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Development of self-adaptive systems for mitigation of response under dynamic excitation

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Contents

Abstract

The thesis concerns the problems of adaptive absorption of impact energy, and its main goal is the development of self-adaptive absorbers for mitigation of the system response under the dynamic excitation.

The problems of mitigating the response of a dynamically excited system are important during design of various engineering constructions, machines and devices. Dynamically excited systems include landing gears, airdrop systems, rescue cushions, bicycle dampers, vehicle bumpers, road barriers, car airbags. The impact energy absorbers for these applications have been developed for many years and they are represented by both passive systems and controllable semi-active systems. The second group is becoming more and more popular due to technological progress in the field of smart materials as well as measurement and control systems. Nevertheless, in order to ensure optimal energy absorption the currently known impact absorbers require the knowledge about the mass of impacting object and its initial velocity. In practice, precise identification of all impact parameters is difficult to be realized due to the short time of the impact energy absorption process. Moreover, in the model used for determination of the optimal control strategy for the absorber, some uncertainties may occur, e.g., friction varying during the exploitation of the device.

The identified limitations and imperfections of the currently used impact absorbers became the motivation to undertake the research and develop new adaptive systems. As part of conducted research, two groups of adaptive absorbers have been distinguished: adaptable devices and self-adaptive devices. The first group are systems which adapt to conditions of the excitation based on the identification of its parameters. However, they are not able to detect and adapt to changes in the impact energy absorption process, and as a result the assumed control strategy may not be optimal or even it can lead to ineffective, dangerous operation of the system. In turn, self-adaptive absorbers are able to adapt to different excitation conditions, as well as to changes occurring in the process of impact energy absorption, and thus they are fully adaptive systems.

The PhD thesis consists of a series of five articles published in journals from the A list of the Ministry of Science and Higher Education. The first two articles of this dissertation present original proposals of adaptive impact energy absorbers, which ensure efficient mitigation of the impact in a passive manner. The characteristic feature of elaborated systems is the fact that despite their

passive operation they allow to obtain a response comparable to the response of semi-active devices. In addition, the use of the single reconfiguration technique just before the impact or at the very beginning of the impact absorption process ensures adaptation of the absorber to the excitation conditions. The third article discusses a controllable inertial absorber, which is a transition from adaptable systems to self-adaptive systems presented in the fourth and fifth article. The fourth article demonstrates the influence of inaccuracies in the identification of the excitation and influence of unpredicted disturbances on the operation of currently used adaptable systems. Simultaneously, a control method with kinematic feedback that guarantees self-adaptation of the fluid-based absorber is introduced. In the fifth article, this method is further developed and the possibility of its application in the case of a series of impacts, including bi-directional excitation, is analyzed. The developed systems are compared with standard, adaptive and optimal solutions.

Streszczenie

Tematyka rozprawy doktorskiej obejmuje zagadnienia adaptacyjnej absorpcji energii uderzeń, a nadrzędnym celem pracy jest opracowanie absorberów samo-adaptacyjnych do łagodzenia odpowiedzi układu pod działaniem wymuszenia dynamicznego.

Zagadnienia łagodzenia odpowiedzi układu wymuszonego dynamicznie są istotne podczas projektowania różnorodnych konstrukcji inżynierskich, maszyn i urządzeń. Układami wymuszanymi dynamicznie są m.in. podwozia lotnicze, systemy zrzutu ładunku, skokochrony, tłumiki rowerowe, zderzaki pojazdów, bariery drogowe, samochodowe poduszki powietrzne. Absorbery energii uderzeń dla wymienionych zastosowań są rozwijane od wielu lat i reprezentowane są zarówno przez układy pasywne, jak i sterowalne układy pół-aktywne. Te drugie stają się coraz bardziej popularne ze względu na rozwój technologiczny w obszarze materiałów inteligentnych oraz układów pomiarowych i sterujących. Niemniej jednak, w celu zapewnienia optymalnej absorpcji energii uderzenia znane dotychczas absorbery energii uderzeń wymagają znajomości masy obiektu uderzającego oraz jego prędkości początkowej. W praktyce precyzyjna identyfikacja wszystkich parametrów uderzenia jest trudna do zrealizowania ze względu na krótki czas procesu absorpcji energii uderzenia. Ponadto, w modelu wykorzystywanym do wyznaczenia optymalnej strategii sterowania absorberem mogą pojawić się pewne niepewności takie jak np. zmieniające się w trakcie eksploatacji urządzenia tarcie.

Zidentyfikowane ograniczenia i niedoskonałości obecnie stosowanych absorberów energii uderzenia stały się motywacją do podjęcia badań i opracowania nowych układów adaptacyjnych. W ramach przeprowadzonych badań wyróżniono dwie grupy absorberów adaptacyjnych: urządzenia adaptowalne oraz urządzenia samo-adaptacyjne. Pierwsza grupa to układy, które adaptują się do warunków wymuszenia na podstawie identyfikacji jego parametrów. Nie są jednak w stanie wykryć i dostosować się do zmian w procesie absorpcji energii, a w rezultacie przyjęta strategia sterowania może okazać się nieoptymalna lub nawet prowadzić do nieskutecznej, niebezpiecznej pracy układu. Z kolei absorbery samo-adaptacyjne są w stanie zaadaptować się zarówno do różnych warunków wymuszenia, jak i do zmian występujących w procesie absorpcji energii uderzenia, a więc są układami w pełni adaptacyjnymi.

Praca doktorska składa się z cyklu pięciu artykułów wydanych w czasopismach z listy A Ministerstwa Nauki i Szkolnictwa Wyższego. Pierwsze dwa artykuły niniejszej rozprawy przedstawiają autorskie propozycje adaptowalnych absorberów energii uderzenia, które zapewniają efektywne złagodzenie uderzenia w sposób pasywny. Cechą wyróżniającą opracowane układy jest fakt, że mimo pasywnej pracy pozwalają na uzyskanie odpowiedzi porównywalnej z odpowiedzią absorberów pół-aktywnych. Ponadto, wykorzystanie techniki jednokrotnej rekonfiguracji układu tuż przed pojawieniem się wymuszenia lub na samym początku procesu absorpcji energii uderzenia zapewnia adaptację absorbera do warunków wymuszenia. W trzecim artykule omawiany jest sterowalny absorber inercyjny, który stanowi przejście od układów adaptowalnych do układów samo-adaptacyjnych zaprezentowanych w artykule czwartym i piątym. Artykuł czwarty demonstruje wpływ niedokładności w identyfikacji wymuszenia, jak i nieprzewidzianych zakłóceń na pracę obecnie stosowanych absorberów adaptowalnych. Jednocześnie wprowadzona zostaje metoda sterowania ze sprzężeniem kinematycznym gwarantującym samo-adaptacyjność absorbera przepływowego. W artykule piątym metoda ta jest rozwinięta i możliwość jej zastosowania w przypadku serii uderzeń, w tym dwukierunkowych, jest analizowana. Opracowane układy są porównywane zarówno z rozwiązaniami klasycznymi, jak i adaptacyjnymi i optymalnymi.

