Sixth World Conference on Structural Control and Monitoring

Proceedings of the 6th edition of the World Conference of the International Association for Structural Control and Monitoring (IACSM), held in Barcelona, Spain

15-17 July 2014

Edited by:

J. Rodellar
Universitat Politècnica de Catalunya-BarcelonaTech, Spain

A. Güemes
Universidad Politécnica de Madrid, Spain

F. Pozo
Universitat Politècnica de Catalunya-BarcelonaTech, Spain

A publication of:

International Center for Numerical Methods in Engineering (CIMNE)
Barcelona, Spain

Analysis and thermodynamic modeling of a pneumatic adaptive absorber*

Grzegorz Mikułowski[1], Rafał Wiszowaty[2] and Jan Holnicki-Szulc[3]

Abstract

The objective of this work was to develop a mathematical model of coupled thermodynamic and mechanical processes proceeding in pneumatic, adaptive absorbers under cyclic loadings. The results of the modelling were to be verified versus experimentally obtained data. The analysis was divided into sections devoted to: forces acting on the piston, thermodynamics of the gas in the absorber's chambers, gas flow through the piezoelectric valve. Three control volumes were distinguished within the absorber's structure in order to analyze the thermodynamic processes. For each control volume analysis of energy balance, thermodynamic state parameters and heat transfer were performed. A set of equations was formulated for each control volume in order to determine: (1) motion of the piston in relation to the acting forces, (2) the gas state evolution, (3) energy balance within each control volume and (4) heat transfer to the surroundings. The obtained results revealed that the proposed approach to modeling was in good agreement with the data obtained experimentally. The controllability of the absorber was successfully reflected by means of the numerical model outcome.

I. INTRODUCTION

The proper modes of dissipation of the redundant mechanical energy are desired in many branches of transport and industry. Passive dampers, which are the most popular, are usually tuned to a fixed range of operating conditions and therefore can not perform optimally in the cases of applications with varying in time mechanical excitations [1], [2], [3], [4]. For such situations semi-active devices are dedicated. From commercial applications there are known dampers based on magnetorheological fluids (MRF), which have the capability of rapid changing their reaction force via a controlled magnetic field influencing an effective viscosity of the MRF [5]. The applied devices are dedicated to rotation problems [6], [7] or linear cases and are sized from small dampers that generate reacting force below 2 kN [8] to large scale dissipaters dedicated to seismic protection of buildings [9]. The disadvantage of this technology is relatively high weight of the devices due to high density of the MRF. Another technique for semi-active devices with adjustable reaction force are oleo-pneumatic absorbers with a control valve positioned within the oil circuit [10], [11]. By adjusting the throttling of the fluid flow, these devices provide an adjustable reaction force. Both of the mentioned techniques are controllable in a predefined range of forces but also reveal a viscous response, which reduces their controllability with the velocity increase. That might be an unfavourable effect in some applications. In the conceptual device described in this paper, gas medium was utilized and therefore the viscous effects were limited.

The pneumatic systems for isolation and vibration mitigation were developed for suspension of precise measuring instrumentation [12] as well as for large structures: seismic protection of buildings or nuclear installations [13]. The principle for both systems was to suspend the protected object on a double chamber interconnected pneumatic springs. In both cases the devices were capable to eliminate or limit vibrations of small amplitudes in comparison to the scale of the whole structure [14]. Since the devices were designed for mitigation of vibration of small amplitudes several simplifications to the modelling approach could be assumed. For instance: linearisation of the mechanical response around the force equilibrium position or description of the capillary flow with the Poiseuille relation derived for viscous flow [15].

In this paper we described an investigation on thermodynamic modelling of a semi-active pneumatic piston-cylinder device that was devoted to operate as an impact absorber. This application introduced a high demand on the dynamic responsiveness. The device should response adequately within 500 Hz bandwidth. For that reason the modelling approach was carefully analysed.

The paper is divided into four sections, which are organised as follows. Section 2 introduces the structure of the absorber and the principle of its operation. In section 3 the analysis of the system is presented and mathematical model proposed. In the following part the experimentally obtained results are compared with the results of simulation, which concludes the paper.

II. PRINCIPLE OF OPERATION OF THE ABSORBER

The conceptual pneumatic adaptive shock absorber was considered as a piston-cylinder device equipped with a fast operated valve positioned in the piston. A schematic structure of the considered device is presented in Figure 1.

*Research supported by National Science Centre of Poland, project no. 2012/05/B/ST8/02971.

