Precise force control for hydraulic and pneumatic press system

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Precise force control for hydraulic and pneumatic press system

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Abstract This paper presents the force control methods for a pneumatic and a hydraulic press. Both systems have been made for educational purposes as well as for experimental testing and verification of different control techniques.

The pneumatic press contains a proportional pressure valve which is used for pressure regulation in a cylinder chamber and has direct impact on controlled force. The hydraulic press contains a servo-solenoid pressure-control valve for regulating the cylinder pressure. The pressing force can also be indirectly measured by a pressure transducer which is installed in the cylinder chamber.

Experimental tests have shown that electrically actuated control components supported by the appropriate measuring devices and computer programs make it possible to improve the characteristics of the hydraulic and pneumatic systems required in modern industrial plants.

Keywords: • press • hydraulic and pneumatic • force control • pressure sensor • force sensor • simulation •

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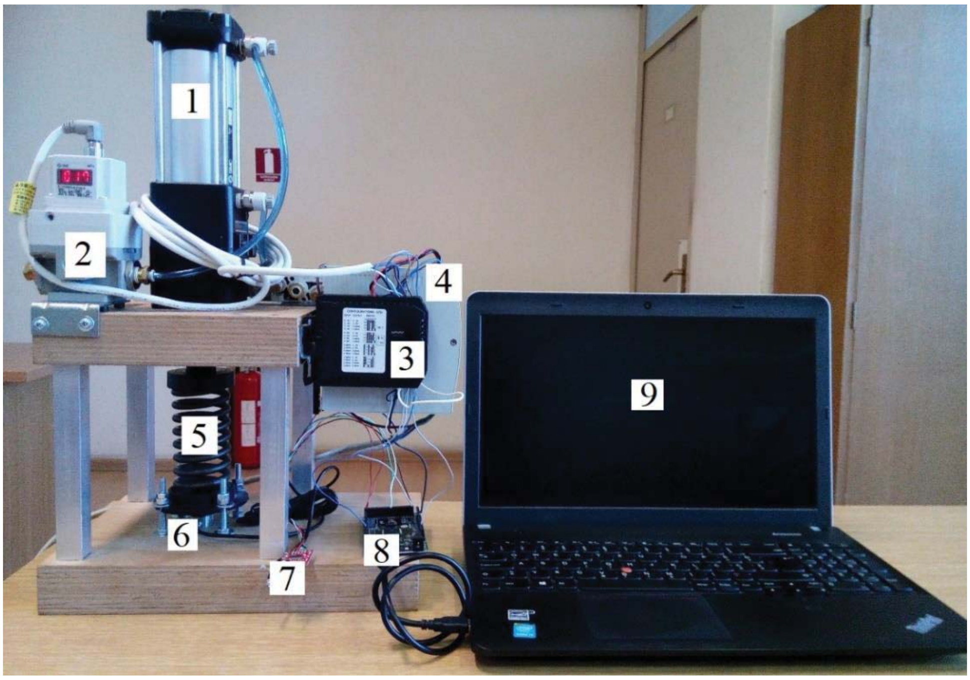
1 Introduction

Presses are one of the most used machine tools in industry for different materials forming. In the past, for pressing tasks in industry, mechanical presses were more frequently used, but nowadays, especially for large pressing force, hydraulic presses take precedence due to their numerous advantages, such as full force through the stroke, the moving parts operate with good lubrication, the force can be programmed, the stroke can be fully adjustable which contributes to the flexibility of application. Also, hydraulic presses can be made for huge force capacities. On the other hand, hydraulic presses are generally slower than mechanical presses, which is overcome with the development of new valves with higher flow capacities, smaller response time and improved control capabilities. Pneumatic presses can be used for less force capabilities, but they can move several times faster than hydraulic presses. Pneumatic presses are extremely adaptable to different process requirements and they are very low-maintenance. Most hydraulic and pneumatic presses used in industry are working in an open-loop and are usually operated manually or using a control device such as programmable logic controller (PLC), where the end positions of the ram are set via dead stops or limit switches. Today's industry is looking for flexible solutions that will be able to achieve some new characteristics of hydraulic and pneumatic systems, such as the ability of controlled motion, the possibility of continuous control of the required values, a simple data transfer and signal processing, the possibility of monitoring and process visualization etc. The increase of micro-electronics in recent years has reduced the cost of computer equipment to a level acceptable for industrial applications which has enabled the implementation of sophisticated control strategies in practice. Therefore, modern hydraulic and pneumatic systems suffered a great evolution towards electronics and microprocessor controlled electro-hydraulic components in order to achieve new control possibilities [1]. Commonly, due to its complexity, almost every advanced controller must be implemented on a digital computer. Such control systems, that have electrically-actuated valves, can respond to complex demands posed by today's technology. The ability of force control or positioning control systems to follow-up varying reference signals is often required for proper operation of the technological process [2], [3]. Therefore, a new quality and significant improvement in the functioning of presses are obtained by applying the force feedback, where the force output is measured by a load cell and the cylinder movement is controlled via proportional or servo valve [4], [5], [6].

This paper presents basic construction of a pneumatic and a hydraulic press and the implementation of force control algorithms. Control algorithms and data monitoring are implemented on a real-time hardware board. The control algorithms are tested via numerical simulations, as well as experimentally to verify their practical use and effectiveness.

2 Experimental setup

A photo of the pneumatic press on which experiment is carried out is shown in Figure 12. A pneumatic double acting cylinder (1) with 100 mm stroke and 50 mm bore is used as actuator to convert compressed air into mechanical power. An electro-pneumatic regulator (2), SMC ITV3050, is used for controlling the pressing force. It has current type input signal in the range 4÷20 mA for the pressure output range of 0÷9 bar and gives an analogue (monitor) output signal in the range of 1÷5 V proportional to the pressure output. The force acting on the spring (5) is directly measured by a disc load cell (6), type TAS606. Rated load of the load cell is 200 kg, while the maximum output is 7.5 mV when the sensor is connected to 5 V power supply from Arduino board (8). The load cell analogue to digital converter with amplifier (7), type HX711, is used to amplify the load cell output signal. The Arduino Mega 2560 board (8) is used for data acquisition which offers 16 analogue inputs with 10-bit resolution, 54 digital input/output pins (of which 15 provides PWM output) and USB connection to the control computer (9). A signal converter (3) is used to convert Arduino 0÷5 V voltage output to 4÷20 mA current input signal for the proportional pressure-control valve. Both the signal converter and the proportional pressure-control valve are powered by 24 V power supply (4). The control algorithm is implemented in Matlab/Simulink software, which allows executing the simulation program in external mode, similar to the Real-Time Workshop tool.



