CONTROL OF A PNEUMATIC SYSTEM FOR MATERIAL STRENGTH TESTING

Željko Šitum[1]*, Marko Miroslav Bača[2], Juraj Benić[1], Željko Alar[2]

[1]*Department of Robotics and Production System Automation, University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, Ivana Lučića 5, HR-10000 Zagreb, Croatia*
[2]*Department of Materials, University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, Ivana Lučića 5, HR-10000 Zagreb, Croatia*

* Corresponding author: Tel.: +385 91 733-4072; E-mail address: zsitum@fsb.hr

---

ABSTRACT

The article describes a prototype device for testing the dynamic stress of materials with the intention of analysing the fatigue behaviour of the materials. A pneumatic cylinder controlled by a proportional pressure regulator is used to achieve the required force. The experimental set-up consists of structural elements with pneumatic components, then a control system with a Controllino device and HMI interface, and the measuring system composed of a dynamometer with an amplifier characterized by high accuracy and sampling frequency characteristics. With this system, it is possible to work in both compression and tension ranges. The basic parameters of the process were identified, then the control-oriented dynamic model of the system was derived, and the synthesis of the PID controller was made for more precise following of the sinusoidal reference signal. Compared to hydraulic solutions that are used to achieve large forces, this pneumatic fatigue testing machine has a compact design, the desired reference is simply set via the HMI interface, it is portable, and it can be used to test materials with lower tensile strength.

Keywords: Pneumatic fatigue testing machine, Material strength testing, Dynamic endurance

---

1. INTRODUCTION

The common intersection of the main areas of production engineering such as technologies, materials science and construction overlap with the analysis of stress and deformation of materials, i.e. on the strength of materials. Material strength is a fundamental feature in the analysis and design of structural parts, as it determines the permitted capacity and load of a complex system (with the addition of a safety factor). If the permissible stress limit is exceeded, the most common approach is correction, which in practice means increasing the dimensions or changing the material. In materials science, tests of mechanical properties of different materials under conditions of long-term dynamic action of variable stress have special significance because the results of these tests determine the dimensional calculation of mechanical structures [1]. Understanding the fatigue mechanism is important for considering various technical conditions and this knowledge is essential for the analysis of strength properties of an engineering structure [2]. Experimental testing of components under static and dynamic loading tests is necessary for ensuring the reliable operation of structural elements. As a result of long-term periodic variable loading, material fatigue occurs. This causes fracture of the material at a significantly lower stress than the maximum tensile strength of the material. In mechanical laboratories, two types of testing machines are most often used to perform fatigue tests with random loading. These are servo-hydraulic devices [3], which generate force and/or torque as the input excitation and electrodynamic shaker tables [4], which apply a force to the vibrating table. For the needs of biaxial in-plane fatigue tests, the servo-hydraulic actuator type is the most common option available in the market [5]. The analysis of mechanical properties of different materials under conditions of long-term dynamic action is often a crucial criterion when designing machine parts, and therefore there are many manufacturers of devices for material strength testing on the market. Many reputed companies around the globe manufacture fatigue testing machines, so these machines are available in different versions and in different price ranges. Three types of fatigue testing machines are the most common types of this machine: strain control, low cycle fatigue, and load control high cycle fatigue. The purpose of fatigue testing is to determine the lifespan that can be expected from a material subjected to cyclic loading. However, fatigue strength and crack resistance are also desirable information about material behaviour. The fatigue life of a material is the total number of load cycles to which the material can be subjected. Fatigue testing is also used to determine the maximum load that a specimen can withstand for a specified number of cycles. These features are extremely important for the use of various materials subject to dynamic loads. For the realization of smaller forces, the dynamic endurance of the material could be analysed on test systems (fatigue testing machines) that use pneumatic actuators. Current intensive technological development requires an increasing number of production procedures with a simultaneous reduction of the material used, so ensuring the reliable operation of structural elements requires experimental testing of components under static and dynamic straining. Modern materials testing devices require the possibility of programmed motion and/or actuator force control, as well as quick adaptation of control algorithms to new requirements in the material testing process. When designing complex systems, it is often quite difficult to predict all the sections where the critical stress occurs. Simulation models can give a good insight into the behaviour of the system during exploitation, but the need for a real experiment is necessary to validate the results. Since it is theoretically impossible to determine the exact value of the dynamic endurance of the material, in practical applications experimental tests are carried out on specialized fatigue testing machines. Such tests provide the necessary parameters for construction calculations to produce components of smaller dimensions that meet the requirements for a longer period of operation. Fatigue testing machines are most often based on a servo-hydraulic drive, they can produce very large forces, but the dimensions of such devices are very large. Nowadays, when many products are made from polymer or metal sheets, there is a need for test devices that can realize smaller forces, which makes the application of pneumatic test devices suitable for these tasks.

