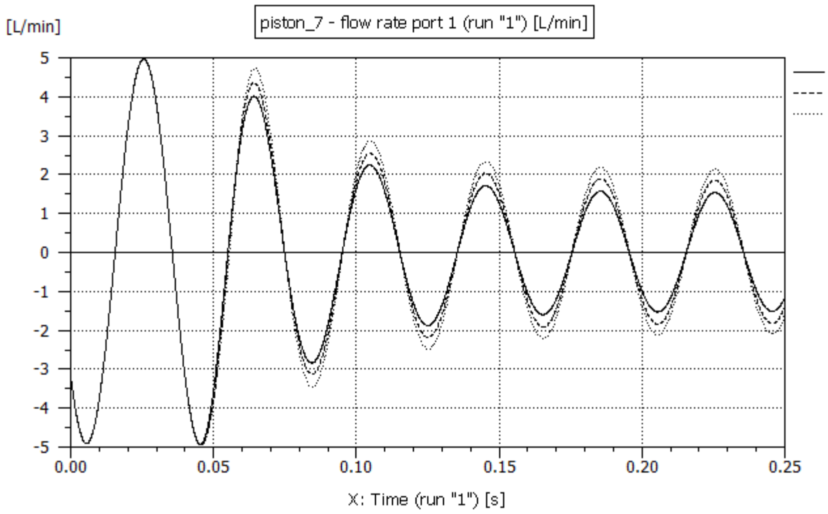

Fig. 16 Flow rate at the input port of the stroking cylinder

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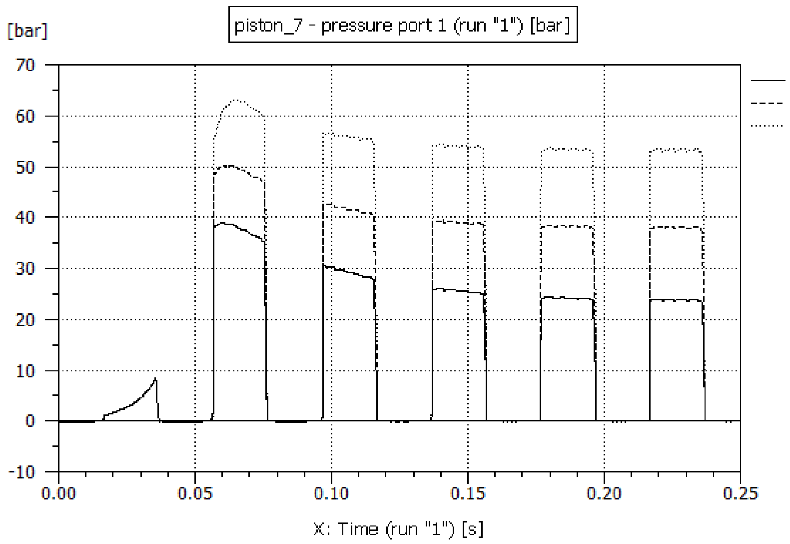