[1]Grzegorz Mikułowski (corresponding author) is with Institute of Fundamental Technological Reaseatch (IPPT PAN), Department of Smart Technologies, ul. Pawińskiego 5B, 02-106, Warszawa, Poland, e-mail: gmikulow@ippt.pan.pl

[2]Rafał Wiszowaty is with Institute of Fundamental Technological Reaseatch (IPPT PAN), Department of Smart Technologies, ul. Pawińskiego 5B, 02-106, Warszawa, Poland, e-mail: rwisz@ippt.pan.pl

[3]Jan Holnicki-Szulc is with Institute of Fundamental Technological Reaseatch (IPPT PAN), Department of Smart Technologies, ul. Pawińskiego 5B, 02-106, Warszawa, Poland, e-mail:holnicki@ippt.pan.pl

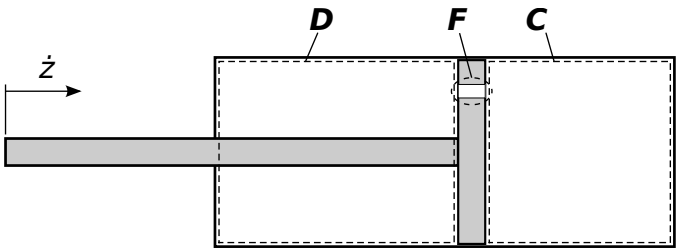

Fig. 1. Schematic cross-section of an adaptive pneumatic shock-absorber

In principle, the system was analysed as being able to transfer a mechanical energy of a moving body connected with the piston rod into the internal energy of the gas and then to dissipate it by heat.

The dissipation process of an external mechanical energy by means of the absorber was conducted within three phases. The first phase was a conversion of the mechanical energy into thermodynamic energy of gas in the process of simultaneous expansion and contraction of the media in two internal volumes of the absorber C and D (Figure 1). In the following phase, in order to counteract the releasing of the accumulated energy via the spring-back effect, a flow through the piston was allowed in a controlled manner and therefore, a spontaneous expansion of the gas within the cylinder's volumes was produced. And the last phase was cooling the gas in the cylinder by heat transfer to the surroundings. The macroscopic effect that was intended to be achieved was elasto-plastic like response with a controllable level of plastic yielding.

The adjustable expansion process allowed to control the reacting force value and the absorbed energy magnitude was adapted in dependence of the applied algorithm. The flow process between the volumes was to be conducted within a period estimated as several milliseconds for the analysed range of impact velocities: up to 5 m/s. Technically it was possible to realize this task via employment of a fast operated piezoelectric valve. With this technique it was possible to control the absorption processes by adjusting the level of mechanical energy dissipated by the system and to control the deceleration and forces acting on the protected objects.

The control algorithm for a considerable dissipative adaptive system based on the absorbers would respond adequately to wide spectrum of excitations. For the analysed conceptual, one-dimensional adaptive absorption system, the process was based on three-stage operation. During the first stage, the energy of the moving object was estimated with a system of electronic non-contact sensors in a few milliseconds before the impact event. The task can be performed with a real-time system realizing velocity determination [16] or more advanced systems devoted to mass identification [17] or determination of excitation time histories [18]. After the magnitude of the energy to be dissipated was determined, the mechanical energy of the object was converted into enthalpy increase of the gas in the absorber. In the third stage, an electronically controlled process of the accumulated energy dissipation was conducted via a thermodynamically irreversible process of spontaneous gas expansion between the internal chambers of the absorber. This process was monitored and controlled by means of electronic pressure and temperature sensors positioned in the absorber's cylinder. The piezoelectrically driven valve, controlled by signal based on pressure and temperature values, enabled to adjust the process of the gas expansion and therefore to maintain the magnitude of the converted energy on a predefined level in accordance to the piston position. The algorithm was implemented on a control module that was using signals related to pressures and temperatures levels in the chambers.

The presented configuration of the absorber enabled to generate the reaction force on a desired level in dependence on the magnitude of energy to be dissipated. Therefore, the device could be considered as adaptive.

III. Mathematical model of the absorber

In order to reflect the mechanical response of the system, it was analysed in terms of mechanical and thermodynamic processes. The analysis was divided into three sections devoted to: dynamics of the piston treated as a rigid body, thermodynamics of the gas in the absorber's chambers and gas flow through the valve. There were distinguished two control volumes: C and D in the absorber's interior as depicted in Figure 2.

A. Forces acting on the piston treated as a rigid body

The total force equilibrium of the piston can be defined as:

$$F_e(t) - F_{p_C}(z, T_C, m_C) + F_{p_D}(z, T_D, m_D) + F_a - F_f(\dot{z}) = 0 \tag{1}$$

where: F_e – external excitation, F_{p_C} – force resulting from pressure p_C in the chamber C, F_{p_D} – force resulting from pressure p_D in the chamber D, F_a – force resulting from ambient pressure, F_f – friction force, z – displacement of the piston, T_C, T_D – gas temperatures in corresponding volumes, m_C – mass of the gas in volume C, m_D – mass of the gas in volume D.

The forces acting on the piston can be formulated as:

$$F_{p_C} = p_C\left(T_C, \rho_C\right) \cdot A_1, \tag{2}$$

$$F_{p_D} = p_D\left(T_D, \rho_D\right) \cdot (A_1 - A_2),] \tag{3}$$

$$F_a = p_a A_2, \tag{4}$$

$$F_f = f_f \, sgn(\dot{z}) \tag{5}$$

denoted as: ρ_C, ρ_D – gas densities in chambers C and D, A_1 – effective piston area, A_2 – area of piston rod radial cross section, p_a – ambient pressure, f_f – friction coefficient.