Structure of the thesis

This thesis consists of following five original articles:

A. Adaptable pneumatic shock-absorber

Faraj R., Graczykowski C., Holnicki-Szulc J.

Journal of Vibration and Control, ISSN: 1077-5463, Vol. 25, No. 3, pp. 711-721, 2019.

Impact Factor 2017: 2.197
Points MNiSW: 45

B. Can the inerter be a successful shock-absorber? The case of a ball-screw inerter with a variable thread lead

Faraj R., Jankowski Ł., Graczykowski C., Holnicki-Szulc J.

Journal of The Franklin Institute, ISSN: 0016-0032, doi: 10.1016/j.jfranklin.2019.04.012, pp. 1-18, 2019.

Impact Factor 2017: 3.576
Points MNiSW: 40

C. Adaptive inertial shock-absorber

Faraj R., Holnicki-Szulc J., Knap L., Seńko J.

Smart Materials and Structures, Vol. 25, 035031, pp. 1-9, 2016.

Impact Factor 2016: 2.909
Points MNiSW: 40

D. Development of control systems for fluid-based adaptive impact absorbers

Graczykowski C., Faraj R.

Mechanical Systems and Signal Processing, ISSN: 0888-3270, Vol. 122, No. 2019, pp. 622-641, 2019.

Impact Factor 2017: 4.370
Points MNiSW: 45

E. Hybrid Prediction Control for self-adaptive fluid-based shock-absorbers

Faraj R., Graczykowski C.

Journal of Sound and Vibration, ISSN: 0022-460X, Vol. 449, pp. 427-446, 2019.

Impact Factor 2017: 2.618
Points MNiSW: 35

In all papers listed above the first author was also the corresponding author.
Impact Factors are given for the year of publication according to the Journal Citation Reports. For recent publications, the latest available values are given.

Points MNiSW are given according to the Annex to the communication of the Minister of Science and Higher Education from January 25, 2017.

Articles are provided in Section 5 and signed declarations of author contributions are presented in Section 6.

The research conducted by the author in the field of impact absorption has resulted in 5 original papers, 16 conference reports, 4 patents granted and 3 patent applications awaiting decision.

Content of the original articles is supplemented by additional materials, which include selected conference reports and patent descriptions. Conference papers discuss additional aspects of the concepts and their extensions. In turn, patents were prepared for protection of the intellectual property rights. The list of appendices is as follows:

a. **Appendix 1. High Performance Pneumatic Shock-absorbers for aeronautical applications**
   Supplement to the article A

   Faraj R.

   ICAS 2018, 31st Congress of the International Council of the Aeronautical Sciences, 2018-09-09/09-14, Belo Horizonte, Brazil, pp. 1-10, 2018.

b. **Appendix 2. Patent pending P. 419285, Pneumatic absorber with adaptable response characteristics, filled with atmospheric air, in particular for mitigation of cargo touchdown**
   Supplement to the article A

   Faraj R., Graczykowski C., Holnicki-Szulc J.

c. **Appendix 3. Patent pending P.428526, Ball-screw absorber with variable thread lead**
   Supplement to the article B

   Faraj R., Jankowski J., Graczykowski C., Holnicki-Szulc J.

d. **Appendix 4. Patent PL 227058, Inertial shock-absorber**
   Supplement to the article C

   Faraj R., Holnicki-Szulc J., Mróz A.

e. **Appendix 5. Self-adaptive fluid-based absorbers for impact mitigation and vibration damping**
   Supplement to the article D

   Graczykowski C., Faraj R.

   ISMA 2018, International Conference on Noise and Vibration Engineering, 2018-09-17/09-19, Leuven, Belgium, pp. 217-228, 2018.

In accordance with the specific goals and thesis of the dissertation the series of original papers can be divided into two parts, which are respectively devoted to:

- Elaboration, development and analysis of adaptable, semi-passive shock-absorbers – articles **A** and **B**.

  The paper **A** covers development and verification of the novel concept of semi-passive pneumatic shock-absorber, which provides adaptable performance due to single reconfiguration technique.

  > Appendix **1** presents supplementary discussion of practical realization of the reconfiguration mechanisms and extension of the concept to the double-chamber shock-absorber.
  > Appendix **2** as the patent description of the technical realization of the concept reveals exemplary design of the system.

  The paper **B** presents elaboration of the innovative concept of semi-passive inertial shock-absorber, which can also be adapted to different excitation conditions by the single reconfiguration of the system.

  > Appendix **3** describes the technical design of the absorber prepared for the patent protection purposes.

- Elaboration, development of control methods providing self-adaptive performance of shock-absorbers – articles **C**, **D**, **E**.

  The paper **C** introduces the concept of adaptive inertial shock-absorber, which allows for successful impact mitigation and can be used to implement different control strategies.

  > Appendix **4** reveals technical solution which has been patented.

  The paper **D** and **E** are devoted to development of new control technique for shock-absorbers, which is aimed at providing their self-adaptive performance. Discussion is presented using the example of single-chamber fluid-based absorbers. The paper **D** reveals limitations and insufficiencies of previously known control methods implemented in shock-absorbers and introduces new control method, which is based on kinematic feedback. The method is developed and further investigated in paper **E** where control system is implemented in double-chamber shock-absorber. The method is adjusted to the case of bi-directional excitations and it is compared with classical, adaptive and optimal control methods.

Appendix **5** is a conference report which discusses additional aspects of control system design and possibility of its application for mitigation of the system response under harmonic excitation.

1. Motivation

Dynamic excitations appear in numerous mechanical systems and, as a result, protection against shock and vibration is still a real engineering problem. Impact absorption processes last for very short time periods and only shocks of limited values of impact velocity can be safely absorbed and dissipated [1]. Consequently, the efficient mitigation of system response under dynamic excitation is not only a challenging requirement for newly designed structures and machines but it is also an attractive field of research. Problems of impact mitigation concern various systems such as aircraft landing gears [2, 3], suspensions of lunar-planetary landers [4], airdrop systems [5, 6], emergency landing airbags for drones [7], rescue cushions for evacuation of people from high buildings [8], protection of offshore constructions against ship impacts [9], as well as other systems, e.g., self-deployable structures [10]. In addition to specialized applications, shock-absorbing devices are also encountered in everyday life, e.g., in bicycle dampers and handlebars [11], helmets [12], bumpers of vehicles [13, 14], road barriers [15] or car airbags [16].