Fig.17 The pressure variation in a cylinder of the barrel during the transients

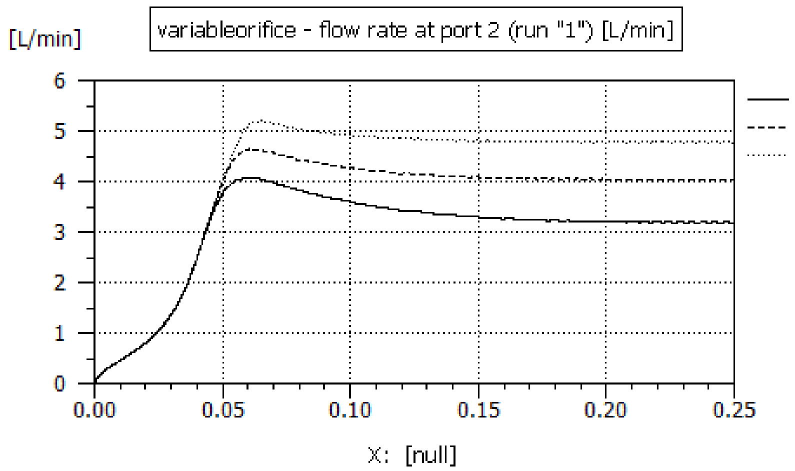

Fig. 18 Flow rate delivered by the servopump to the variable load orifice

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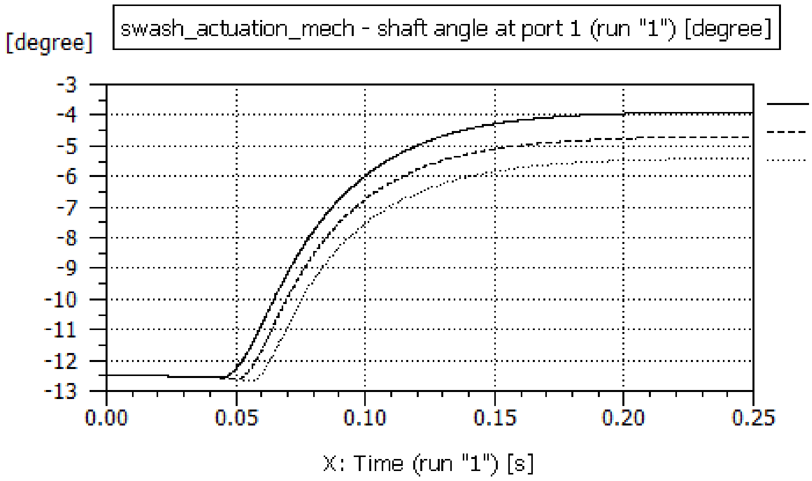

Fig. 19 Variation of the angle of the swash plate during the transients

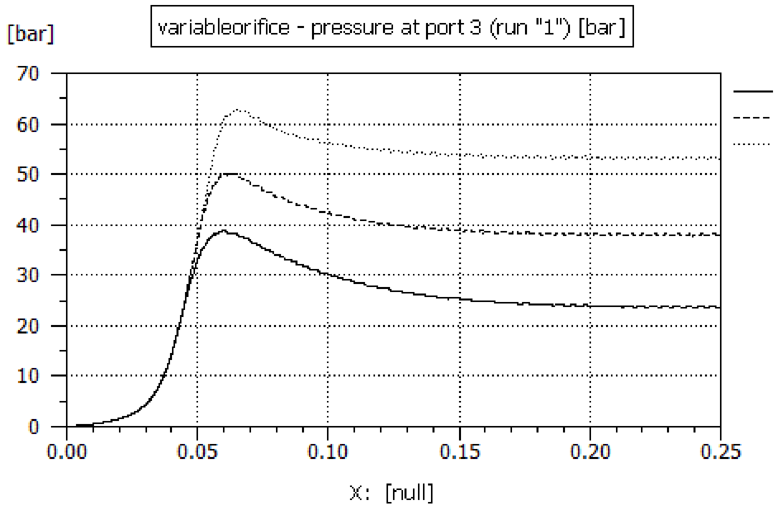

Fig. 20 Pressure variation at the input of the load orifice

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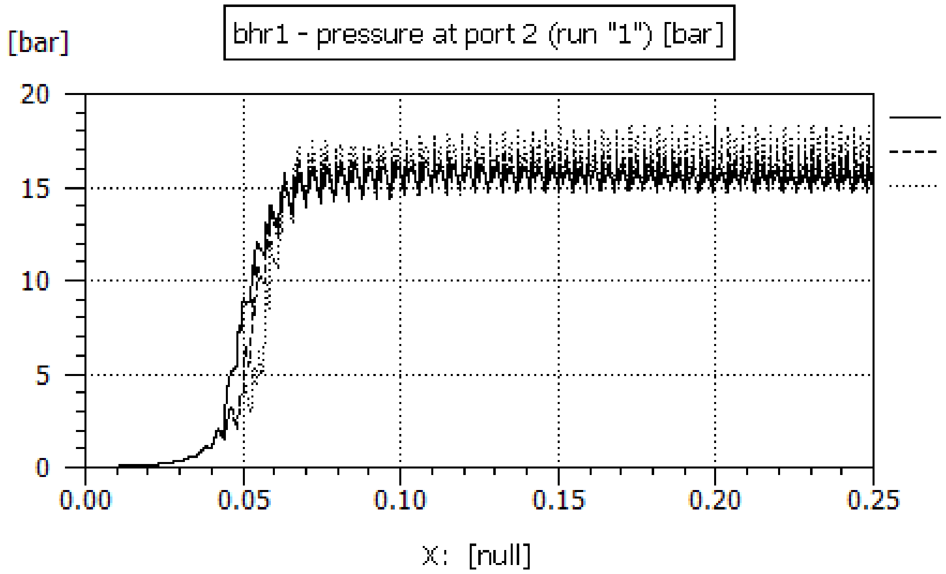