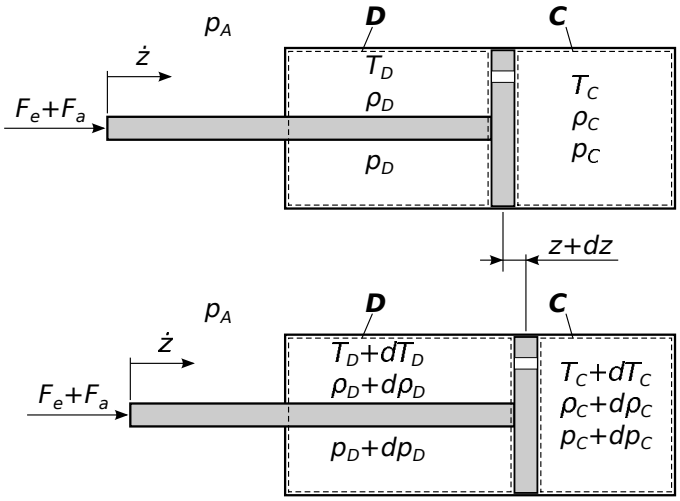

Fig. 2. Scheme of the absorber

B. Thermodynamics of the gas in the absorber's chambers

The thermodynamic processes in the chambers were described with energy balance equations for top and bottom chamber, gas state equation and the relations governing the compressible fluid flow through the applied valve. The following assumptions were introduced during the analysis:

Ideal gas law

The gas filling volumes C and D was dry nitrogen, which was operated in the temperature above 200 K. Therefore, the assumption of the ideal gas was approved to be valid as the state of the gas was not approaching the critical point:

$$pv = RT \tag{6}$$

Uniform-state, uniform-flow process

Volume of the absorber's chambers and the speed of sound determine the time necessary for the gas to reach a uniform state. Since the considered chambers have dimensions on the order of 0.1 m and the speed of sound in normal conditions is approximately 340 m/s, the time it takes for the gas to reach a uniform pressure is negligibly small. Furthermore, the gas was assumed to mix instantaneously in the chambers, so the fluid was described by a uniformly distributed temperature in each chamber.

The further assumptions regarding gas dynamics were formulated as follows:

1) the state of the mass entering the valve was constant within the time steps and uniform over the volume of the valve where flow occurred,
2) the state of the mass within the control volumes could changed with time, but at any instant of time the state was uniform over the entire control volume,
3) the changes in kinetic energy of the gas were negligible,
4) the inertia and gravity forces of the gas were negligible.

Thermodynamic processes assumption

During the operation of the absorber, the gas was simultaneously compressed and decompressed in the chambers separated by the piston. The processes that took place could not be defined clearly as isothermal or adiabatic, since the expected rate of the process varied in dependence on the work conditions. For low velocity operations the processes could be treated as close to isothermal, since in such a case there was enough time to evacuate the energy by heat from the gas and its temperature remained unchanged. Otherwise, during fast processes the time was too short for the energy transfer in a significant magnitude to occur and therefore, the process could be treated as adiabatic. In between of the mentioned cases the process could be assumed to be polytropic:

$$p\,V^n = const. \tag{7}$$

where p – pressure, V – volume, n – polytropic coefficient. Adequately, $n = 1$ for isothermal process and $n = 1.4$ for adiabatic process.

In the analysed case, the model being developed was to reflect the behaviour of the absorber under wide range of operational conditions and be valid with no regard to the piston kinematics. Therefore, it was not adequate to describe the gas processes with the polytropic model with constant parameter n.

Therefore, in each step of the computations, the state of the gas in the control volumes was calculated in the following manner: primarily, by assumption that the gas changed its state adiabatically ($n = \kappa = 1.4$) and secondarily, the gas state parameters were recalculated on the basis of internal energy balance analysis with the influence of heat exchange taken into account. That approach allowed to update the final state of the gas. During each time step the following analysis was conducted:

1) determination of the gas state on the basis of the adiabatic model,
2) determination of the internal energy of the gas,
3) determination of the heat exchange between the control volume and the surroundings,
4) determination of the energy balance in the control volume,
5) recalculation of the gas state parameters on the basis of the energy balance equation.

This method allowed to account for changes in the thermodynamic processes in time and therefore to reflect the polytropic-like process, as if the parameter n was changing with time dependently on the operating conditions.