At present, more and more frequent practice is the usage of so-called smart devices, which are based on functional materials and advanced control systems. Large number of scientific papers and engineering reports reveal an increase of effectiveness and reliability of these solutions. Moreover, there are commercial implementations of advanced smart solutions, e.g., fully controlled car suspensions based on magnetorheological fluids [17]. In contrast, there are still some applications for which it is incessantly required to develop and improve current passive solutions. The illustrative example is the rescue cushions used by fire brigades. For this kind of system legal restrictions and operational requirements are very high and they include short time of system preparation, low mass of the system, incombustibility, simplicity of handling and many others. As a result, it is hard to propose the adequate electronically controlled solution, which will simultaneously meet all above demands. The attractive solutions for applications which require implementation of simple and reliable protection against impact and vibration can be a passive shock-absorber with several operational modes [18], airbag with valve of selectable relief pressure [19] or structure, which effective stiffness and energy absorption properties are adjusted before the impact by proper inflation [20]. In case the complexity of shock-absorbing system is not limited by practical restrictions, smart devices can be used in order to increase impact absorption efficiency, extend range of operational conditions or to provide adaptation to actual dynamic excitation. The exemplary smart systems for impact mitigation are among others: magnetorheological energy absorbers [21], electrorheological dampers [22] or pneumatic adaptive absorbers with piezoelectric valves [23].

According to conducted literature review, effectiveness of impact mitigation systems developed so far was based primarily on prediction [24] or identification of the impact [25 26]. Some contributions concerning methods for identification of the impact loading can be found [27 28]. Moreover, strategies for optimal mitigation of identified single impact were also proposed, e.g., [29]. Nevertheless, industrial implementation of impact mitigation systems which are based on loading identification, is still a challenging task. In addition, in practice uncertainties, disturbances or unexpected changes of absorber parameters can appear. As a result classical methods of shock-absorber adaptation may fail.

The variety of shock-absorbers and big number of their possible applications motivate to address insufficiencies of both passive and controlled solutions. As a result the general goal of presented research will include **development of more efficient and adjustable passive absorbers**, as well as **elaboration of control methods and algorithms** in order to ensure **robust and adaptive performance** of semi-active absorbers.

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2. Background

For clear discussion concerning research area of this dissertation this section is divided into three subsections. The first subsection is devoted to formulation of the impact mitigation problem which facilitates differentiation of absorber types and discussion on the control techniques in the further part of this work. In the second subsection various types of shock-absorbers are presented and their adaptive capabilities are analyzed. In the third subsection already known control methods are discussed.

Please note that nomenclature used in papers A-E may vary. For clearness of the dissertation a consistent nomenclature is introduced for discussion of the research background and presented articles.

2.1. Formulation of the impact mitigation problem

The system considered in the presented work is shown schematically in Fig. 1, which presents two possible approaches to the modelling of 1 degree-of-freedom (1DOF) system under impact excitation. The impact conditions are simulated by assuming initial velocity v_0 of the object of mass M, or alternatively the impact is defined by time-dependent external force of impulsive character F_{ext}. The object is amortized by the shock-absorber possessing variable parameters, which allow to obtain desired characteristics of the system under various excitation conditions. In this thesis several types of shock-absorbers are considered and they are discussed in more detail in Sec. 4.

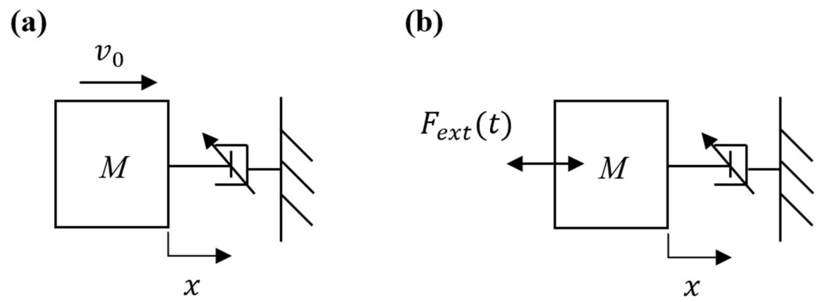

Figure 1. Schemes of the 1DOF system under impact conditions defined by: (a) initial velocity v_0, (b) impulsive external force $F_{ext}(t)$.

In order to evaluate performance of the developed shock-absorber or compare devices based on different operation principles or different physical phenomena, e.g., hydraulic vs. frictional dampers, several quality indices are commonly used. They include mean reaction force, impact absorption efficiency or so-called stroke efficiency [30]. Some quality measures can also be used to define design requirements, objective functions or constraints for optimization problems. Depending on the application of the shock-absorber, different criteria are considered as critical, e.g., for landing gears the reaction force is minimized [31 32] and the impact absorption efficiency is maximized [33], whereas for airdrop systems, it is more important to limit decelerations acting on the protected cargo [34]. Although these criteria seem to be similar for simple systems at the first glance, in case the absorber is optimized for a range of impact conditions the result will significantly vary so they has to be differentiated and chosen relevantly to the considered application. In reference to the classical vibration mitigation problems also other goal functions are used. They include among others the root mean square deflections of the object suspension, the root mean square accelerations of the object or indices combining contradictory goals by the use of weight coefficients [35]. Within presented dissertation the main goal of the shock-absorber operation is defined by two requirements. The first demand is the dissipation of entire impact energy E_{imp}, which consists of the initial kinetic energy of the amortized object $E_{init} = \frac{Mv_0^2}{2}$ and the work done by external force F_{ext} on the curve L:

$$E_{imp}(t) = \frac{Mv_0^2}{2} + \int_L \vec{F}_{ext}(t)\mathrm{d}\vec{s}, \tag{1}$$

where: $L: t \to x(t) \in [0, d],\ d$ is the absorber stroke.

In case the vertical orientation of the shock-absorber is considered, e.g., application as a suspension of landing gear or in the airdrop system, the change of potential energy should be included in (1).

Let now assume that the only force acting on the decelerated object, except the excitation force F_{ext}, is the absorber reaction force F_{abs} and the impact absorption process lasts from $t = 0$ to $t = T$, the absorber response has to meet the following integral condition:

$$\int_L \vec{F}_{abs}\mathrm{d}\vec{s} = E_{imp}(T), \tag{2}$$

where the final time T is defined as the moment when the system reaches static equilibrium, i.e., the resultant force as well as velocity of the object are equal to zero:

$$T = t : \ddot{x}(t) = 0, \dot{x}(t) = 0 \text{ and } x(t) \in [0, d]\,. \tag{3}$$

The second requirement for the system operation is as high as possible mitigation of the impact, which is manifested by the values of absorber reaction force F_{abs} or deceleration of the object. These quantities are treated as the quality index:

$$q = F_{abs} \text{ or } q = \ddot{x}. \tag{4}$$

In particular the maximal values of the quality index q should be minimized. As a result, in case the 1 DOF system is considered, the optimal impact mitigation problem can be defined in general form by the system of three equations:

$$m\ddot{x} + F_{abs}(x, \dot{x}, \ddot{x}, u, t) = F_{ext}(t),\ \dot{x}(0) = v_0,\ x(0) = 0, \tag{5a}$$

$$u = \arg\min_{\bar{u}} \max_{t} q(x, \dot{x}, \ddot{x}, \bar{u}, t)\,, \tag{5b}$$

$$\int_d F_{abs}(x, \dot{x}, \ddot{x}, u, t)\mathrm{d}\vec{s} = \frac{Mv_0^2}{2} + \int_L F_{ext}(t)\mathrm{d}\vec{s}, \tag{5c}$$

where: x is the displacement of working element of the shock-absorber, e.g., piston of the pneumatic damper or screw of the inerter, u – control variable, parameter of the system which is used for changing absorptive/dissipative characteristics of the shock-absorber, t is the time.