The aim of this work was to produce a universal pneumatically driven test device on which static and dynamic loading tests can be carried out. For static tests, the accuracy of the system is very important, and the response speed is less important, while for dynamic tests, the response speed is essential for achieving higher operating frequencies, and the accuracy is a secondary requirement [6]. The use of pneumatics makes it possible to achieve the necessary forces for testing materials of lower strength, the system will be of smaller dimensions and suitable for desktop and mobile use in laboratories for testing the dynamic properties of materials.

The experimental set-up must meet the necessary norms and recommendations for work in test laboratories and must be able to carry out a complete dynamic test of the material.

2. EXPERIMENTAL TEST DEVICE

The constructed experimental system, shown in **Figure 1**, consists of a pneumatic system, a control system with HMI interface, and a measuring device with an amplifier. The construction of the test device is made from standard aluminium profiles due to the high modularity and wide choice of shapes that allow the connection of various profiles using angle joints. The upper horizontal beam is used as a support for the pneumatic cylinder. It is movable along the vertical axis and ensures the required position of the pneumatic actuator during the testing procedure [7].

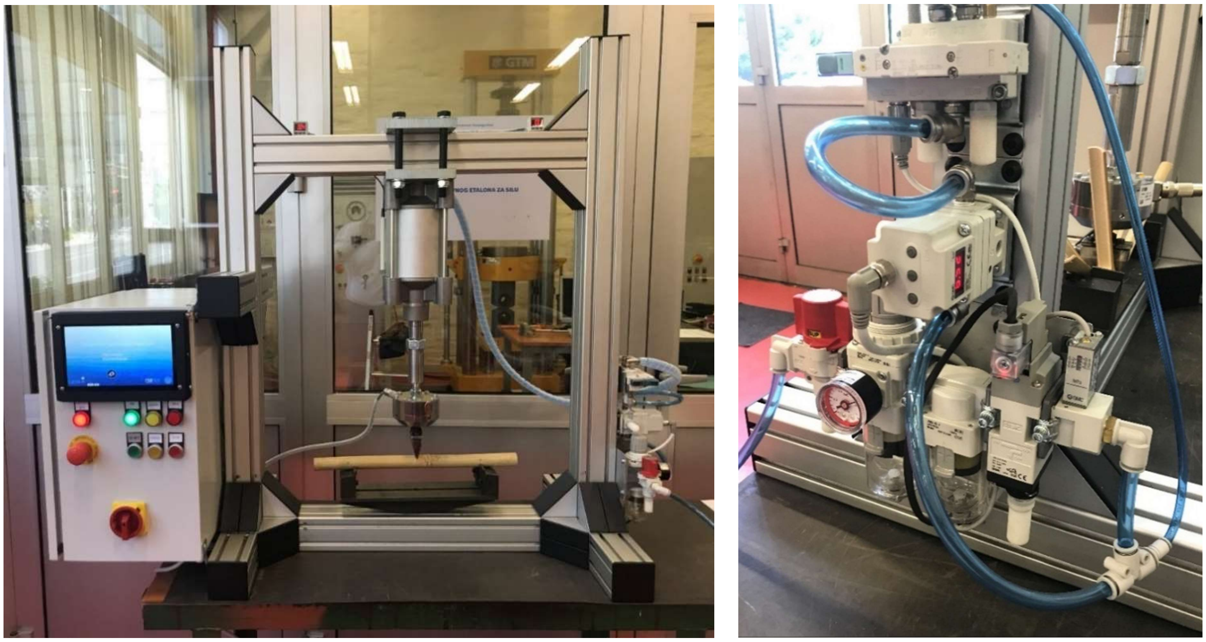

Figure 1: Fatigue testing machine

The pneumatic system of the test device is physically detached from the electrical part, which enables simple corrective maintenance with easy access to all components. The air preparation unit is made as a compact component and consists of a manual 3/2 valve, a micro air filter, an electromagnetic 3/2 valve and a pressure switch. **Figure 2** shows the pneumatic scheme of the experimental system.