Fig. 21 Pressure variation in the passive chamber of the swash plate actuator during the transients

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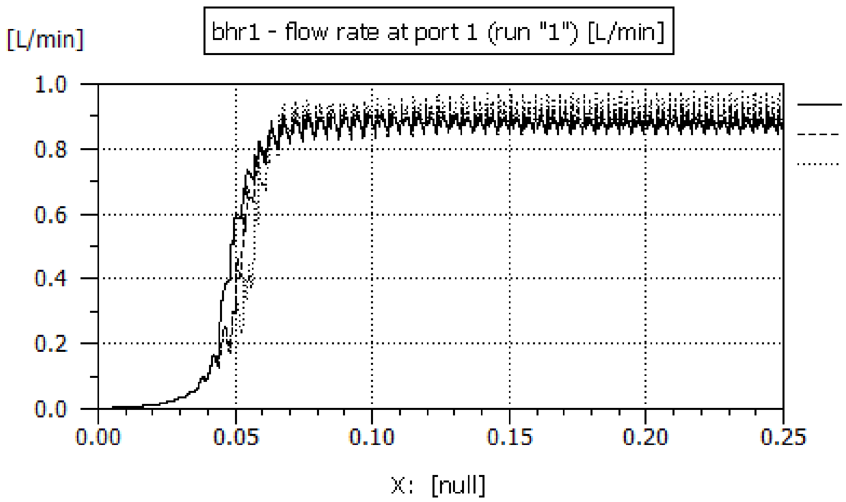

Fig. 22 The flow rate consumed by the servopump through the hydraulic resistance of the passive actuator chamber during the transients

Figs. 23, 24 and 25 present the simulated behavior of the servopump flow rate for periodic signals applied to the servo controller, generating periodical solenoid forces. Three types of signals currently used in practice were applied: sinusoidal, triangular and rectangular ones. A very good tracking capacity is observed, with a maximum delay of 100 ms.