The system of equations (5) has to be complemented by definition of the absorber reaction force F_{abs}, which is typically determined from additional differential equations and kinematic constraints describing dynamics of the impact absorbing system. Specific models of shock-absorbers analyzed within the thesis were presented in papers A-E.

2.2. Shock-absorbers – state-of-the-art and state-of-the-practice

In both the scientific literature and the industrial applications a variety of shock-absorbing devices and dissipative structures can be found. Absorbers can be divided into several groups according to: physical phenomena during impact absorption process, type of control method used for adjustment of system characteristics or adaptive properties of the absorber.

Classification of shock-absorbers based on their adaptive capabilities is one of the original achievements of this contribution and it is discussed in Sec. 4.

As mentioned in Sec. 1 one of shock-absorbers' applications are road barriers and bumpers of vehicles. In these systems the impact absorption is still mainly realized in passive manner and for protection against crash the effect of plastic deformation of metal structures is utilized [36]. There is a large variety of deformable impact absorbers, e.g., tubes of different shapes [37, 38], honeycomb structures [39, 40] and thin-walled structures [41]. Within the group of passive shock-absorbers also reusable devices can be found. They consists of frictional dampers [42], inertial systems [43], hydraulic and pneumatic dampers [33], as well as passive airbags [44].

Another group of shock-absorbers, which are currently at the top of researchers' interests, are semi-active devices. They usually exploit the same physical phenomena as passive absorbers, but they are additionally equipped with electronics and executive components for adjustment of system parameters. The important feature of semi-active absorbers is the fact that they are controlled without introduction of additional energy to the protected system [45]. In case of frictional dampers the control parameter is the normal force [46, 47, 48], whereas characteristics of airbags and fluid-based absorbers are changed using controllable valves [49, 50, 51, 52] or functional materials [53, 54, 55]. Smart materials which can be applied for development of controllable shock-absorbers include electrorheological [56] and magnetorheological fluids [57], magnetorheological elastomers [58], magneto-strictive [59], electro-strictive [60] and piezoelectric materials [61], as well as shape memory alloys [62]. Moreover, also concepts for adjustment of classical deformable structures can be found in literature, e.g., pressurizing the vehicle longitudinal frontal members [63] or de-stiffening of the deformation zone of the front-end structure of motor vehicle by the use of pyrotechnic detachable connectors [64]. The variety of functional materials and smart devices, together with their higher and higher efficiency and reliability, allows to solve real engineering problems by the use of semi-active control.

For completeness of the discussion on the types of shock-absorbers the group of active absorbers has to be distinguished. In contrast to semi-active devices, the response of active absorbers is connected with introduction of the energy into the system in order to counteract the excitation. As a result, inappropriate calculations of the control signals can lead to instability of the system [65]. For better understanding of the difference between semi-active and active absorbers it is worth to compare pneumatic absorbers under dynamic excitation. In case the control is performed exclusively on the valve of the absorber the problem relates to the control of system parameter and there is no additional energy transmitted to the system. In turn, in case the pneumatic absorber is equipped with compressor or inflator the reaction force can be changed rapidly. On the one hand the system response can be better than in case of semi-active control, on the other hand the additional energy is transmitted to the system. Nevertheless, active devices used as shock-absorbers can be found in scientific literature [66, 67, 68] and they are developed for industrial applications, e.g., active suspension of vehicles [69].

2.3. Control methods and techniques

According to the methodology of Adaptive Impact Absorption (AIA) [70, 25] the control strategy for mitigation of the system response under dynamic excitation assumes three consecutive actions:

- Identification of the impact excitation.
- Calculation of optimal feasible response of the system.
- Realization of the control scenario in order to obtain desired impact mitigation.

Since the germ of the AIA approach was proposed about two decades ago [71], it has been successfully applied for a number of impact absorption problems [72]. In papers A and B of this thesis the AIA approach is used, but realized by the use of innovative concepts of adaptable absorbers, which do not utilize real-time control during the impact absorption process in contrast to typical semi-active solutions. Further part of the dissertation relates to the development of alternative, more robust approach, which ensures self-adaptive performance of the system. Detailed discussion of this approach is provided in the next sections. Aiming at relevant discussion of control methods applied so far in the field of shock-absorbers and impact mitigation systems the wide literature review has been conducted and different control approaches are discussed in this section.

In reference to the AIA problems researchers have been focused on the elaboration of relevant controllable devices and determination of desired operation of the impact absorbing system under dynamic excitation. Technical solutions developed for adaptive performance of AIA structures were discussed in a number of PhD dissertations [73 26 55] and numerous patents, e.g., [20 51]. Semi-active control strategies were also discussed [74 29 55]. Nevertheless, previous works assumed that the impact excitation has to be precisely identified. Dangerous effects resulting from imprecise determination of the impact excitation are presented in paper D of this dissertation.

Articles D and E discusses control method, which allows to obtain optimal impact mitigation without full knowledge about the excitation.

Control of the semi-active absorbers is usually performed in closed loop with different types of feedback, e.g., force [75, 3] or acceleration feedback [76]. In order to utilize such systems within AIA approach the value of the absorber reaction force, or alternatively deceleration, is calculated based on the identified values of excitation parameters. Then, the system tracks the force/acceleration path until the end of absorber stroke. According to analyses presented in paper D of this thesis, selection of the particular quantity for the feedback of closed-loop system may result in different robustness of the shock-absorber operation. Paper E presents versatility of the control method, which is proposed by the author and utilizes coupling of the system kinematics and thermodynamic parameters of the fluid flow.

Control systems are often differentiated in terms of method on which their operation is based, e.g., classical, adaptive, optimal or robust controllers. Fundamentally all these methods differ from each other mainly in the goal, which is selected for system design and its tuning or optimization. In classical control theory the attention is paid to direct evaluation of the system response [77], e.g., implementation of the PID controller and assessment of the speed of system response, control error, etc. Within the optimal control problems the objective functions are minimized [78] by solving variational problems [79]. In turn, robust control techniques are aimed at ensuring system stability and predetermined response characteristics in case of uncertainty or error in the model [80]. As a result the problem of robust control concerns finding the controller which stabilizes the system and minimizes the system sensitivity to disturbances [81, 82]. In contrast the adaptive control approach is focused on the adjustment of the control law when process or system parameters change [83].