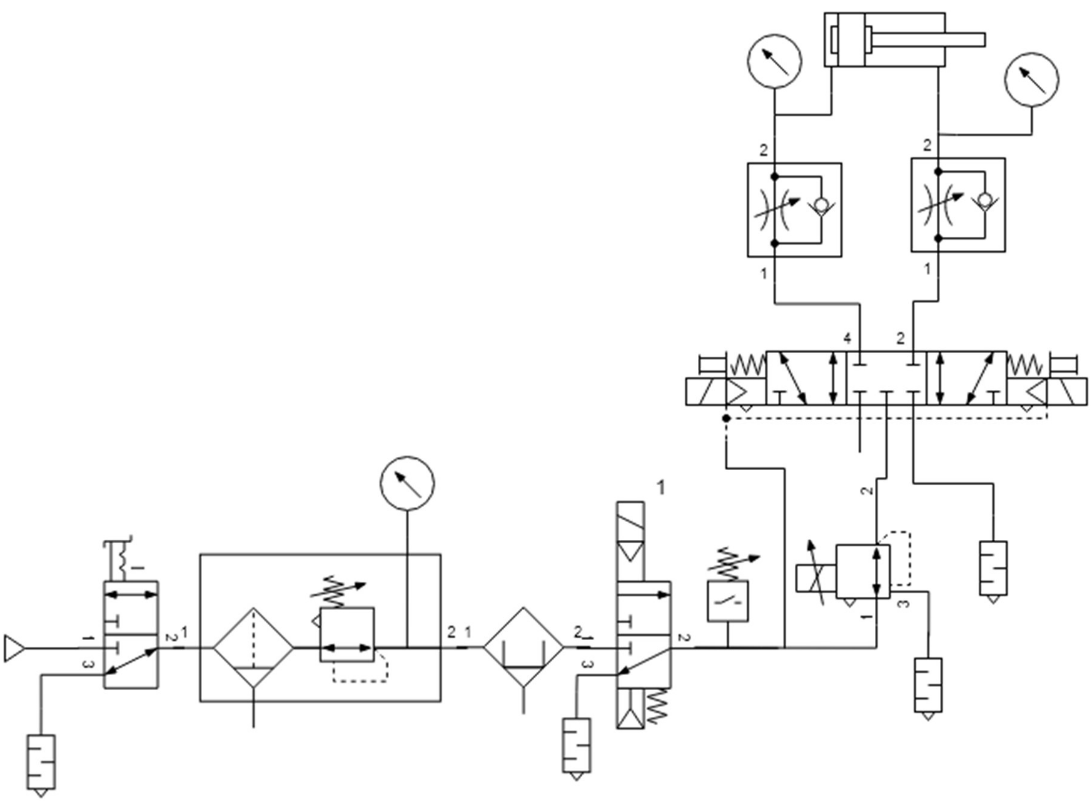

Figure 2: Pneumatic scheme of the experimental system

Dynamic loading of test samples is achieved by applying forces of different frequencies and amplitudes. The diameter of the piston and the stroke of the pneumatic cylinder are selected according to the force requirements, considering standard industrial pressures. To prevent *stick-slip* effect, a smooth, slow-moving cylinder of the new generation, type SMC CY-96YDB125-100 was chosen, which has lower friction values at lower pressures than standard cylinders. A proportional pressure regulator SMC ITV2050-312N3 was used to control the force of the pneumatic cylinder. An external pilot solenoid 5/3 valve SMC SY5301RK-5U1 with a metal seal was used to supply compressed air to both cylinder chambers. Two flow control valves are used only in the preparatory phase, to strongly reduce the flow and decrease the speed of the actuator movement to achieve a proper effect of the actuator on the test sample in the clamping device. During the test phase, they are fully open. The maximum dimension of the test sample is 50 cm x 100 cm (w x h), and the length of the sample can be adjusted by using supports (e.g. ski testing). The maximum displacement of the cylinder is 100 mm. The amount of force depends on the pressure of the compressor, which is set to 10 bar, and multiplied by the surface of the cylinder gives a maximum force of 11 kN.

Controllino Maxi Power Automation PLC with software that is fully compatible with Arduino is used as a control device, **Figure 3**. The basis of the Controllino device is the ATmega328P microcontroller, which is an integrated chip that controls devices and processes, stores and executes the program. All input-output connectors have electrostatic discharge protection, so it is a control device that combines the advantages of open source and programming methods with the standards and safety of industrial PLCs. At the beginning of the program, Controllino initializes all necessary variables, then certain pins are assigned the property of input or output pins. The direction of movement of the cylinder is controlled by sending an electrical signal to the electromagnetic valve, and the pressure in the cylinder is determined by the voltage signal on the proportional pressure valve. The device for communication between the system and the operator is an LCD screen with a diagonal of 9", which will be suitable for entering the necessary parameters and displaying graphs of measurement data from the sensor. The working environment of the designed system are normal laboratory conditions, so industrial quality devices are not required. **Figure 4** shows the HMI system SK-90DT manufactured by 4D Systems, which enables simple communication and data exchange with the control unit. The screen is mounted on the electrical cabinet.

Figure 3: Control device

Figure 4: HMI interface of the test device

The force measuring sensor is a dynamometer axially connected to the cylinder piston. The dynamometer has a measuring amplifier which enables reading the applied force and sending a signal to the PLC. The dynamometer, HBM type U10M and ClipX amplifier, is characterized by high accuracy and sampling frequency and enables operation in the compressive and tensile range. Calibration was done on the built-in dynamometer and accuracy in class 1 was achieved. On 100,000 performed cycles, about 2% were below the required tolerance. **Figure 5 s**hows the setup for testing the flexural strength specimen made of bamboo, where the dynamometer for force measurement is attached to the piston rod of the pneumatic cylinder. For the procedure of testing mechanical properties of bamboo sample, a device for holding the sample was also made.