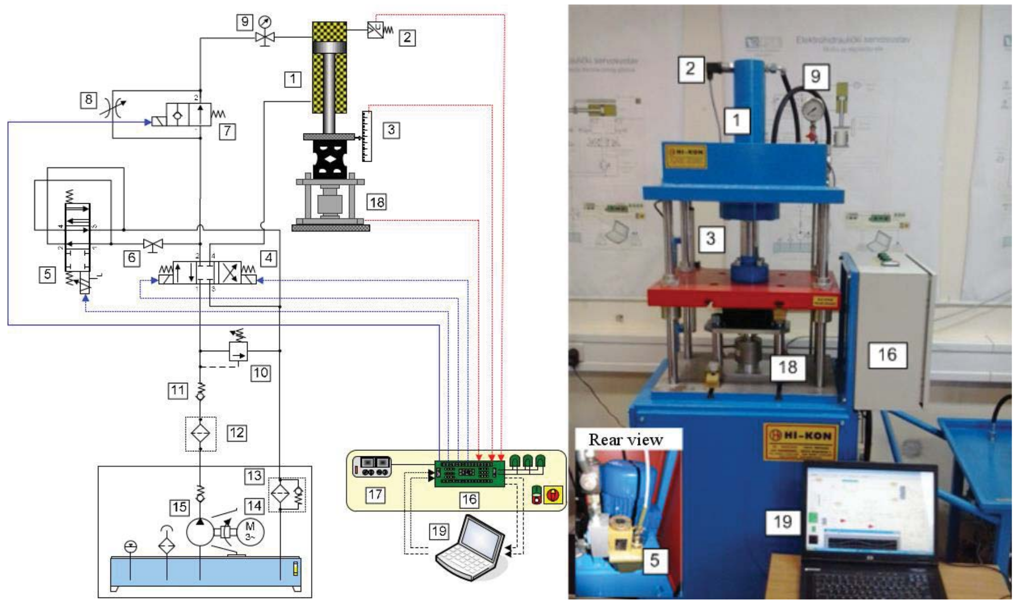
1-Pneumatic cylinder, 2-Proportional pressure-control valve, 3-Signal converter, 4-Power supply, 5-Spring, 6-Load cell, 7-A/D converter, 8-Arduino board, 9-Control computer

Figure 12: Pneumatic press

A schematic diagram and a photo of the hydraulic press on which experiments have been carried out are shown in Figure 13. The hydraulic cylinder (1) which is used to actuate the press is a double acting 300 mm stroke cylinder with 80 mm bore and 60 mm diameter rod. The control of the pressing force is accomplished using an electro-hydraulic servo valve (5) with a box chopper amplifier and ±10 V analogue input signal. Maximum pressure in the system is limited by a pressure relief valve (10). The force acting on the rubber bumper is directly measured by the force sensor (18), which is a compression load cell. The pressure inside the cylinder chamber is measured by a pressure transducer (2), with a measuring range 0 to 250 bar and an output signal 0 to 10 V. In this system it is also possible to measure displacement of the press by using a micro-pulse linear transducer (3). Since the servo valve is installed in the system, particular attention should be given to ensure the cleanliness of oil, so a high pressure filter (12) and a return flow filter (13) are set in the hydraulic circuit. The hydraulic power is provided by a hydraulic gear pump (15), with a volumetric displacement of the pump of 2.6 cm^3/rev and the maximum nominal pressure of 25 MPa. Data acquisition in the system is handled by a National Instruments DAQCard-6024E (for PCMCIA), which offers both 12-bit analogue input and analogue output. Control algorithms were developed in the Matlab/Simulink environment supported by Real-Time Workshop (RTW) program. Detailed description of the system can be found in [5].

The considered experimental electro-hydraulic and electro-pneumatic systems have been made in the Laboratory for automation and robotics at the Faculty of Mechanical Engineering and Naval

Architecture. The modules are used for research purposes in the field of hydraulic and pneumatic systems control, as well as for training students.



1-Hydraulic cylinder, 2-Pressure transducer, 3-Micro-pulse linear transducer, 4-Solenoid 4/3 valve, 5-Servo valve, 6-Shut-off valve, 7-Solenoid 2/2 valve, 8-Throttling valve, 9-Manometer, 10-System pressure relief valve, 11-Ball check valve, 12-Pressure filter, 13-Return flow filter, 4-Three-phase electric motor, 15-Hydraulic pump,16-Electronic interface, 17-Electric rectifier, 18-Force sensor (load cell), 19-Control computer

Figure 13: Hydraulic press, a) schematic diagram, b) photo

3 System modelling

3.1 Pneumatic press system modelling

The problem under consideration is shown in Figure 14. The goal is to make the actuator compress an object with a desired force. The force producing element is a double-acting pneumatic cylinder driven by a proportional pressure-control valve. The mathematical model of the pneumatic actuator interaction with its environment is given bellow. The force balance equation for the mechanical part of the system can be written as follows:

$$m\,\ddot{x} = F_{\mathrm{p}} - b\,\dot{x} - k_{\mathrm{e}}\,x \tag{1}$$

where m, b, x represent the mass, damping and displacement of the actuator, while k_e represent spring stiffness. The actuator force F_p allows interaction between the cylinder piston and its environment and its given by:

$$F_{\mathrm{p}} = A(p - p_a) \tag{2}$$

where A is piston area, p is inlet pressure and p_{a} is atmospheric pressure.