Impact mitigation systems based on the AIA approach and designed using classical methods and well known PID controller [84, 85, 86] can be applied to provide assumed reaction force scenario. In literature, some modification of the classical approach can be found, e.g., landing gear controlled by self-regulating fuzzy PD controller [49]. In the field of semi-active landing gears, also robust control has been applied, e.g., suspension system with H_∞ performance [87], in which the controller is designed in such a way that the system response under random excitation is maintained in the bounded range. Moreover, the use of optimal control theory was reported in scientific papers. In [88] discussion on the semi-active magnetorheological landing gear controlled based on minimization of the quadratic goal function is presented. In turn, [89] proposes application of Nonlinear Model Predictive Control, which is based on the optimization on the finite horizon and solved using genetic algorithm.

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3. Objectives

3.1. Goals of the research

The main goal of conducted research is to develop efficient adaptive absorbers dedicated for mitigation of loading caused by dynamic excitations, in particular of impact character. The starting point in presented dissertation is identification of insufficiencies and limitations of currently known absorbers and their controllers. The research conducted so far by other researchers has been focused on the problems with known or identified impact excitations and elaboration of smart semi-active solutions for optimal loading mitigation. As a response, this work presents adaptable semi-passive absorbers ensuring optimal response by single reconfiguration of the system according to identified excitation conditions. Simultaneously, in order to exploit the potential of currently known semi-active absorbers and to avoid problems appearing in case of imprecise impact identification the problem of not entirely known excitation is considered and the kinematics-based control method is elaborated. Such problem has not been formulated and considered before, so presented analyses possess a certain level of novelty. The final result of the thesis is improvement of the absorbers' robustness to uncertainties and variety of possible loading conditions.

Specific goals of the work are as follows:

1. Critical analyses and comparison of different operation principles of shock-absorbers under dynamic excitations, determination of their robustness to disturbances of various kind and process uncertainties, evaluation of systems' limitations and imperfections, indication of adaptive capabilities and systems classification in terms of adaptation principles – differentiation of adaptable and self-adaptive shock-absorbers.

2. Proposal of **adaptable pneumatic shock-absorber** which ensures optimal impact mitigation without real-time control – concept elaboration, design of adaptation mechanism, mathematical modelling, numerical simulation and experimental validation.

3. Elaboration of **adaptable inerter for impact mitigation** which ensures optimal deceleration of the object under impact excitation without real-time control – concept

elaboration, mathematical modelling, numerical simulation and proposal of the adaptation technique by single adjustment of the flywheel moment of inertia.

4. Elaboration of **adaptive inertial shock-absorber** and proposal of **control technique** for mitigation of the impact excitation.

5. Elaboration of control systems which ensures **self-adaptive** performance i.e., provides optimal absorption of **unknown impact**, **robustness to disturbances** and **automatic re-adaptation** of the system in case of **subsequent excitations**.

Table 1. Realization of research goals within presented papers.

		Goal of the research				
Paper no.		**1.**	**2.**	**3.**	**4.**	**5.**
	A					
	B					
	C					
	D					
	E					

3.2. Theses of the dissertation

The research presented in the dissertation was aimed at proving the following theses:

1. **Optimal deceleration** of the amortized object can **be obtained in passive manner for both the fluid-based and inertial shock-absorber**.
2. **Optimal response of the fluid-based shock-absorber** can be obtained **for different impact excitations without online control** during impact absorption process.
3. **The inertial absorber** can be used for impact mitigation and **optimal deceleration** of the amortized object can be obtained **for different excitations without online control** during impact absorption process.
4. **Successful impact absorption** can be obtained with **the use of proposed inertial absorber** which is **controlled based on** the measurements of **contact interface's deflections**.
5. **Optimal deceleration** of the object under impact excitation can be provided **without full knowledge about the excitation**.
6. Absorbers, which utilize proposed **Hybrid Prediction Control**, **provide efficient impact absorption, which is comparable with systems based on excitation identification but in contrast to them, HPC-based absorbers are robust to unknown disturbances**.
7. **Hybrid Prediction Control** allows to **sub-optimally mitigate series of subsequent impacts**, even if their parameters are unknown and excitation is bi-directional.

Table 2. Theses of the dissertation investigated within presented papers.

		Thesis of the dissertation						
		1.	**2.**	**3.**	**4.**	**5.**	**6.**	**7.**
Paper no.	**A**							
	B							
	C							
	D							
	E							

4. Overview of the results

According to presented background and conducted research the author decided that in terms of adaptation capabilities and operation principles adaptive shock-absorbers should be divided into two groups. The first group of absorbers are **adaptable** devices, the systems which **adapt to the excitation** conditions before or the beginning of the impact absorption process and they **does not adjust** their **control to additional disturbances, change of system parameters or sudden appearance of next excitations**. Systems which adapt their characteristics according to identified impact conditions belong to this group. More advanced and simultaneously more practical group of shock-absorbers is represented by **self-adaptive** devices, which **adjust their characteristics to both the external excitation and the variety of disturbances and parameters changes**.

In Fig. 2 general block diagrams of adaptable and self-adaptive impact mitigation systems are presented. In case of adaptable shock-absorber the realization of predefined control scenario can be performed in open loop as well as in closed loop. The most important feature of adaptable system is that it does not have the feedback responsible for adjustment of the impact absorption scenario. After initial phase of impact identification the system realizes assumed control strategy, e.g., maintains assumed value of reaction force, which should ensure dissipation of entire impact energy. Nevertheless, in case the process changes due to unexpected disturbances or the identification of the excitation is inexact the response of the system will no longer be optimal and it can even results in system failure. The example is hitting absorber bottom if mass of the impacting object is underestimated.

In turn, the self-adaptive system does not rely on the impact identification and it possesses the loop responsible for path-planning. Consequently, the optimal reaction force of the self-adaptive absorber is determined autonomously by the control system. The use of path-planning loop relates also to the evaluation if any changes in the process appear and the corresponding re-adaptation of system path should be done. Control system should detect that additional disturbances are present or additional energy is transmitted by next excitation. In order to obtain self-adaptive shock-absorber the path-planning loop should be designed in such way that the excitation as well as disturbances has not to be known. Analyses of the impact absorption problems allowed the author to find kinematics-based control law, which ensures robust and sub-optimal operation of the system without full knowledge about the impact excitations and possible disturbances.