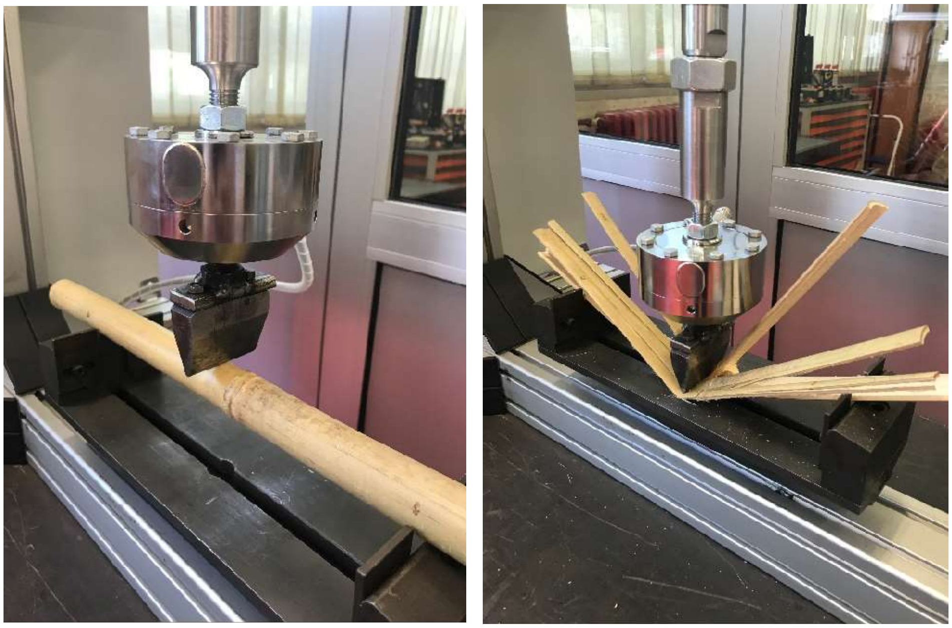

Figure 5: Testing the flexural strength

3. DYNAMIC MODEL OF THE TEST SYSTEM

The dynamic model of the test system was obtained by determining the transfer function of the proportional pressure valve and the dynamic model of the pneumatic cylinder, which includes the friction of the cylinder and the stiffness of the test object.

3.1. Dynamic model of the proportional pressure valve

The proportional pressure valve sets the output pressure proportional to the input voltage, indicating a linear characteristic as shown in **Figure 6**, with a linearity of ±1% of full scale guaranteed by the valve manufacturer [8].

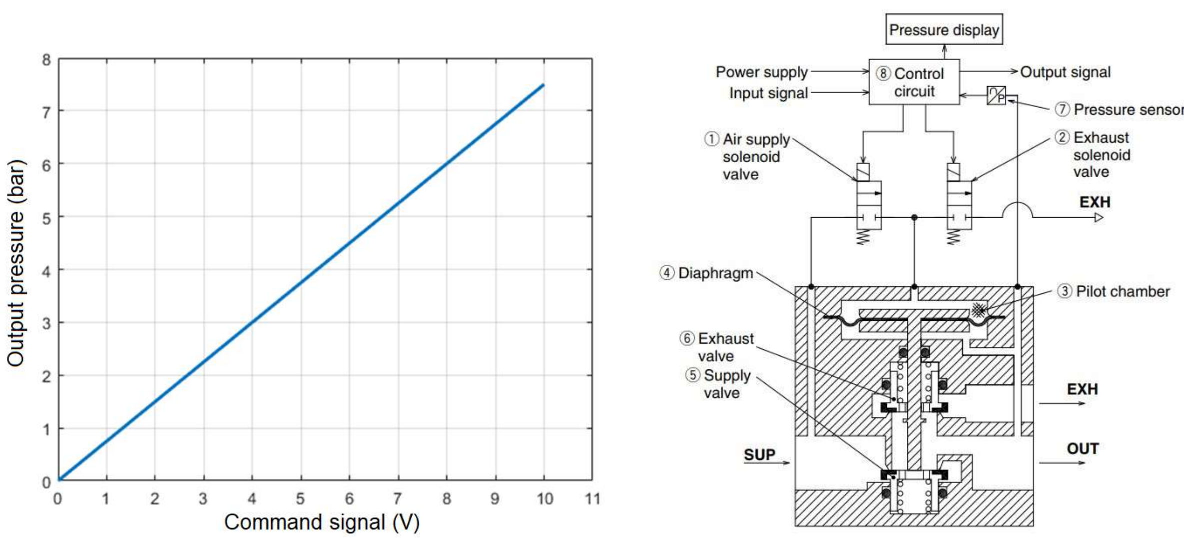

Figure 6: Static characteristic of the proportional pressure valve and working principle diagram

The forward gain K_v of the proportional pressure valve is determined from the calibration curve of the valve, which presents the relative pressure outlet versus the input voltage signal and is calculated as K_v=0.075 MPa/V. The value of the time constant T_v of the proportional pressure valve was determined experimentally based on the transient response of the output pressure to the step reference signal. The pressure is measured using a built-in sensor in the housing of the proportional valve. Three different input voltage signals of 3, 6 and 9 V are set on the pressure regulating valve. According to the static characteristic, the stationary value will be different, so the responses are scaled. **Figure 7** shows the pressure responses in the system for different voltage signals supplied to the proportional pressure valve, which are scaled to the forward gain K_v.