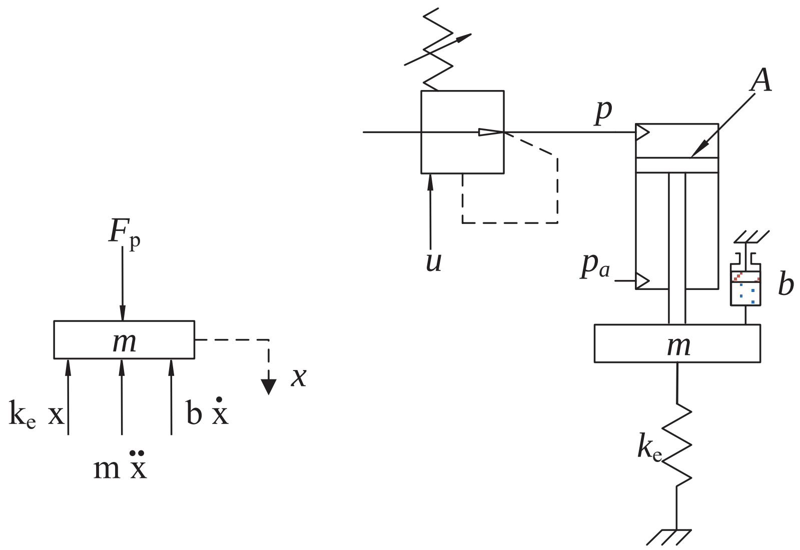


Figure 14: Schematic model of the pneumatic press

The output of interest on the experimental test system is the force F of the pneumatic press and that force is equal to the compression of a spring x multiplied by its stiffness k_e:

$$F = k_e\, x \tag{3}$$

From (3) the velocity and acceleration are given by:

$$\dot{x} = \frac{\dot{F}}{k_e} \text{ and } \ddot{x} = \frac{\ddot{F}}{k_e} \tag{4}$$

Substituting equations (2), (3) and (4) into equation (1) gives the force balance equation for the pneumatic actuator and it can be written as follows:

$$\frac{m}{k_e}\ddot{F} = F_p - \frac{b}{k_e}\dot{F} - F$$

$$\ddot{F} = \frac{k_e}{m} A(p - p_a) - \frac{b}{m}\dot{F} - \frac{k_e}{m} F \tag{5}$$

The mathematical model of the proportional pressure control valve is obtained from experimentally measured pressure response on a step command input. The pressure responses for step input signals with amplitudes of 1 V, 2 V and 5 V are shown in Figure 4. The pressure transient-response of the valve has an aperiodic form, and can be approximated by a first-order lag term as follows:

$$T_v \frac{dp(t)}{dt} + p(t) = K_v\, u(t) \tag{6}$$

where T_v is a time constant and K_v is the gain of the proportional pressure-control valve.

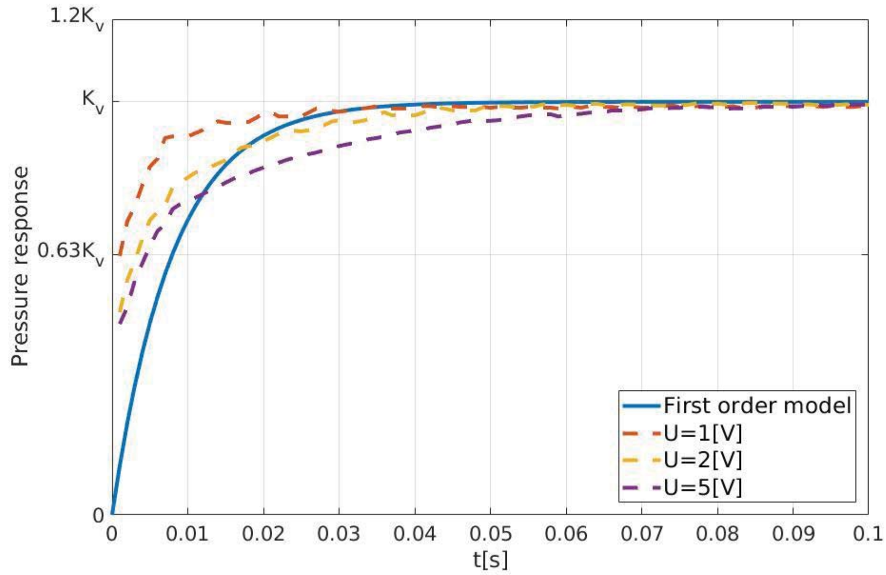


Figure 15: Step response of the proportional valve

The force control loop uses a PID controller given by the following expression:

$$u(t) = K_\mathrm{p}\, e(t) + K_\mathrm{i} \int e(t)dt + K_\mathrm{d}\, \dot{e}(t) \tag{7}$$

where K_p, K_i, K_d represents proportional, integral and derivate gain and $e(t)$ represents the control error between the measured force and the desired force trajectory given by:

$$e(t) = F_\mathrm{r}(t) - F(t) \tag{8}$$

The mathematical model of the system can be written in the standard state-space form as $\dot{\boldsymbol{x}} = \boldsymbol{f}(\boldsymbol{x}(t), \boldsymbol{u}(t))$ where state variables $\boldsymbol{x} = [x_1\; x_2\; x_3\; x_4]^T$ are chosen as: $x_1 = p$, $x_2 = F$, $x_3 = \dot{F}$ and $x_4 = \int e(t)$, then the simplified representation of the actual system dynamics which includes the controller can be written in the state-space form as follows:

$$\begin{aligned}
\dot{x}_1 &= \frac{K_\mathrm{v}}{T_\mathrm{v}}\, u(t) - \frac{1}{T_\mathrm{v}}\, x_1 \\
\dot{x}_2 &= x_3 \\
\dot{x}_3 &= \frac{k_\mathrm{e}}{m}\, A(x_1 - p_a) - \frac{b}{m}\, x_3 - \frac{k_e}{m}\, x_2 \\
\dot{x}_4 &= e(t) = F_\mathrm{r}(t) - F(t)
\end{aligned} \tag{9}$$

The state-space model is given in general nonlinear form and it is used for numerical simulation of the pneumatic press. The desired output of the system is the force F (variable x_2).