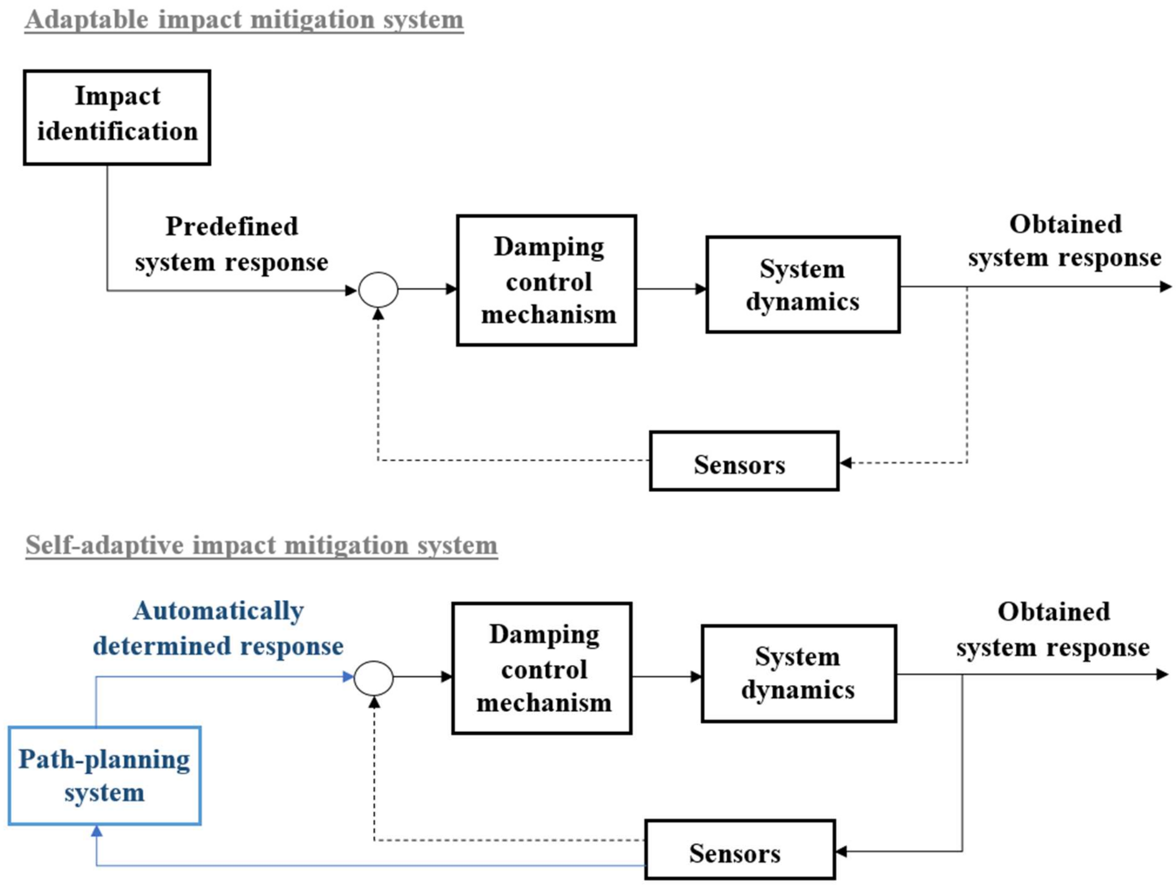


Figure 2. General schemes of adaptable and self-adaptive impact mitigation systems.

4.1. Adaptable shock-absorbers

Elaborated adaptable systems utilize **the single reconfiguration technique** for adaptation to different impact conditions. The proposed concepts are simple alternatives to the solutions based on the identification of dynamic excitation and executed by control systems operating in real time. For development of new adaptable shock-absorbers it was also assumed that the impact conditions are known, but in contrast to typical semi-active devices the adaptation is performed by single mechanical adjustment of the system and **after this action the system operates in purely passive manner**.

Semi-passive pneumatic shock-absorber

At first, the adaptable pneumatic shock-absorber was introduced in paper A, which includes detailed description of the proposed solution, derivation of the mathematical model of the absorber, discussion on the results obtained by the numerical simulations and experimental validation, which proves the concept of system operation. The construction of the device is based on two concentric cylinders (Fig. 3a): the first with narrow slots and the second with properly shaped vents, whose geometry is calculated for selected particular values of decelerated object's mass and its initial velocity. Shape of vents is determined to ensure fastest possible growth of the pneumatic force and then maintaining minimal constant level of reaction force, which will provide absorption and dissipation of entire impact energy. Relative movement of the cylinders results in compression of the gas until the time instant when slots and vents start to overlap and orifice of variable area is created. From this moment the gas release from the absorber occurs and it depends on the actual orifice area, which is a function of relative position of cylinders (Fig. 3c). At the final stage of the process the absorber reaction force has to be decreased to zero so the vent width is increased to reduce the pressure difference between absorber interior and environment. Consequently, the rebound of impacting object is avoided.

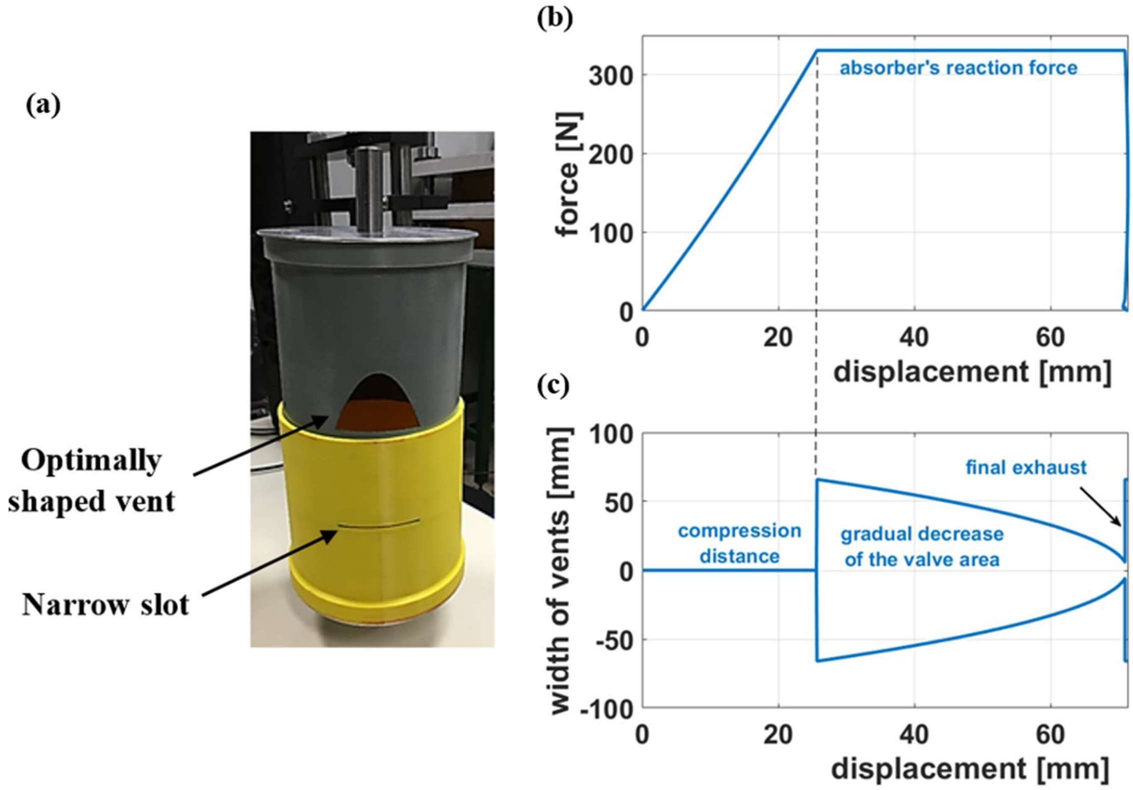


Figure 3. Proposed adaptable pneumatic shock absorber ensuring optimal impact response by overlapping slots and vents.