The assumption of the ideal gas allowed to calculate the thermodynamic state parameters as:

$$p_C(z) = p_{C1}\left(\frac{V_{C1}}{V_{C1} + A_1\,z}\right)^{\kappa}, \tag{8}$$

and:

$$T_C(z) = T_{C1}\left(\frac{V_{C1}}{V_{C1} + A_1\,z}\right)^{\kappa-1}, \tag{9}$$

where: V_{C1} – initial volume of chamber C, p_{C1} – initial pressure in chamber C, T_{C1} – initial temperature in chamber C.

$$p_D(z) = p_{D1}\left(\frac{V_{D1}}{V_{D1} - (A_1 - A_2)\,z}\right)^{\kappa}, \tag{10}$$

and:

$$T_D(z) = T_{D1}\left(\frac{V_{D1}}{V_{D1} - (A_1 - A_2)\,z}\right)^{\kappa-1} \tag{11}$$

where: V_{D1} – initial volume of chamber D, p_{D1} – initial pressure in chamber D, T_{D1} – initial temperature in chamber D.

Mass continuity and energy balance

For case of a general control volume with the uniformity assumptions the mass continuity equations for volumes C and D take the form respectively:

$$\dot{m_C} + \dot{m}_{Ce} - \dot{m}_{Ci} = 0, \quad \dot{m}_D + \dot{m}_{De} - \dot{m}_{Di} = 0 \tag{12}$$

where: m_C – mass of the gas in the control volume C, $\dot{m}_{Ce}$ – mass leaving the control volume C, $\dot{m}_{Ci}$ – mass entering the control volume C. Quantities m_D, $\dot{m}_{De}$, $\dot{m}_{Di}$ concern to the volume D and have analogous meaning.

The energy balance in the control volume can be noted as [21]:

$$\dot{Q}_C + \dot{m}_{Ci}\,h_D = \dot{W}_C + \dot{m}_{Ce}\,h_C + \frac{d}{dt}\left(m_C\,u_C\right), \tag{13}$$

$$\dot{Q}_D + \dot{m}_{Di}\,h_C = \dot{W}_D + \dot{m}_{De}\,h_D + \frac{d}{dt}\left(m_D\,u_D\right), \tag{14}$$

where: $\dot{Q}_C$ – heat transferred to the control volume C, h_C – specific enthalpy of the gas occupying control volume C, $\dot{W}_C$ – work done by the gas in the control volume C, u_C – specific internal energy of the gas in volume C. The quantities of index D describe the same values related to volume D.

The $\dot{Q}_C$ and $\dot{Q}_D$ depended on: (1) the difference between temperatures of the gas in the control volume and the surroundings, (2) the area of cylinder walls that the gas was in contact with, and (3) the material constants:

$$\dot{Q}_C = A_C(z) \cdot \alpha \cdot (T_C(t) - T_a(t)), \tag{15}$$

$$\dot{Q}_D = A_D(z) \cdot \alpha \cdot (T_D(t) - T_a(t)), \tag{16}$$

where: $A_C(z)$ – area of cylinder being in contact with gas in volume C, α – heat transfer coefficient, $T_a(t)$ – ambient temperature. The quantities of index D describe the same values related to volume D.

According to the continuity principle, the mass of gas leaving the control volume C was equal to the mass of gas entering the control volume D and vice versa. That led to the property of mass transferred between the volumes as:

$$\dot{m}_{Ct} = -\dot{m}_{Dt} \tag{17}$$

where

$$\dot{m}_{Ct} = \dot{m}_{Ce} - \dot{m}_{Ci} = \begin{cases} \dot{m}_{Ce} > 0 & when \quad p_C > p_D \\ -\dot{m}_{Ci} < 0 & when \quad p_C < p_D \end{cases} \tag{18}$$

as well

$$\dot{m}_{Dt} = \dot{m}_{De} - \dot{m}_{Di} = \begin{cases} \dot{m}_{De} > 0 & when \quad p_D > p_C \\ -\dot{m}_{Di} < 0 & when \quad p_D < p_C \end{cases} \tag{19}$$

Let us denote the transferred mass of gas as:

$$\dot{m}_t = \dot{m}_{Ct} = -\dot{m}_{Dt} \tag{20}$$

The specific enthalpies h_C and h_D of the gas in volumes C and D respectively were different in general. Therefore when the assumption of isenthalpic flow through the valve hold true, the specific enthalpy of the gas transferred between the volumes depended on the flow direction, and was equal to:

$$h_C = c_p T_C, \quad \text{when the gas leaved volume C} \tag{21}$$

and

$$h_D = c_p T_D, \quad \text{when the gas leaved volume D} \tag{22}$$

where: c_p – specific heat of the gas by the constant pressure.

In the considered range of the gas temperatures 200 – 400 K the specific heat of the gas was assumed to be constant.