3.2 Hydraulic press system modelling

For the hydraulic press system, the considered model and the obtained solution are similar to the previously presented model for the pneumatic press, as shown in Figure 16. The force producing element, similar to the pneumatic press, is a double acting hydraulic cylinder controlled by a pressure type servo valve. The simplified nonlinear mathematical problem describing the hydraulic actuator interacting with its environment is derived below and the complete mathematical model is given in [7].

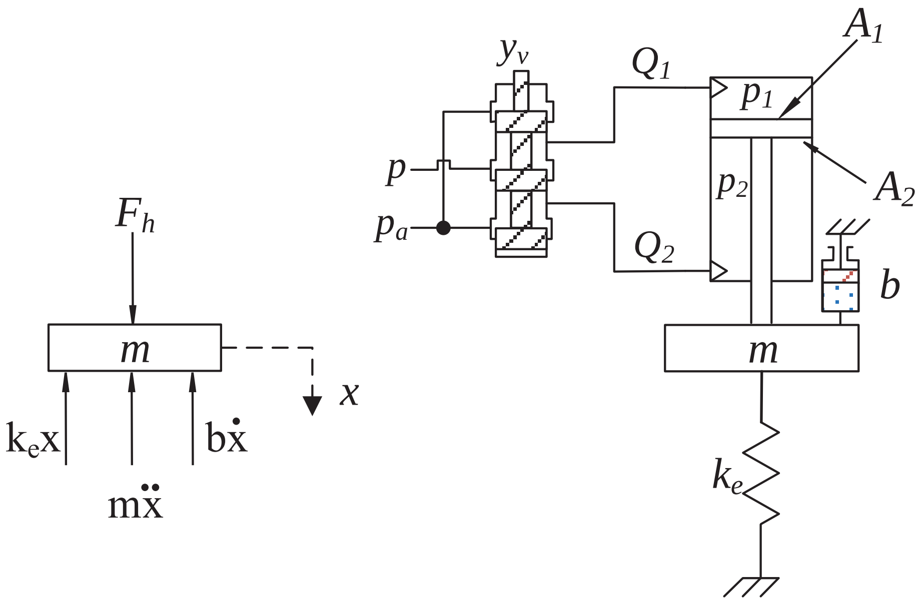


Figure 16: Schematic model of the hydraulic press

The force balance equations for the mechanical part of the system can be written as follows:

$$m\,\ddot{x} = F_{\mathrm{h}} - b\,\dot{x} - k_{\mathrm{e}}\,x - F_{\mathrm{fc}}(\dot{x}) \tag{10}$$

where, similar to the pneumatic press model, *m, b, x* represent the mass, damping and displacement of the actuator, while k_e represents the spring stiffness. Here we assume a classical friction model that includes the viscous and Coulomb friction, F_v and F_{fc} respectively, and is given by:

$$F_{\mathrm{friction}}(\dot{x}) = F_{\mathrm{v}} + F_{\mathrm{fc}}(\dot{x}) = b\,\dot{x} + f_{\mathrm{c}}\,sgn(\dot{x}) \tag{11}$$

The actuator force F_h allows interaction between the piston and its environment and is given by:

$$F_{\mathrm{h}} = p_1 A_1 - p_2 A_2 \tag{12}$$

where A_1 and A_2 are the cross sectional piston areas, p_1 is the inlet pressure and p_2 is the outlet pressure. The output of interest is similar to the case of the pneumatic press and it's actually the force F of the hydraulic press. It is supposed that the force is equal to the compression of the rubber spring x multiplied by its stiffness k_{e}:

$$F = k_{\mathrm{e}}\,x \tag{13}$$

From (13) the velocity and acceleration are as follows:

$$\dot{x} = \frac{\dot{F}}{k_{\mathrm{e}}} \text{ and } \ddot{x} = \frac{\ddot{F}}{k_{\mathrm{e}}} \tag{14}$$

Substituting equations (12), (13) and (14) into equation (10) gives the force balance equation for the hydraulic actuator and it can be written in following form:

$$\frac{m}{k_\mathrm{e}}\ddot{F} = F_\mathrm{h} - \frac{b}{k_\mathrm{e}}\dot{F} - F - F_\mathrm{fc}(\dot{x})$$
$$\ddot{F} = \frac{k_\mathrm{e}}{m}(p_1 A_1 - p_2 A_2) - \frac{b}{m}\dot{F} - \frac{k_\mathrm{e}}{m}F - \frac{k_\mathrm{e}}{m}F_\mathrm{fc}(\dot{x}) \tag{15}$$

From [7] the expression for the piston and rod side actuator pressures may be written as follows:

$$\dot{p}_1 = \frac{\beta}{V_1}\left(Q_1 - \dot{V}_1\right) = \frac{\beta}{V_0 + A_1 x}\left(Q_1 - A_1\dot{x}\right) \tag{16}$$

$$\dot{p}_2 = \frac{\beta}{V_2}\left(Q_2 - \dot{V}_2\right) = \frac{\beta}{V_0 - A_2 x}\left(-Q_2 + A_2\dot{x}\right) \tag{17}$$

where V_1 and V_2 are total fluid volumes in the two cylinder chambers, V_0 is the half-volume, A_1 and A_2 are the annulus area of the piston and rod side of the cylinder.

If we suppose that the bandwidth of the servo valve is much higher than the dynamics of the control system, then the valve dynamics can be neglected and the flow is taken as a linear function of the control signal u:

$$Q(y_\mathrm{v},\ p) = K\,u \tag{18}$$

where K is the flow/signal gain of the valve and its value is estimated from the flow/signal characteristic curve.