In order to obtain optimal performance of the system when the excitation conditions change, comparing to the values of object mass and its initial velocity for which system was manufactured, the shape of the absorber vents should be adjusted. It can be performed by the use of shutter system, whose exemplary design is presented in Fig. 4. The use of single shutter was analyzed in the paper A (Fig. 4a) as the simplified adaptation strategy and the resulting response of the system turned out to be suboptimal. The use of more complex shutter system (Fig. 4b) provides possibility to adjust the gas release more adequately and the obtained system response is closer to the optimal path. Comparison of the proposed system with typical semi-active control strategy is provided in the paper and even in case the error in modelling of the gas release process is introduced during the stage of system design, the performance of the adaptable pneumatic absorber is comparable with performance of semi-active device with fast valve controlled in real time.

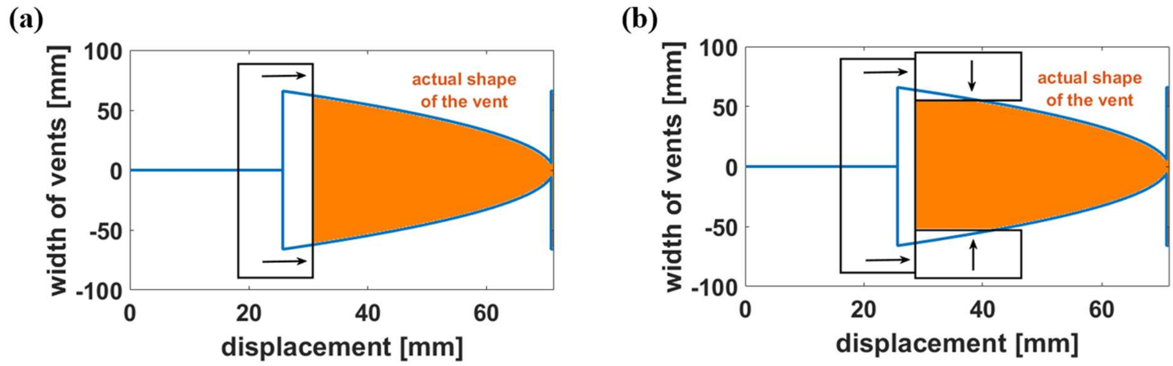
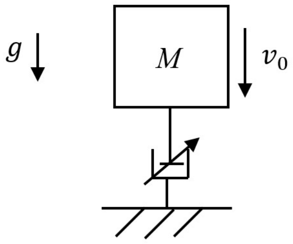


Figure 4. Proposed adaptation mechanisms based on shutters.

The most important aspects of the article A are summarized in Tab. 3., which includes considered problem formulation, assumptions for system modelling, requirements for the system and description of applied control and adaptation method. Finally, original contributions are indicated.

Table 3. Summary of the content of article A.

Adaptable pneumatic shock-absorber – paper A	
Solved problem	Find $A_v(x) \geq 0$ such that $\int_0^d F_{abs}\,\mathrm{d}x = E_{imp}(M, v_0) + Mgd$ and $\max_{x\in[0,d]} F_{abs}(x)$ is minimal, for other impact conditions (M, v_0) use mechanical adaptation mechanism to adjust $A_v(x)$. 
Assumptions for system modelling	• Cylinders of the absorber are non-deformable. • Ideal gas, adiabatic process, isentropic gas release below critical velocity due to small overpressure inside the cylinder relative to environment. • No uncontrolled leakages of the gas occur and the friction force is negligible.
Requirements for the system	• Mitigation of low energy impact with energy at the level of 20 J. • High impact absorption efficiency and possibly low value of reaction force. • Small self-weight, low manufacturing costs and high reliability.

Control and adaptation technique	• Minimization of reaction force obtained by two stage operation of the system: fastest possible growth of reaction force to the value ensuring absorption of the entire impact energy within available stroke.
	• Single reconfiguration technique is proposed:
	  ○ Calculation of the valve opening $A_v(x)$ for assumed excitation conditions in terms of the displacement x, manufacturing of determined vent shapes.
	  ○ Adjustment of the vent shape based on the identified impact conditions, it is performed at the very beginning of the process, e.g., by using shutters.
	  ○ After system reconfiguration passive operation of the device during entire impact absorption process.

Contributions	• The fluid-based system which provides optimal response in passive manner is introduced. Realization of the optimal valve opening in terms of time $A_v(t)$ is replaced by implementation of the mechanism ensuring the optimal valve opening in terms of displacement $A_v(x)$. • Mathematical model of the system is derived and applied for numerical simulations and design of the laboratory demonstrator. • The single reconfiguration technique aimed at system adaptation to different impact conditions is introduced and performance of the absorber in case of simplified adaptation strategy, i.e., only adjustment of the compression distance by the use of single shutter, is analyzed. • Results of laboratory tests prove the concept and correctness of derived mathematical model. • Proposed system is compared with typical passive absorber and semi-active solution in both the ideal design case and the case when error in modelling of the gas release is introduced.
Supplementary materials	• Appendix 1 presents mechanisms allowing for practical implementation of system adaptation based on the single reconfiguration technique. Approach is also discussed for the double-chamber pneumatic shock-absorber. • Appendix 2 is a patent description of the adaptable pneumatic shock-absorber, which reveals exemplary technical implementation of the device.

Semi passive ball-screw shock-absorber

In order to show that the single reconfiguration technique can be applied for design of various shock-absorbers, the second adaptable shock-absorber based on the ball-screw mechanism was elaborated. In contrast to pneumatic absorber, the proposed device utilizes inertial effect to ensure efficient impact mitigation. As shown in Fig. 5 the absorber is composed of the screw and the nut with the flywheel, which are mounted in the housing. Movement of the screw results in the reaction force due to conversion of the kinetic energy of translational movement into kinetic energy of rotational movement. The reaction force is transmitted from the screw to the impacting object and causes its deceleration. Properly manufactured thread of the screw with variable thread lead allows to obtain optimal deceleration of the amortized object and minimal, constant value of absorber reaction force.