The work done by the gas was defined for the particular control volumes as:

$$\dot{W}_C = p_C(z)\frac{dV_C}{dt}, \tag{23}$$

$$\dot{W}_D = p_D(z)\frac{dV_D}{dt}. \tag{24}$$

C. Mass flow rate on the valve

The valve was assumed to operate in a bistable, on-off mode. In the opened position the mass flow rate $\dot{m}_t$ (20) depended on Mach number Ma; on the gas state parameters at the inlet of the valve: p_0, T_0. In the model, the values were taken as equal to the actual values of the parameters in the control volumes as:

$$p_0 = \begin{cases} p_C & when \quad p_C > p_D \\ p_D & when \quad p_C < p_D \end{cases} \tag{25}$$

$$T_0 = \begin{cases} T_C & when \quad p_C > p_D \\ T_D & when \quad p_C < p_D \end{cases} \tag{26}$$

The flow was assumed to be an adiabatic process (there was no heat exchange between the gas and the walls of the valve and but with mechanical losses [19]). The flow losses were described with the discharge coefficient C_d treated as a characteristic design parameter of the valve [20]. In accordance to the throttled flow theory [20], [21], the flow was assumed to be choked when the Mach number Ma was close to 1. Then the mass flow rate of the gas exchanged between the chambers can be expressed in the form [19]:

$$\dot{m}_t = \begin{cases} C_d \frac{Ma\, A\, p_0 \sqrt{\frac{\kappa}{RT_0}}}{\left[1+\frac{(\kappa-1)Ma^2}{2}\right]^{\frac{\kappa+1}{2(\kappa-1)}}}, & \text{if } Ma < 1 \\ C_d\, A\, p_0 \sqrt{\frac{\kappa}{RT_0}} \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}}, & \text{if } Ma = 1 \end{cases} \tag{27}$$

where: A – cross section of the flow duct, κ – isentropic coefficient, R – gas constant.

D. Governing equations

By substituting equations: (23), (24), (12) to the equations of energy balance (13) and (14), and taking into consideration (18), (19) and (21), (22), the internal energy of the gas in the control volume C and D can be calculated as:

For $p_C > p_D$:

$$\frac{d}{dt}(mu)_C + p_C(z)A_1\frac{dz}{dt} + \dot{m}_t h_{tC} = 0 \tag{28}$$

$$\frac{d}{dt}(mu)_D - p_D(z)(A_1 - A_2)\frac{dz}{dt} - \dot{m}_t h_{tC} = 0 \tag{29}$$

For $p_C < p_D$:

$$\frac{d}{dt}(mu)_C + p_C(z)A_1\frac{dz}{dt} - \dot{m}_t h_{tD} = 0 \tag{30}$$

$$\frac{d}{dt}(mu)_D - p_D(z)(A_1 - A_2)\frac{dz}{dt} + \dot{m}_t h_{tD} = 0 \tag{31}$$

E. Control algorithm formulation

The control procedure for the pneumatic adaptive shock absorber was aimed at maintaining a predefined level of difference between forces (2) and (3) acting on the piston. The pressure of the gas had direct impact on the reaction force generated by the absorber. The valve opening control function had the following form:

$$C(t) = \begin{cases} C_{open}, & \text{if } F_e(t) > F_{ref} + \Delta F \\ C_{close}, & \text{if } F_e(t) < F_{ref} - \Delta F \end{cases} \tag{32}$$

where: $C_{open/close}$ – signal of opening/closing the valve, $F_e(t)$ – reaction force, F_{ref} – reference level, ΔF – tolerance range.

IV. Experimental program conducted on the adaptive absorber

The considered concept of the adaptive pneumatic absorber was demonstrated in laboratory scale with a small demonstrator presented in Figure 3. The device was designed in dimensions that allowed to dissipate energy of 40 J per stroke.

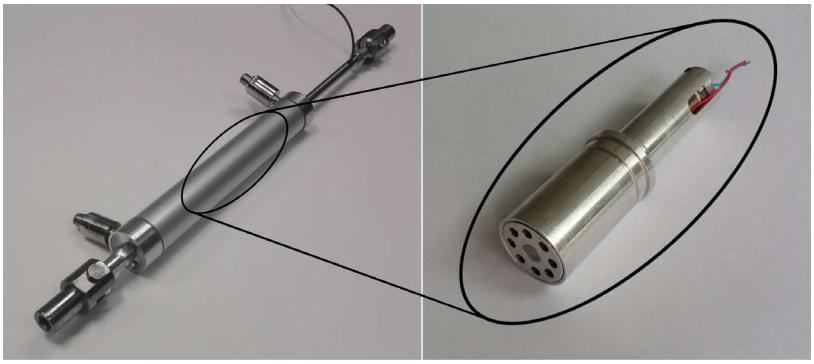

Fig. 3. View of the absorber and the piezoelectric valve

The demonstrator was a piston-cylinder device equipped with a valve positioned in the piston. In order to ensure the system controllability, the proper bandwidth of its excitation was realized with a piezoelectric stack. The adaptive control system of the demonstrator was fed with signals from two pressure and two temperature sensors positioned within the absorber's housing as depicted in Figure 4.