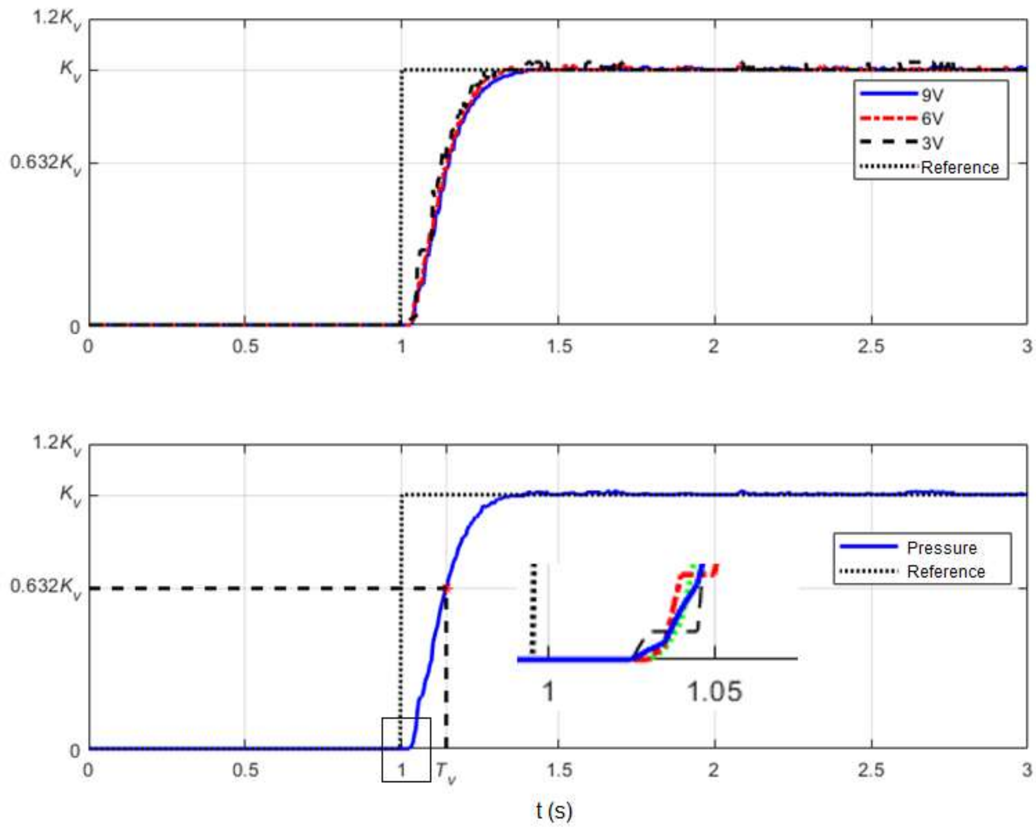

Figure 7: Pressure transient response

The second diagram in Figure 7 shows the mean value of the transient response curves for the previous three levels of voltage signals. From the transient response, the value of the time constant of the proportional pressure valve was calculated as the time required to reach the amount of 63.2% of the stationary value of the response, which gives the time constant T_v= 0.1422 s. The final expression of the PT1 term can be represented by the following transfer function:

$$\mathrm{G}(s) = \frac{p(s)}{u(s)} = \frac{K_\mathrm{v}}{T_\mathrm{v}\,\mathrm{s} + 1} = \frac{0{,}075 \cdot 10^6}{0{,}1422\; s + 1} \tag{1}$$

For a more accurate valve model, a transport delay (dead time) can be added to the proportional term in the simulation program, which according to the zoomed part in Figure 7 has a value of T_u=0.03 s. However, the transfer function with the added dead time makes the synthesis of the controller for the implementation of the experiment more complex.

3.2. Dynamic model of the pneumatic cylinder

The dynamic model of the pneumatic cylinder is obtained using the expression for the balance of forces of the mechanical system, assuming that air is an ideal gas.

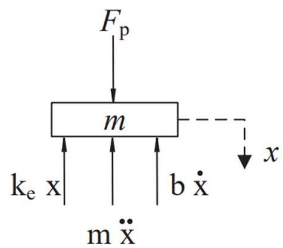

Figure 8: Forces on the cylinder piston

From [9], the reaction force of the test specimen is represented as a spring force. Given that dynamic testing of materials is primarily conducted within the elastic region of the material, such a representation of the reaction force is valid. Consequently, the first and second derivatives of the spring force are as follows:

$$x = \frac{F_c}{k_e} \quad , \quad \dot{x} = \frac{\dot{F}_c}{k_e} \quad , \quad \ddot{x} = \frac{\ddot{F}_c}{k_e} \tag{2}$$

The balance of forces on the cylinder piston can be given by the following expression:

$$m\,\ddot{x} = F_p - b\,\dot{x} - k_e\,x \tag{3}$$

where the applied force F_p is defined as the pressure differential across the piston multiplied by the cross-sectional area of the piston (A), which includes both the piston and connecting rod, m represents the piston mass, k_e is the stiffness constant, b is the viscous friction coefficient, and x is the position of the cylinder. The choice of the Newtonian friction model is based on the use of a smooth cylinder in the experimental setup, which exhibits low friction at lower speeds of cylinder movement and practically eliminates stick-slip motion.