If we derive the expression for F_h and substitute equations (16), (17) and (18) into equation (12) then the acting force derivate is given by:

$$\dot{F}_\mathrm{h} = \dot{p}_1 A_1 - \dot{p}_2 A_2 = \frac{A_1\beta}{V_0 + A_1 x}(K u - A_1\dot{x}) - \frac{A_2\beta}{V_0 - A_2 x}\left(-\frac{K}{\varphi}u + A_2\dot{x}\right) =$$
$$= \left(\frac{\beta A_1 K}{V_0 + A_1\frac{F}{k_\mathrm{e}}} + \frac{\beta A_2\frac{K}{\varphi}}{V_0 - A_2\frac{F}{k_\mathrm{e}}}\right)u - \left(\frac{\beta A_1^2}{k_\mathrm{e}V_0 + A_1 F} + \frac{\beta A_2^2}{k_\mathrm{e}V_0 - A_2 F}\right)\dot{F} \tag{19}$$

If we choose PI controller, then the control signal will be equal:

$$u(t) = K_\mathrm{p}\,e(t) + K_\mathrm{i}\int e(t)dt \tag{20}$$

where K_p, K_i, represent proportional and integral gain of the controller and $e(t)$ represents the error between the measured force and the desired force trajectory given by the following expression:

$$e(t) = F_{\mathrm{r}}(t) - F(t) \tag{21}$$

Similar to the case of pneumatic press, the state-space model can be written as $\dot{\boldsymbol{x}} = \boldsymbol{f}(\boldsymbol{x}(t), \boldsymbol{u}(t))$ where state variables $\boldsymbol{x} = [x_1\ x_2\ x_3\ x_4]^T$ are chosen as: $x_1 = F, x_2 = \dot{F}, x_3 = F_{\mathrm{h}}$ and $x_4 = \int e(t)$, then the simplified representation of the actual system dynamics, which includes the controller, can be written in the state-space form as follows:

$$\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{k_{\mathrm{e}}}{m} x_3 - \frac{b}{m} x_2 - \frac{k_{\mathrm{e}}}{m} x_1 - \frac{k_{\mathrm{e}}}{m} F_{\mathrm{fc}}(x_2) \\
\dot{x}_3 &= -\left(\frac{\beta A_1^2}{k_{\mathrm{e}} V_0 + A_1 x_1} + \frac{\beta A_2^2}{k_{\mathrm{e}} V_0 - A_2 x_1} \right) x_2 + \left(\frac{\beta A_1 K k_{\mathrm{e}}}{k_{\mathrm{e}} V_0 + A_1 x_1} + \frac{\beta A_2 K k_{\mathrm{e}}}{(k_{\mathrm{e}} V_0 - A_2 x_1)\varphi} \right) u \\
\dot{x}_4 &= e(t) = F_{\mathrm{r}}(t) - F(t)
\end{aligned} \tag{22}$$

The state-space model is given in a general nonlinear form and it is used for numerical simulation of the hydraulic press. The desired output of the system is the force F (variable x_1).

4 Simulation model and numerical results

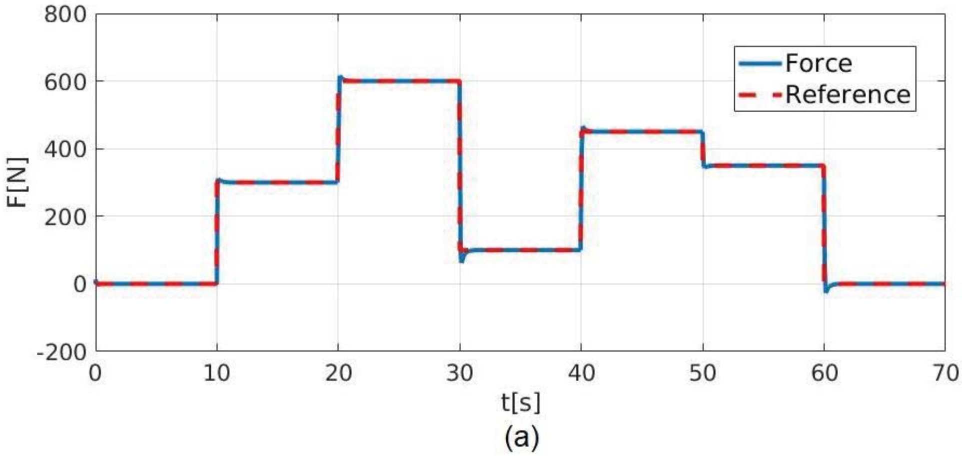
The numerical simulations are carried out in Matlab/Simulink program to verify the stability of the process for proposed controllers, both for pneumatic and hydraulic press. The simulations were performed using the Matlab ODE solver. In these examples, the relative and absolute tolerance for the ODE routine was set to 10^{-6}. The developed controllers were tested for a sinusoidal and square wave reference signal.

4.1 Pneumatic press simulation model and numerical results

The simulation was performed using parameters listed in Table 1, which were obtained from manufacture's data sheets and measurements, while corresponding results are shown in Figure 17. During simulation it was found that the values greater than 1 for proportional gain K_{p} and values greater than 2 for integral gain K_{i} of PI controller have resulted in a significant oscillatory response of the cylinder pressure. The simulation showed that the control algorithm could achieve good abilities for tracking square wave reference forces with fast response to changes in reference trajectory.