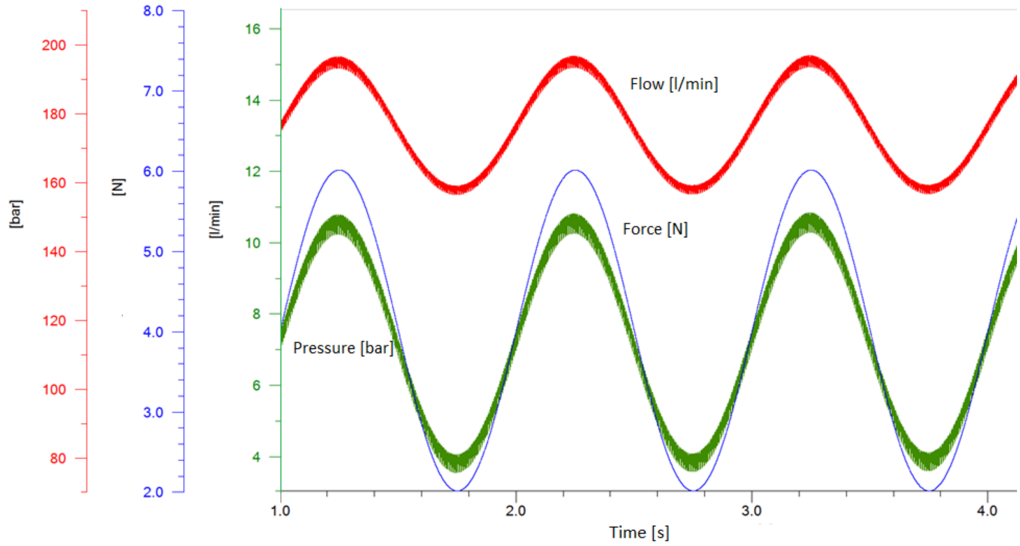

Fig. 23 Servopump simulated response for sine input signal (f = 1.0 Hz)

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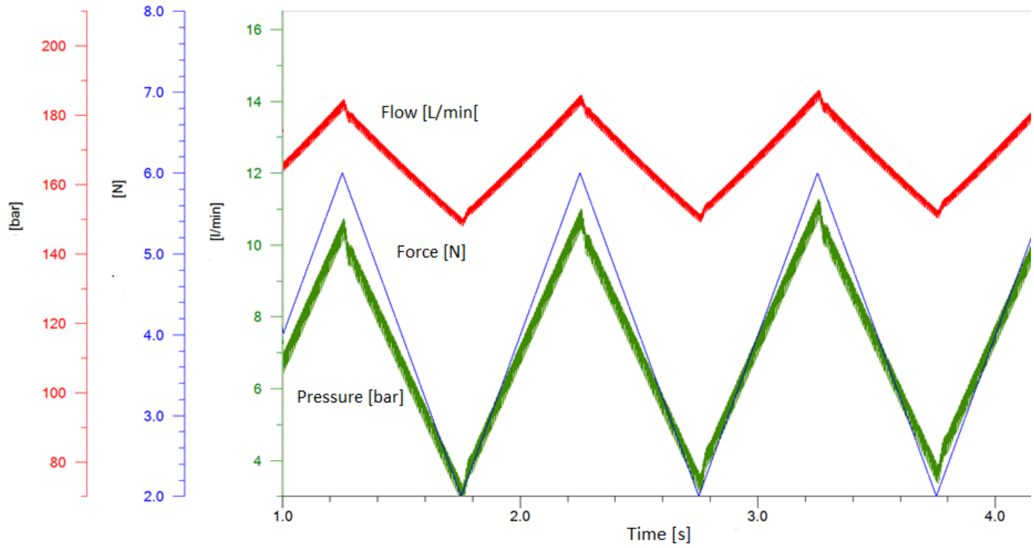

Fig. 24 Servopump simulated response for triangular signals (f = 1Hz).

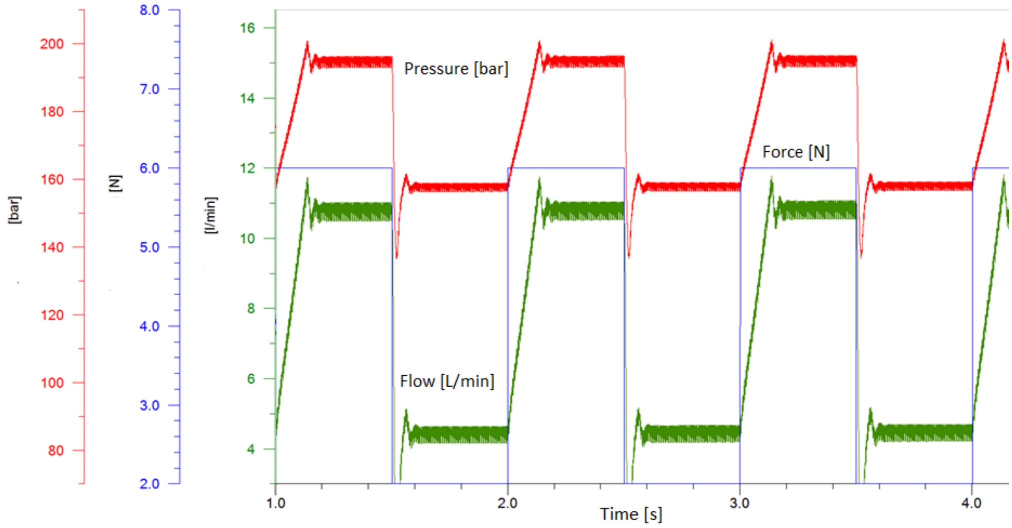

Fig. 25 Servopump response for rectangular signals (f = 1 Hz).