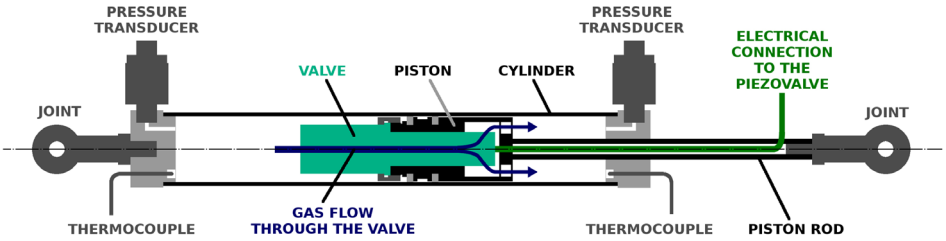

Fig. 4. Main elements of the absorber

The testing program of the absorber was conducted by means of a hydraulic excitation system. The system consisted of an electronically controlled hydraulic actuator, positioned horizontally in-line with the tested adaptive absorber (Figure 5). The actuator-absorber connections were realized via cup-and-ball joints in order to prevent the transmission of bending moments and shear forces into the structure of the tested specimen. The actuation system enabled to examine the absorber under periodic axial loading with the displacement reference signal.

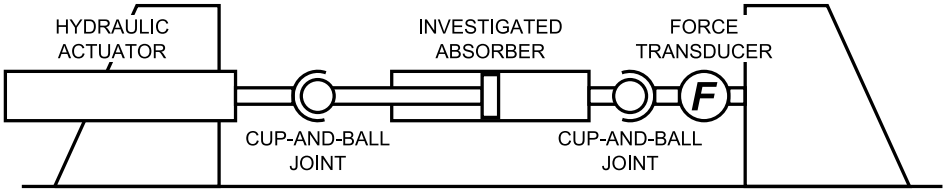

Fig. 5. Scheme of the testing stand

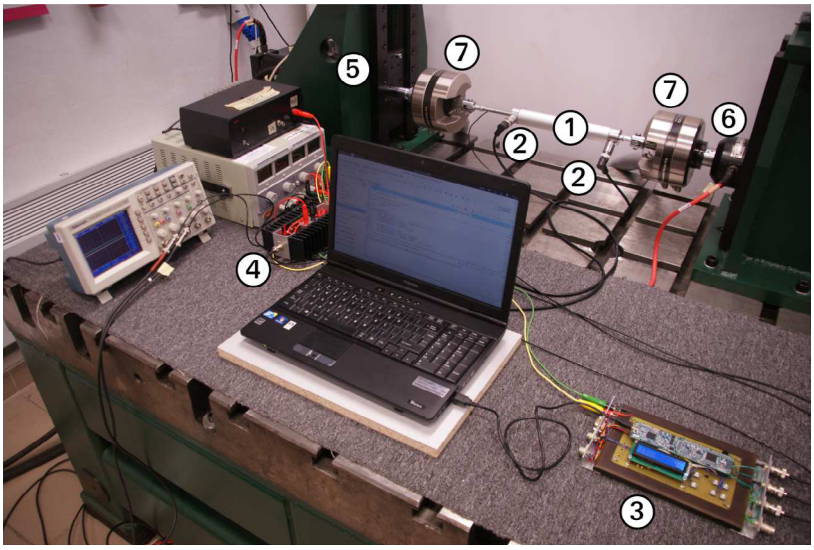

Fig. 6. Laboratory testing stand

The complete experimental stand is depicted in Figure 6 and consisted of:

1) adaptive absorber,
2) pressure transducers,
3) electronic control unit,
4) piezoelectic valve supplier,
5) hydraulic excitation system,
6) force transducer,
7) hydraulic grips.

The testing program was defined in a way to allow verification of the mathematical model under variety of excitation condition. The parameters were: rate of the displacement, initial pressure value inside of the cylinder and the expected (controlled) magnitude of the maximal reaction force. The testing program contained as follows:

1) kinematic excitation with triangle signal, amplitude 40 mm, frequency adopted to the required velocity of the piston
2) velocity of the piston: 0.25 m/s,
3) initial gas pressure in the cylinder: 0.5 MPa,
4) ambient pressure: 100 kPa,
5) ambient temperature: 292 K,
6) magnitude of the maximal reaction force: 100 N, 200 N, 300 N, 400 N, 500 N.

V. Results

The results obtained during testing sesion were compared to data coming from the numerical simulation. The comparison was conducted for several operation conditions of the absorber in accordance with the testing program. The principle objective of this task was to verify the proposed mathematical model versus experimental data. The parameters of the numerical model utilized in the example analysis are summarised in Table 1.