The dynamical model of the hydraulic cylinder is obtained by inserting equations (2) into equation (3) as follows:

$$\ddot{F}_c = \frac{k_e}{m}\,(A_1\,p_1 - A_2\,p_2) - \frac{b}{m}\,\dot{F}_c - \frac{k_e}{m}\,F_c \tag{4}$$

The analysed system will work exclusively in the compression range, so the pressure in the second chamber will always be approximately atmospheric pressure.

$$p_2 = p_a \tag{5}$$

The final expression for the dynamics of the pneumatic cylinder where the force is the output value is as follows:

$$\ddot{F}_c = \frac{k_e}{m}\,A_1\,p_1 - \frac{k_e}{m}\,A_2\,p_a - \frac{b}{m}\,\dot{F}_c - \frac{k_e}{m}\,F_c \tag{6}$$

The control error e is expressed as the difference between the reference and actual force:

$$e = F_R - F_c \tag{7}$$

and this value will be the input to the controller which defines the output pressure on the proportional valve. A dynamometer is additionally connected to the piston rod of the cylinder, as well as other equipment for achieving contact with the test specimen and for load measurement. Equation (6) contains parameters whose values need to be determined, such as the mass m of the cylinder piston and the attached measuring equipment, the coefficient of viscous friction b and the stiffness constant k_e. The mass of the cylinder piston and attached measuring equipment was determined using an external dynamometer and amplifier connected to the piston rod of the cylinder. The average value of three force measurements is converted into mass to be consistent with the unit of measurement in equation (6), and thus we obtain a precise amount of mass m=6.4269 kg.

Figure 9: Estimation of process parameters using a rubber bumper

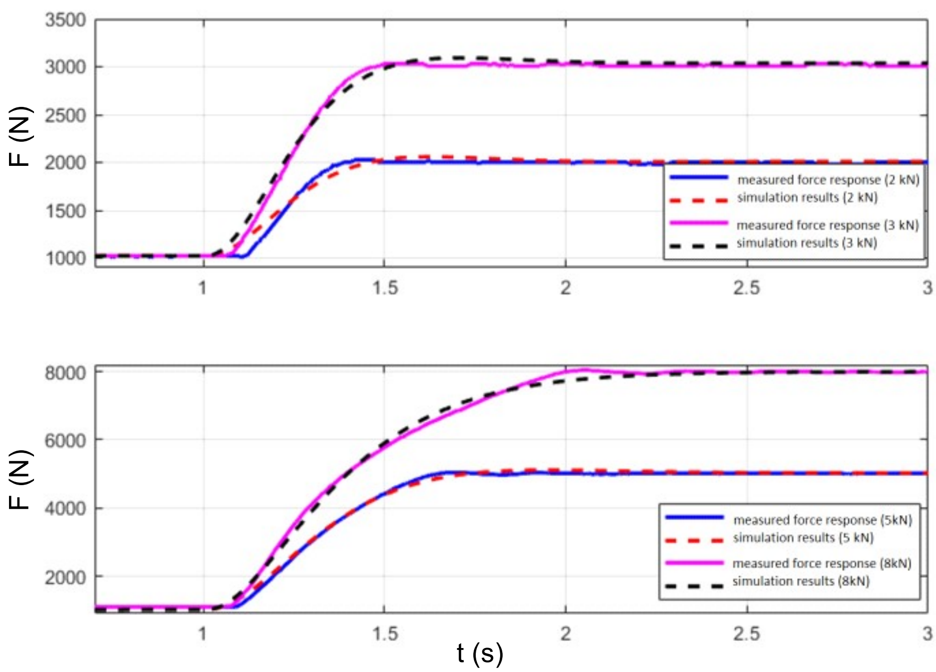

Figure 10: Comparison of measured force response and simulation model with estimated parameters

The values of the parameters b and k_e are difficult to determine directly as in the case of mass, especially in the case of the spring stiffness constant k_e which approximates the reaction force of the test specimen, since the specimens are always different. By measuring the experimental response of the system, it is possible to obtain the actual behaviour of individual system components, so this approach was used to determine unknown process parameters. The parameters b and k_e were determined from the experimental response of the force applied to the rubber bumper and the estimation of the parameters of the simulation model using the normalized root mean square error - NRMSE method. This dimensionless method is used to measure the difference between prediction models and recorded data and is often used in statistical data analysis. A rubber bumper, shown in **Figure 9**, was used to simulate the test specimen, which can achieve larger deformations for the applied force. In the initial position, the cylinder piston touches the surface of the rubber bumper. The solenoid valve is open during the entire experiment, so its dynamics does not affect the pressure building process. The pressure increase in the cylinder is controlled by a proportional pressure valve, and its dynamics will be considered. A comparison of the simulation and experimental results for several cylinder pressure (force) values is shown in **Figure 10**, while the process parameters used in the simulation are given in **Table 1**.