Table 1: Parameters for pneumatic press system

Parameter	Value	Parameter	Value
m	1 kg	T_{v}	0.008 s
A	$1.963 \cdot 10^{-3}$ m^2	K_{v}	0.18 MPa/V
k_{e}	80 N/mm	K_{p}	0.5
b	80 N/(m/s)	K_{i}	2
p_a	101 kPa	K_{d}	0.01



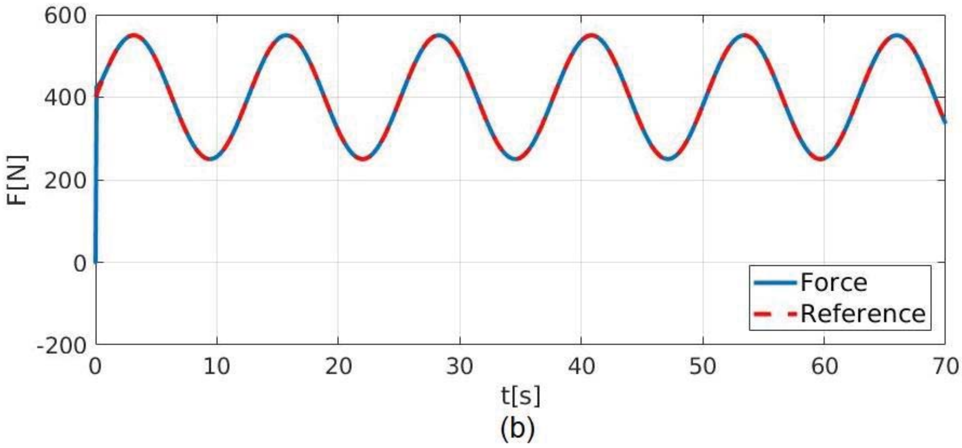


Figure 17: Pneumatic press simulation results, (a) square wave force tracking, (b) sine wave force tracking

4.2 Hydraulic press simulation model and numerical results

The numerical simulations for the hydraulic press were carried out with parameters listed inTable 2, which were obtained from manufacture's data sheets and measurements.

The results obtained through numerical simulations are shown in Figure 18. These results confirm that the PI controller couldn't follow square wave reference signal without tracking error and also unwanted high frequency oscillations occurred in the control variable.

Table 2: Parameters for hydraulic press system

Parameter	Value	Parameter	Value	Parameter	Value
β	$1.4 \cdot 10^{7}$ N/m^2	V_0	$7.5398 \cdot 10^{-4}$ m^3	b_a	520 N s/m
A_1	$50.26 \cdot 10^{-4}$ m^2	m	50 kg	f_c	120 N
A_2	$28.27 \cdot 10^{-4}$ m^2	k_e	$5.46 \cdot 10^{-5}$ N/m	K_p	6
φ	1.7778	K	$8.33 \cdot 10^{-6}$ m^3/V s	K_i	2

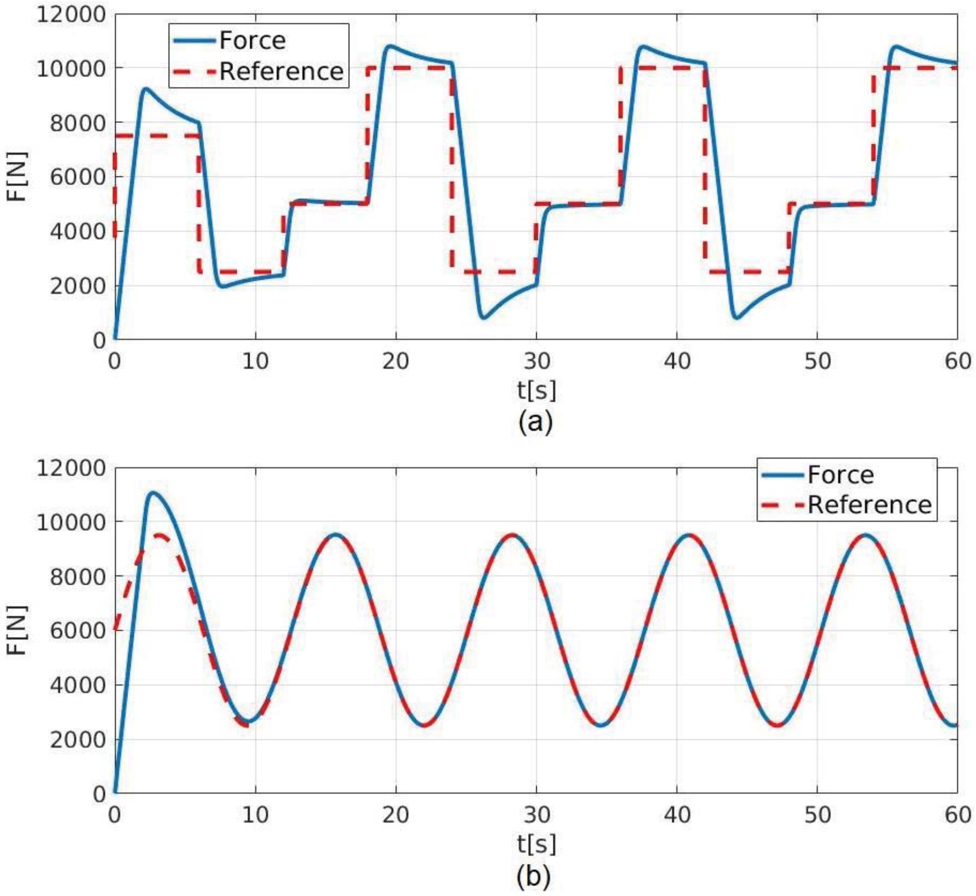


Figure 18: Hydraulic press simulation results, (a) square wave force tracking, (b) sine wave force tracking

5 Experimental results

The control algorithms are implemented in Matlab/Simulink environment, which allows executing the simulation program in real time. Such implementation of control algorithms allows real time data monitoring and online tuning of controller parameters. Block diagrams are used for the controller design procedure which gives simple data flow representation. The experimental results obtained for both systems are compared with those obtained from simulations

5.1 Pneumatic press experimental results

Implementation of a PID controller for the pneumatic press system is given in Figure 20. By activating the switch in controller model it is possible to choose between sine and square wave input signal. The controller is implemented on a standard laptop PC with a sampling rate of 20Hz. During experiments, signals that were recorded are the pressing force, the error between the reference force and the pressing force, the control signal on the proportional pressure-control valve and the pressure from the pneumatic actuator inlet.