TABLE I
MODEL PARAMETERS

Piston area	A_1	8.042e-04	[m^2]
Piston rod area	A_2	7.854e-05	[m^2]
Cylinder wall thickness	d	0.002	[m]
Cylinder inner diameter	D	0.032	[m]
Initial volume C	V_{Cini}	9.402e-05	[m^3]
Initial volume D	V_{Dini}	1.165e-05	[m^3]
Adiabatic constant	$n = \kappa$	1.4	[-]
Gas constant	R	296.8	[J/(kg K)]
Specific heat by constant volume	c_v	743	[J/(kg K)]
Specific heat by constant pressure	c_p	1039	[J/(kg K)]
Heat transfer coefficient	α	20	[W/(m^{2}K)]
Valve discharge coefficient	C_d	0.6	[-]
Ambient pressure	p_a	100e3	[Pa]
Ambient temperature	T_a	292	[K]

Figure 7 depicts the absorber's reaction forces in the domain of displacement for five levels of the control parameter. The absorber's response measured during the testing program was compared with the results of numerical modelling. The plots prove that the proposed model satisfactorily reflects the behaviour of the laboratory demonstrator for all tested operation conditions.

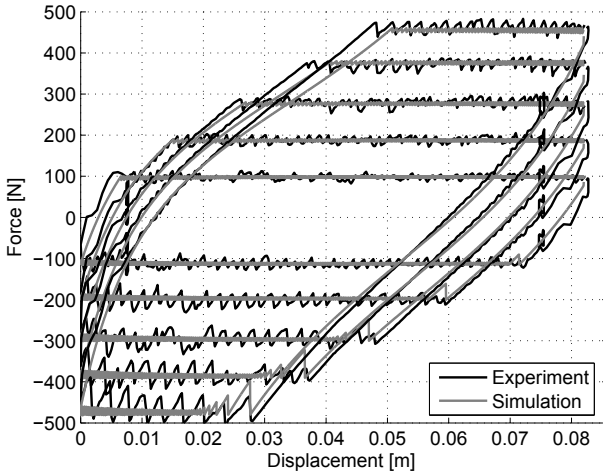

Fig. 7. Reaction force of the absorber in domain of the piston displacement.

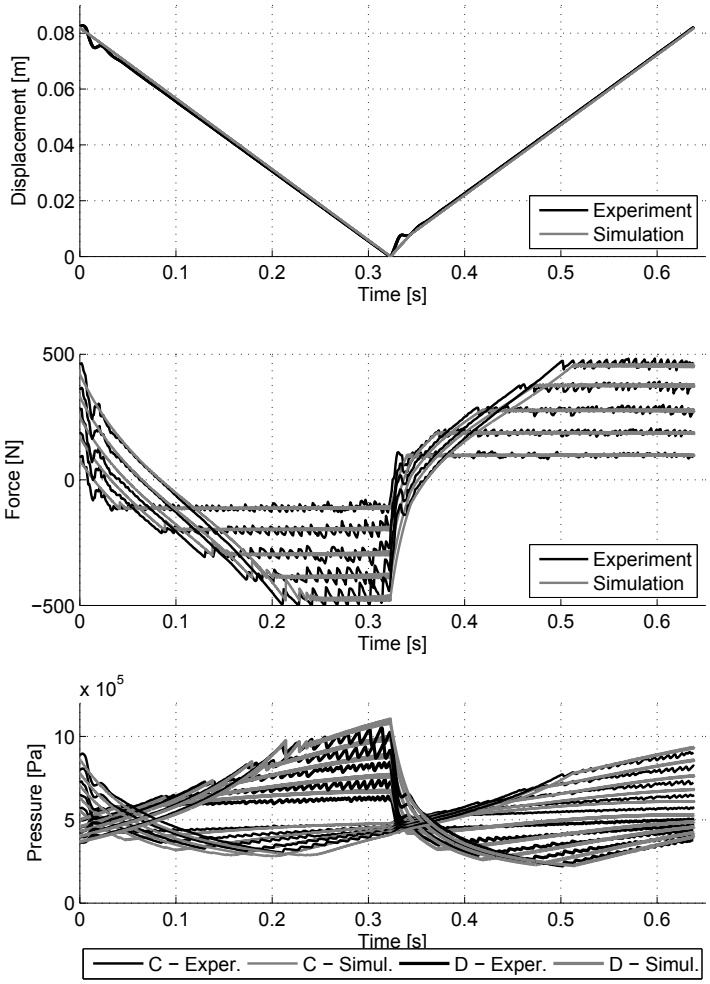

Fig. 8. Excitation path (displacement) and response (force and pressure) of the adaptive absorber. Comparison of experimental and numerical results.

Figure 8 depicts the absorbers response in the domain of time. The top plot represents the excitation path, which was realized during tests. The middle plot depicts the reaction forces and the bottom plot shows pressures in volumes C and D. All of the presented results confirm the proper functioning of the presented mathematical model.

VI. CONCLUSIONS

In this paper a method for mathematical modelling of an adaptive pneumatic absorber was presented. The absorber is a new conceptual device that might be utilized in various fields of technique and therefore, from design point of view, it was crucial to develop a method for reflecting its performance by means of numerical methods. The presented method's advantage is its simplicity and relatively small demands for computation resources in comparison to CFD methods of modelling. Therefore, the proposed model can be treated as a tool complementing the CFD methods of design. The model can be successfully used as tool for simulating of extended systems consisting of the adaptive absorbers.