Table 1: Parameters of the dynamic system model

Cylinder parameter	Value	Unit
Mass m	6.4269	kg
Piston area A_1	12.272×10^3	mm^2
Rod end area A_2	11.468×10^3	mm^2
Friction coefficient b	74	Ns/mm
Stiffness constant k_e	266	N/mm

By estimating the unknown parameters from the measured responses of the process and comparing them with the simulation results, a good matching between the experimental and simulation results can be observed for several pressure values (and thus the cylinder force).

4. EXPERIMENTAL SYSTEM CONTROL

For system control, the parallel structure PID controller is often applied, which has the derivation of the error signal as an input to the D-term. Although this approach is often used in industry, it should be avoided due to the appearance of large forcing of the control signal during step reference changes. Also, during each measurement of the controlled value, the noise of the measured signal appears, and the derivation significantly increases the noise level, so such a signal is almost unusable. To eliminate this problem, a PID controller with a speed estimation filter was used, which is shown in **Figure 11**.

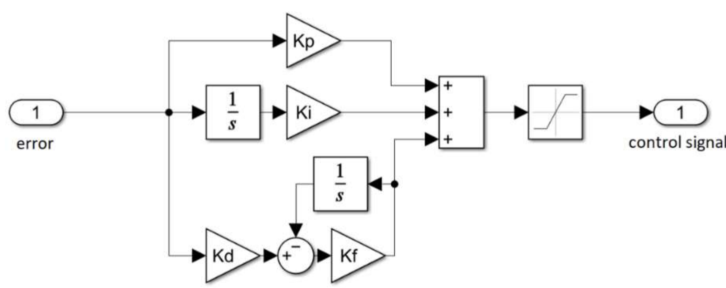

Figure 11: PID controller with speed estimation filter

The parameters of the PID controller used in the simulation and experiment were: K_p=0.0337, K_i=0.0961, K_d=0.0038, while the speed estimation filter was set to the value K_f=250. The initial parameters of the controller were determined from the experimentally recorded transient response of the process in the open loop, and then minor adjustments were made during the control process. The designed controller was checked on the experimental system and the response comparison is shown in **Figure 12**. It can be observed that there is quite a good matching between the simulation results and the experimental response in the control process for lower frequencies of the sinusoidal reference signal and a certain phase delay for the case of higher frequencies.

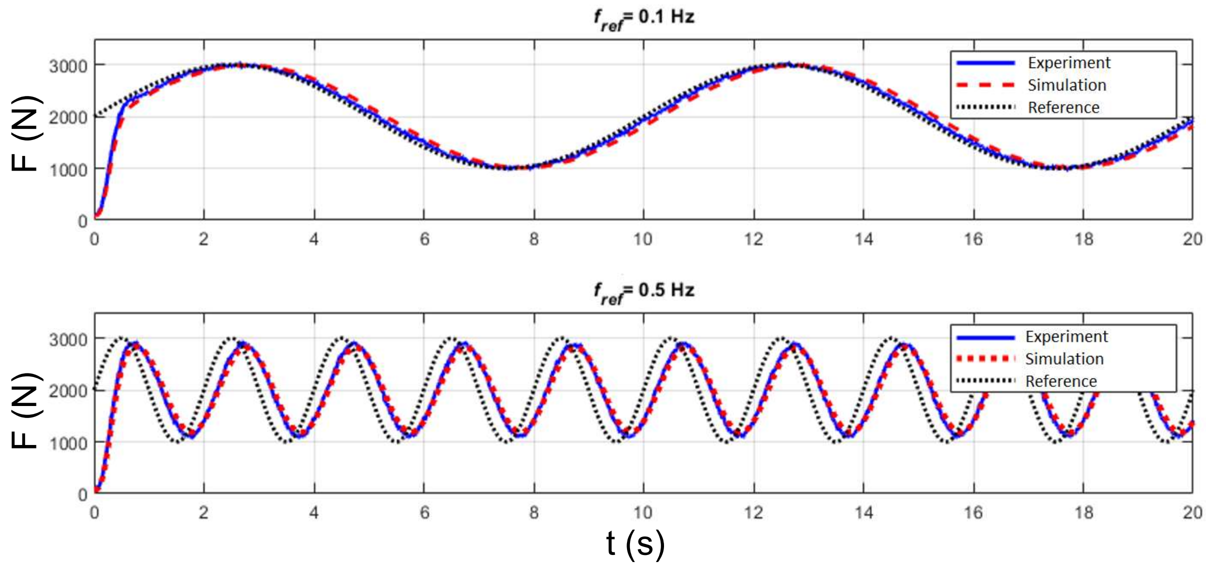

Figure 12: Comparison of experimental and simulation results using PID controller

During the testing, it is important to keep the amplitude deviation below the required value (e.g. below 10%), so the regulator is set to achieve the force output value slightly below the reference value. If the achieved force is significantly lower than the required value, in the control program the number of test cycles will be increased by the number of such inappropriate amplitudes. During dynamic testing of materials, the phase delay is less relevant and does not significantly affect the implementation of the experiment. Further improvements towards more accurate tracking of higher frequency reference signals are expectedly possible using feedforward control methods.