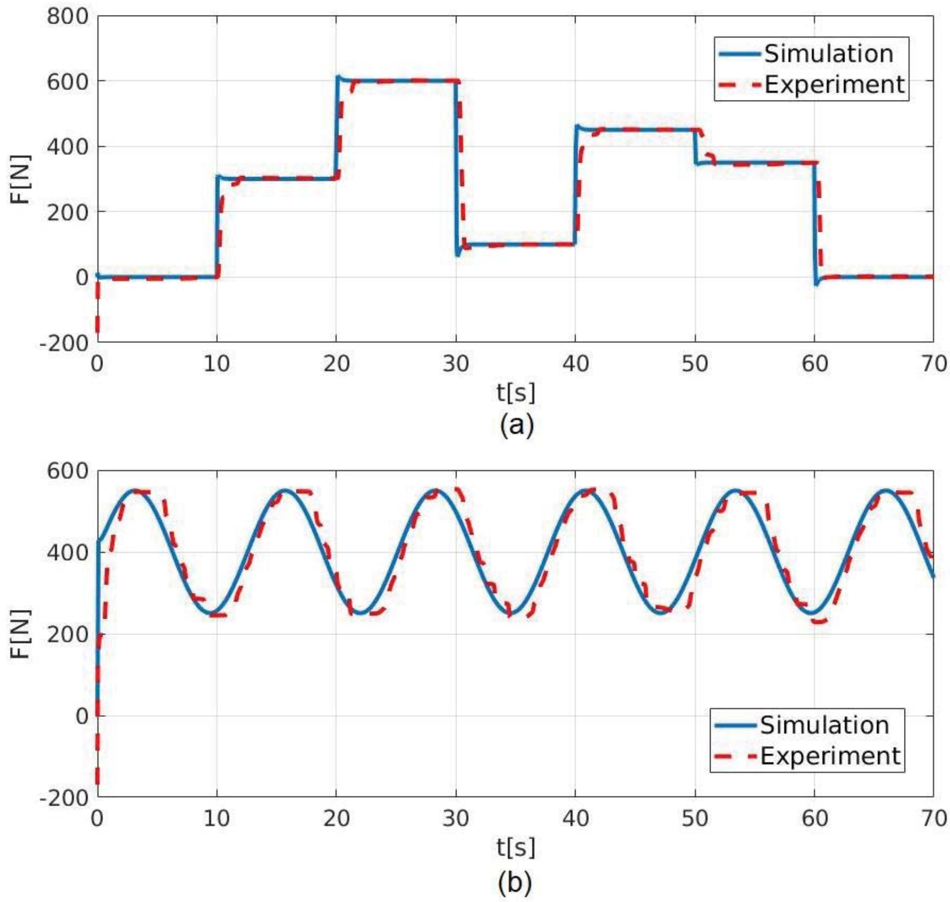


Figure 19: Pneumatic press experimental results, (a) square wave force tracking, (b) sine wave force tracking

The first experimental test was carried out for the square wave input signal and the results are shown in Figure 19. The measured results exhibit an aperiodic response, without overshoot and with small output error. The second experiment was carried out for the sinusoidal signal of frequency 0.08 Hz. The force response is shown in Figure 19. The experiment demonstrates that the PID controller follows the reference sinusoidal input signal with significant control error. Both experiments were carried out with PID gains obtained from numerical simulations.

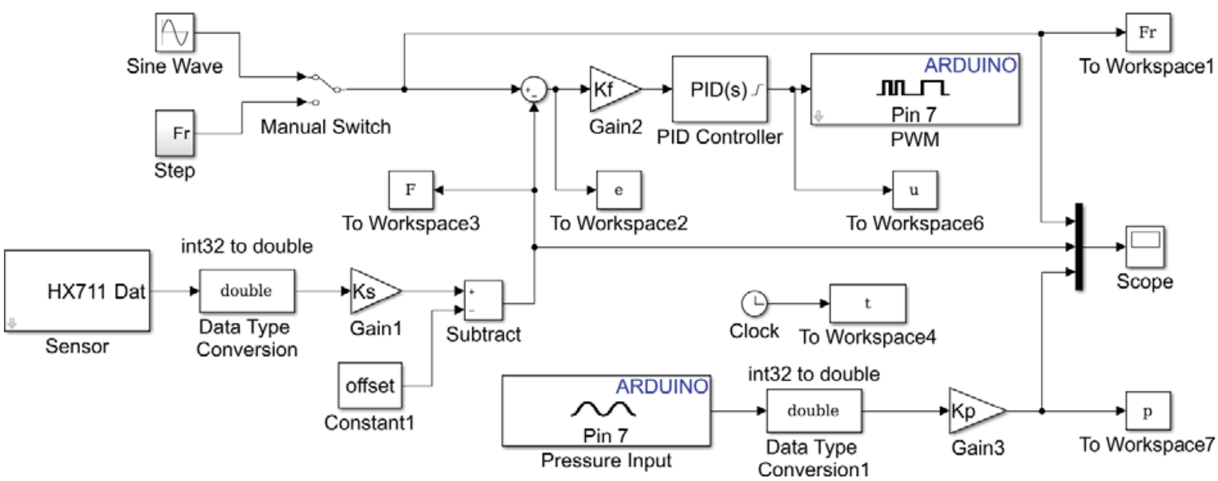


Figure 20: Simulink model used for experimental testing of the pneumatic press

5.2 Hydraulic press experimental results

The model using PI controller for the hydraulic press force control is shown in Figure 21. The switch shown on the Simulink model is used for choosing between square and sine wave input. The controller is implemented on a standard laptop PC with sampling rate of 100Hz. Signals that are being acquired are the pressing force, the control signal for the servo valve, and the pressure from the hydraulic cylinder inlet.