3. Experimental validation of the simulations

The flow control module of the servopump (Fig. 26) was tested using sinusoidal and triangular waveforms supplied by a STANDFORD SYSTEMS precision analog waveform generator. The medium servopump load pressure was generated using two types of resistances:

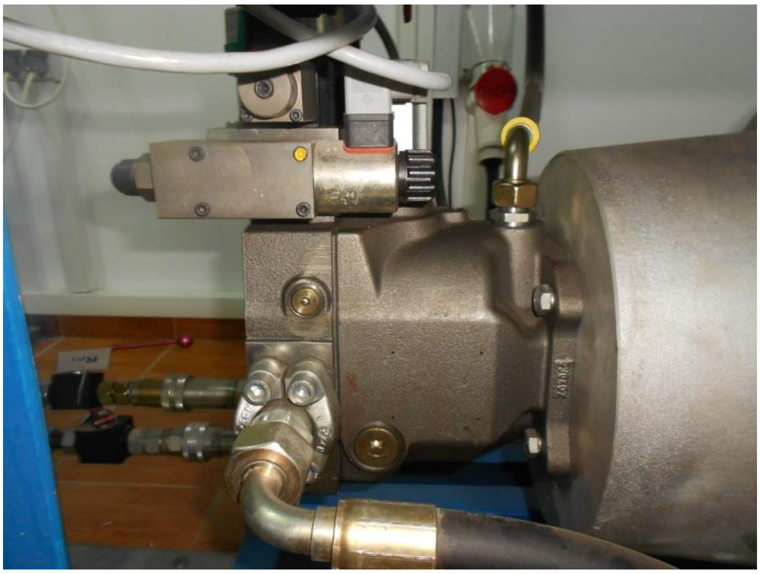

Fig. 26 Lateral view of the tested servopump PV046R1K1T1NUPG.

a) A high-dynamics two-stage servovalve from Parker (SE2E), which includes a digital servo controller; this can be voltage controlled, so it is compatible with the signal generator;
b) A high-dynamics proportional industrial servovalve DFPlus (DDV) with connected load ports, with variable opening. In both cases the control voltage was recorded by a PXI based data acquisition system with LabVIEW software [15]. All experiments indicate a very good dynamic behavior of the flow servopump. The 100…200 ms delay which depends of the frequency of the excitation signal can be neglected in all industrial applications. Fig.. 27, 28 and 29 present some of the representative dynamic tests.

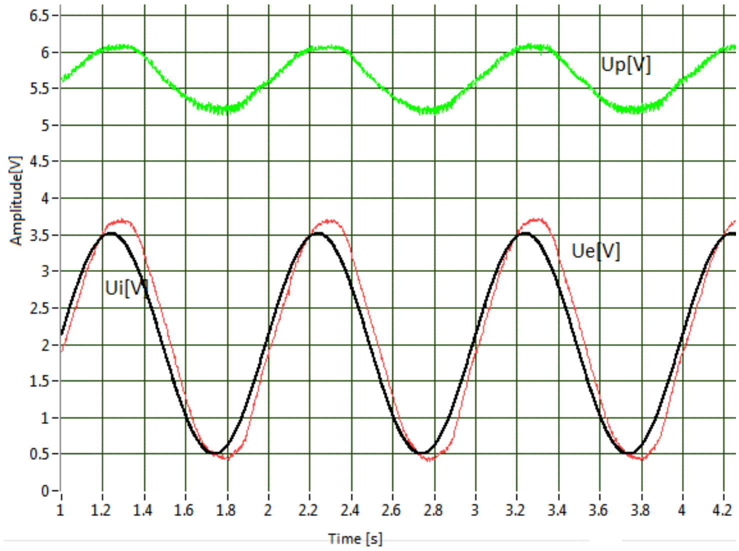

Fig. 27 Experimental frequency response for sine input with
f = 1.0 Hz and p_{medium} = 196.0 bar.