ACKNOWLEDGMENT

The support of Structural Funds in the Operational Programme Innovative Economy (IE OP) financed from the European Regional Development Fund Projects Modern material technologies in aerospace industry (POIG.0101.02-00-015/08) and Polish National Science Centre Project AIA (DEC 2012/05/B/ST8/02971) are gratefully acknowledged.

REFERENCES

[1] C.M. Harris and A.G. Piersol, editors. Harris' Shock and Vibration Handbook (5th edition).McGraw-Hill, 2002.

[2] J. Holnicki-Szulc, P. Pawłlowski, M. Wiklo, Design of adaptive structures under random impact conditions, Advances in Smart Technologies in Structural Engineering, Springer, pp. 45–67, 2004.

[3] G. Mikułowski and Ł. Jankowski, Adaptive landing gear: optimum control strategy and potential for improvement. Shock and Vibration, vol. 16, pp. 175–194, 2009.

[4] C. Graczykowski, J. Holnicki-Szulc, Protecting offshore wind turbines against ship impacts by means of Adaptive Inflatable Structures, Shock and Vibration, IOS Press, vol.16, no.4, pp. 335-353, 2009.

[5] B.F. Spencer-Jr., S.J. Dyke, M.K. Sain, and J.D. Carlson, Phenomenological model of a magnetorheological damper, Journal of Engineering Mechanics, vol. 123, no. 3, pp.230 – 238, 1997.

[6] A.Pregowska, R. Konowrocki, T. Szolc, On the semi-active control method for torsional vibrations in electro-mechanical systems by means of rotary actuators with a magneto-rheological fluid, Journal of theoretical and applied mechanics, vol. 51, no. 4, pp. 979–992, 2013.

[7] B. Dyniewicz, A. Pregowska, C. I. Bajer, Adaptive control of a rotating system, Mechanical systems and signal processing, vol. 43, no. 1-2, pp. 90–102, 2014.

[8] G. Mikułowski and J. Holnicki-Szulc, Adaptive landing gear concept – feedback control validation, Smart Mater. Struct., vol. 16, pp. 2146–2158, Jan. 2007.

[9] G. Yang, B. Spencer, H. Jung, and J. Carlson, Dynamic modeling of large-scale magnetorheological damper systems for civil engineering applications, Journal of Engineering Mechanics, vol. 130, no. 9, pp.1107–1114, 2004.

[10] M. Makowski, L. Knap, and W. Grzesikiewicz, Vibration investigation of vehicle equipped with controlled piezoelectric dampers, Journal of KONES Powertrain and Transport, vol. 18, no. 4, 2011.

[11] G. Mikułowski, R. Leletty, Advanced Landing Gears for Improved Impact Absorption, in Proc. 11th International Conference on New Actuators, Bremen, pp. 363-366, 2008.

[12] J-H. Lee and K-J. Kim, Modelling of nonlinear complex stiffness of dual-chamber pneumatic spring for precision vibration isolations, Journal of Sound and Vibration, vol. 301, pp. 909–926, 2007.

[13] G. S. Aver'yanov, R. N. Khamitov, and A. V. Zubarev, Dynamics of oscillatory systems with controllable shock absorbers, Russian Engineering Research, vol. 28, no. 6, pp. 543–547, 2008.

[14] G.S. Aver'yanov, R.N. Khamitov, A.V. Zubarev, and A.A. Kozhushko, Dynamics of controlled pneumatic shock-absorber systems for large objects, Russian Engineering Research, vol. 28, no. 7, pp. 640–642, 2008.

[15] C. Erin, B. Wilson, and J. Zapfe, An improved model of a pneumatic vibration isolator: theory and experiment, Journal of Sound and Vibration, vol. 218, no. 1, pp. 81–101, 1998.

[16] K. Sekuła, C. Graczykowski, J. Holnicki-Szulc, On-line impact load identification, Shock and vibration, vol. 20, no. 1, pp. 123–141, 2013.

[17] G. Suwała, Ł. Jankowski, A model-free method for identification of mass modifications, Structural control & Health monitoring, vol. 19, no. 2, pp. 216–230, 2012.

[18] Q. Zhang, Ł. Jankowski, Z. Duan, Simultaneous identification of excitation time histories and parametrized structural damages, Mechanical systems and signal processing, vol. 33, pp. 56–68, 2012.

[19] G. Mikułowski, R. Wiszowaty, J. Holnicki-Szulc, Characterisation of a piezo-electric Valve for an adaptive pneumatic shock-absorber, Smart Materials and Structures, vol. 22, no. 125011 (12pp), 2013.

[20] M. A. Boles Y. A. Cengel, Thermodynamics, McGraw Hill, second edition, 1994.

[21] R. E. Sonntag G. J. van Wylen, Fundamentals of classical thermodynamics, Wiley, 1965.