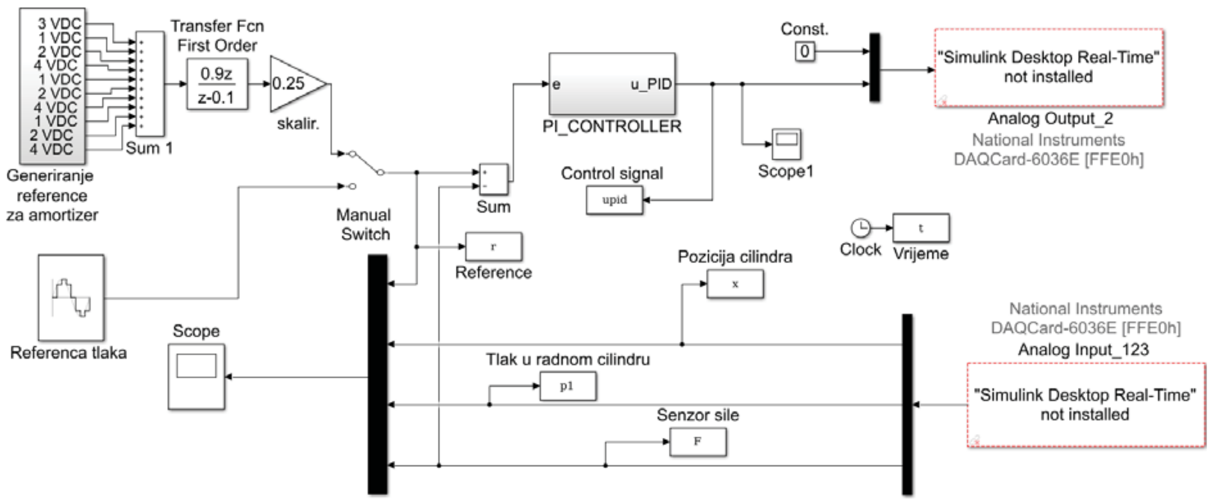
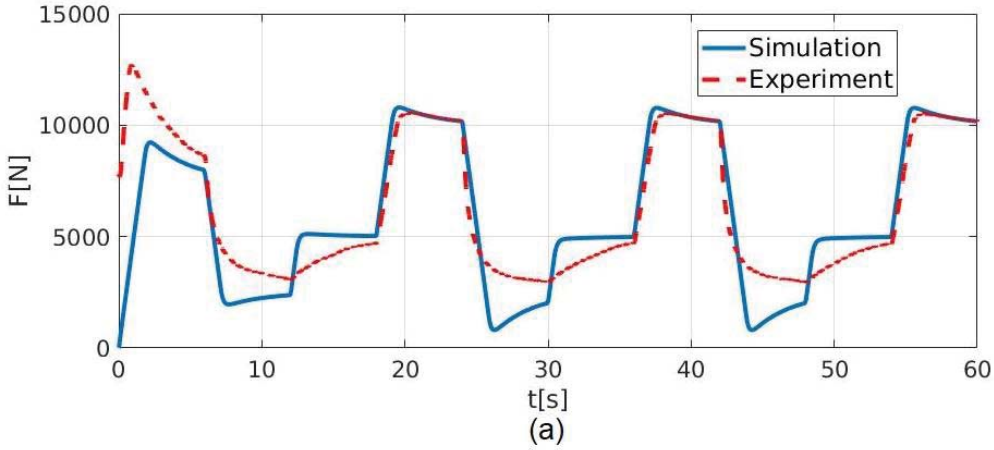


Figure 21: Simulink model used for experimental testing of the hydraulic press

Experimental results for the hydraulic press are shown in Figure 22. PI regulator gains, used for experimental test, are the same gains used in numerical simulation test. The value for proportional (K_p) gain was set at 6, and value for integral (K_i) gain was set at 2. The first test was carried out for the square wave reference signal. Results showed similar trend as the results from numerical simulation. Therefore, the linear PI regulator follows the reference square wave input signal with significant control error because of many nonlinearities of the hydraulic systems. The second test was performed with the sinusoidal wave signal with the frequency of 0.08 Hz. From observed experimental results it can be concluded that PI controller shows inferior results for the sinusoidal wave than for the square wave input signal, which was expected.



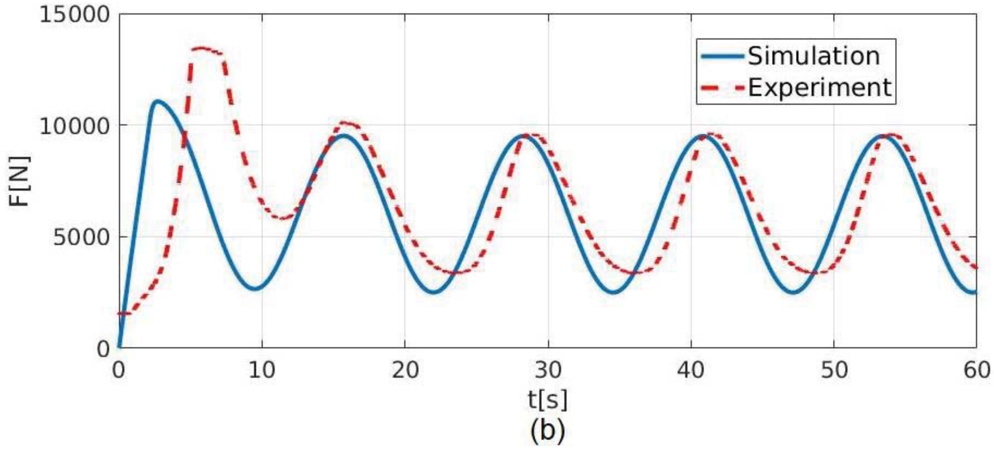


Figure 22: Hydraulic press experimental results, (a) square wave force tracking, (b) sine wave force tracking

6 Conclusion

According to the experimental results, it can be concluded that in the case of PID controlled force of the pneumatic press, valid results can be obtained only for square wave input signal. Moreover, numerical simulations didn't present the same results as the experimental tests, because of many simplifications in the mathematical model. For the hydraulic press system PI controller could only be used for tracking desired constant input force. For the square and the sinusoidal wave input signal it is needed to consider new ways of the system control. The experimental results show that there are no high frequency oscillations in the control signal, unlike the oscillations which have appeared in numerical simulations. For better systems control, new control algorithms should be considered, such as sliding mode control, back stepping control or fuzzy control. Mentioned control algorithms could probably give better results than PID controller.

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