5. CONCLUSION AND OUTLOOK

Practical realization of own laboratory educational test device, which can be used for experimental demonstration of the fracture mechanics of materials caused by dynamic stress, analysis of the resistance of materials to cyclic stress and the development of cracks due to material fatigue is a valuable achievement in deepening knowledge of materials. An important feature of this test device is the use of pneumatic components to achieve the required dynamic forces and a user interface with a touch screen to set the parameters of the material testing process, while reference and measured force values are displayed on the screen in real time. A simplified control-oriented dynamic model of a pneumatic cylinder controlled by a proportional pressure valve has been derived. The process parameters were analysed and experimentally identified. Based on the dynamic model of the process, the parameters of the PID controller were determined and the experimental results for the task of tracking the sinusoidal reference force were obtained. Satisfactory results of tracking the reference force were obtained for sinusoidal signals of lower frequencies, but due to the slow dynamics of the control components when tracking sinusoidal references of higher frequencies, there were significant delays in tracking the reference force. However, for dynamic testing of materials, the realization of a pulsating response is required, and the accuracy of tracking the periodic signal is a secondary criterion. In further research, some more advanced control methods will be applied to explore the possibilities of improving the dynamic response and accuracy of the system. To avoid readjustment of the controller for each test specimen (depending on the stiffness of the test specimen), a state controller with parameter estimation will be applied. The developed system is completely open for upgrading new user applications, which is quite difficult with available commercial systems. Also, the price of used components of approximately 5,000 euros is many times lower than similar available systems offered on the market, especially those with hydraulic actuators.

NOMENCLATURE

A_1 , A_2	Cylinder piston area and rod end area	m^2
e	Force control error	N
F_c	Spring force of the test specimen	N
F_p	Applied force on the cylinder piston	N
F_R	Reference force	N
$G(s)$	Transfer function	-
K_f	Speed estimation filter	-
K_p , K_i , K_d	PID controller gains	-
K_v	Forward gain of the proportional valve	Pa/V
k_e	Stiffness constant	N/m
b	Viscous friction coefficient	Ns/m
m	Mass of the piston and measuring equipment	kg
p_1 , p_2	Pressures in chamber 1 and chamber 2	Pa
p_a	Atmospheric pressure	Pa
s	Laplace operator	-
T_v	Time constant	s
u	Control signal	V
x	Cylinder position	m

REFERENCES

[1] Gela T (2006) Mechanical Design Handbook, Chapter 6: Properties of Engineering Materials. McGraw-Hill Book Company

[2] Murugan S (2020) Mechanical Properties of Materials: Definition, Testing and Application. Int. J. of Modern Studies in Mechanical Engineering (IJMSME) 6, 2, 28-38

[3] Costa P, Nwawe R, Soares H, Reis L, Freitas M, Chen Y, Montalvão D (2020) Review of Multiaxial Testing for Very High Cycle Fatigue: From 'Conventional' To Ultrasonic Machines. Machines, 8, 25; doi:10.3390/machines8020025

[4] Ghielmetti C, Ghelichi R, Guagliano M, Ripamonti F, Vezzù S (2011) Development of a fatigue test machine for high frequency applications. Procedia Engineering 10, 2892–2897. https://doi.org/10.1016/j.proeng.2011.04.480

[5] McAlorum J, Rubert T, Fusiek G, Niewczas P, Zorzi G (2018) Design and Demonstration of a Low-Cost Small-Scale Fatigue Testing Machine for Multi-Purpose Testing of Materials, Sensors and Structures. Machines, 6(3), 30, https://doi.org/10.3390/machines6030030

[6] Marques J M E, Benasciutti D, Niesłony A, Slavič J (2021) An Overview of Fatigue Testing Systems for Metals under Uniaxial and Multiaxial Random Loadings. Metals, 11, 447. https://doi.org/10.3390/met11030447

[7] Šitum Ž, Benić J, Pejić K, Bača M M, Radić I, Semren D (2021) Design and control of mechatronic systems with pneumatic and hydraulic drive, Int Conf Fluid Power 2021. Maribor, September 16-17, 2021, Maribor, Slovenia, 179-193.

[8] Technical documentation Electro-Pneumatic Regulator/Electronic Vacuum Regulator ITV Series, SMC ITV 2050-312N3, https://static.smc.eu/pdf/ITV-G_EU.pdf

[9] Benić J, Rajčić N, Šitum Ž (2017) Precise force control for hydraulic and pneumatic press system, Int Conf Fluid Power 2017. Maribor, September 14-15, 2017, Maribor, Slovenia, 57-71.