4. Conclusions

Industrial automatic electrohydraulic systems development essentially depends on the progress in the field of the automatic digital systems, which include hybrid interfacing elements like the high-speed distribution valves or the fast-response flow servopumps, with automatic pressure and power control. The most promising research is being done for integrating the electrohydraulic automated systems with electrical ones, creating hybrid systems which are interconnected through CAN networks, adapted to modern real-time systems [9], [10]. In the modern high complexity equipment field as well as in all the hydraulic energy consumers, the optimal control of primary energy sources by distributed and hierarchized computing systems is continuously researched. A good example of progress is the Ethercat system which is an Ethernet-based fieldbus system, invented by Beckhoff Automation [16]. The protocol is standardized in IEC 61158 and is suitable for both hard and soft real-time requirements in automation technology.

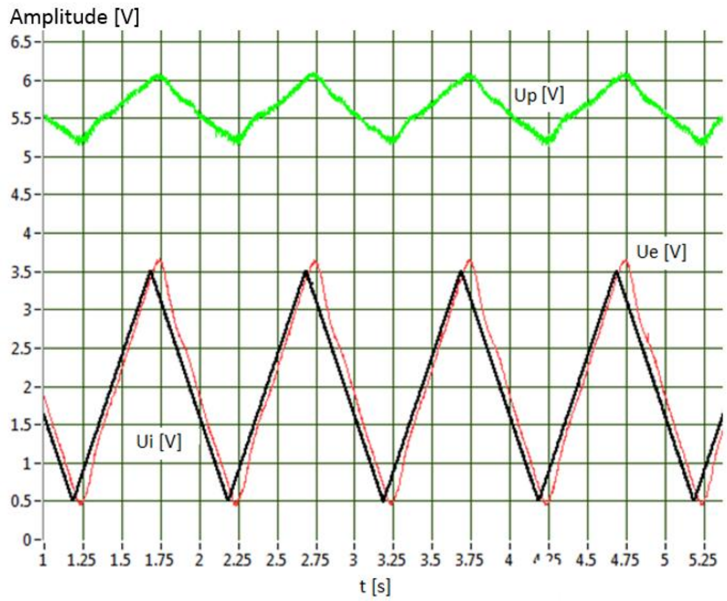

Fig. 28 Experimental frequency response for triangle input with
f = 1.0 Hz and p_{medium} = 196.0 bar

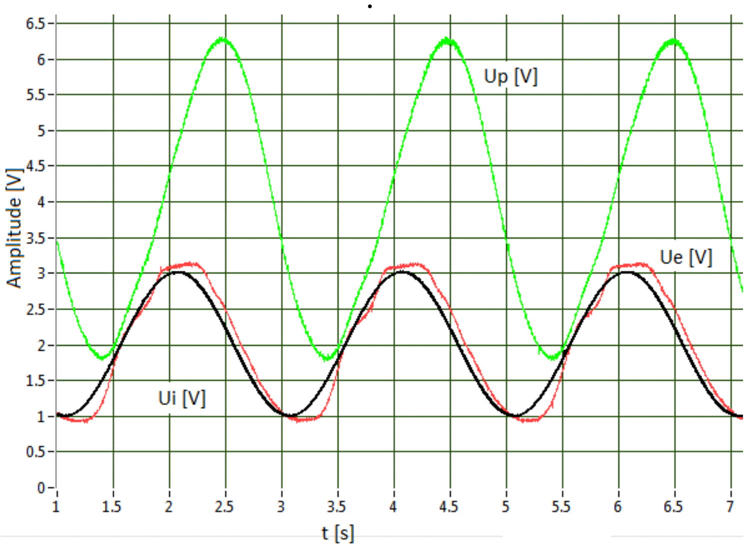

Fig. 29 Experimental frequency response for f=0.5 Hz and p_{medium}=140 bar
generated by a DFPlus industrial servovalve with connected load ports.

R E F E R E N C E S

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