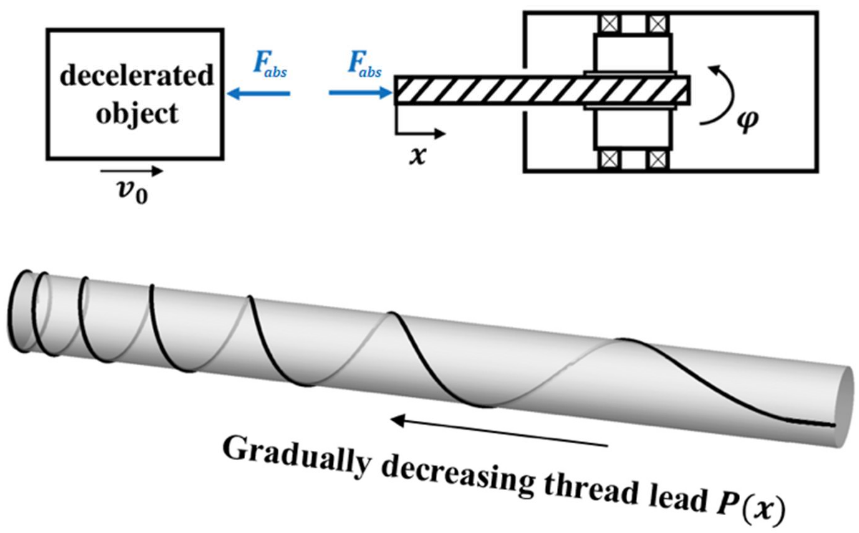
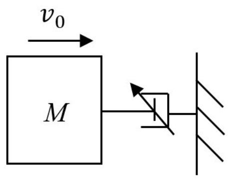


Figure 5. The concept of ball-screw inerter with variable thread lead for optimal impact mitigation.

The applied variability of the thread lead $P(x)$ results in two terms of absorber reaction force: inerter force proportional to the screw acceleration and additional damping-like force, which typically does not appear in inertial dampers, cf. Eq. (11) in paper B. As it has been proved, the optimal shape of ball-screw thread does not depend on the initial velocity of the impacting object. This result is unusual when the system is compared with other types of shock-absorbers. In general, the optimal change of control parameters influencing the system response depends on both parameters: value of the object mass M and its initial velocity v_0. Independence of the optimal thread geometry from the impact velocity provides some kind of system self-adaptation. For particular value of the object mass the obtained system deceleration will be constant and minimal. It should be highlighted that for different impact velocities the reaction force level will differ although provided by the same system configuration. According to theoretical analyses presented in the article B, adaptation to different mass values M of the impacting object is possible and it can be obtained using single reconfiguration of the system, which is realized by the change of the flywheel moment of inertia I_b before or at the very beginning of the impact absorption process.

The most important aspects of the article B are summarized in Tab. 4., which includes considered problem formulation, assumptions for system modelling, requirements for the system and description of applied control and adaptation method. Finally, original contributions are indicated.

Table 4. Summary of the content of article B.

Adaptable ball-screw inerter – paper B	
Solved problem	Find $P(x) \geq 0$ such that $\int_0^d F_{abs}\,\mathrm{d}x = E_{imp}(M, v_0)$ and $\sqrt{\frac{1}{T}\int_0^T \left(F_{abs}(t) - \frac{E_{imp}}{d}\right)^2 \mathrm{d}t}$ is minimal, for other impact conditions (M) change I_b to obtain desired $F_{abs}(P(x), I_b)$. 
Assumptions for system modelling	• Thread is parallel to the screw axis at the beginning of the absorber stroke in order to avoid initial inelastic collision, which is typically mitigated by the use of additional contact interface. • Thread elasticity and backlash are negligible. • The friction in bearings is negligible but friction on the thread connection is considered.
Requirements for the system	• Impact mitigation with minimal, constant value of absorber reaction force – 100% impact absorption efficiency. • No need for equipping the system with additional contact interface. • Passive operation and possibility to implement adaptation mechanism based on the single reconfiguration technique.

Control and adaptation technique	• Minimization of the reaction force by minimization of the root mean square of the deviation of absorber reaction force from globally optimal value, which fulfills the condition of entire impact energy absorption.
	• Single reconfiguration technique is proposed:
	  o Calculation of the thread lead $P(x)$ for assumed excitation conditions, defined by impacting object mass M, in terms of the displacement x by the use of inverse dynamics approach applied at the stage of system design and manufacturing.
	  o Adjustment of the flywheel moment of inertia I_b based on the identified impact value of impacting object mass M, it is performed at the very beginning of the process.
	• After system reconfiguration passive operation of the device during entire impact absorption process.

Contributions	• The inertial system which provides optimal response in passive manner is introduced. • Mathematical model of the system is derived and applied for theoretical analyses and conducting numerical simulations. • Methodology of system design is described in detail and possibility of absorber adaptation to impact excitations is indicated. • Passive system is optimized for a range of impact conditions – different values of mass M of the decelerated object.
Supplementary materials	• Appendix 3. is a patent description of the ball-screw inerter with variable thread lead, it reveals exemplary technical implementation of the device.

4.2. Adaptive inertial shock-absorber

Although the concept of adaptive inertial shock-absorber presented in paper C was elaborated as the first solution from the set of devices presented in this dissertation, it should be placed between adaptable solutions discussed in previous section and self-adaptive absorbers discussed in the next section. Let me remind, adaptable systems which are proposed within the thesis are based on the single reconfiguration of the system and then passive operation of the system during entire impact absorption process. In turn, self-adaptive absorbers, which are discussed later, are fully controlled but allows to avoid identification of the excitation and adapt to disturbances appearing during the process.

The adaptive inertial shock-absorber (Fig. 6.) is composed of the rod with threaded sleeve and three threaded rings. Connections between elements are indicated in the figure as interfaces 1, 2 and 3. Threads on interface 1 and 3 are opposite-oriented and interface 2, which is placed between inner and middle ring, is a dissipative connection, e.g., viscous. The amortized object is decelerated due to elastic contact with the screw which transfer impact energy into energy of rotating parts. The proposed operation of the device assumes two phases: the first when impact energy is accumulated in outer ring and the second when the inner ring is driven by the screw in opposite direction. As a result on the interface 2 a significant difference in relative velocities is obtained (Fig. 7.) and consequently, the efficient damping of components movement is realized. According to the case study discussed in the paper C the control performed on the absorber can be based on the measurements of the contact interface deflections. For adaptation of the system to different impact conditions moment of the outer ring inertia and time instant of the first phase termination and simultaneously the start of the second phase can be adjusted.

The most important aspects of the article C are summarized in Tab. 5., which includes considered problem formulation, assumptions for system modelling, requirements for the system and description of applied control and adaptation method. Finally, original contributions are indicated.

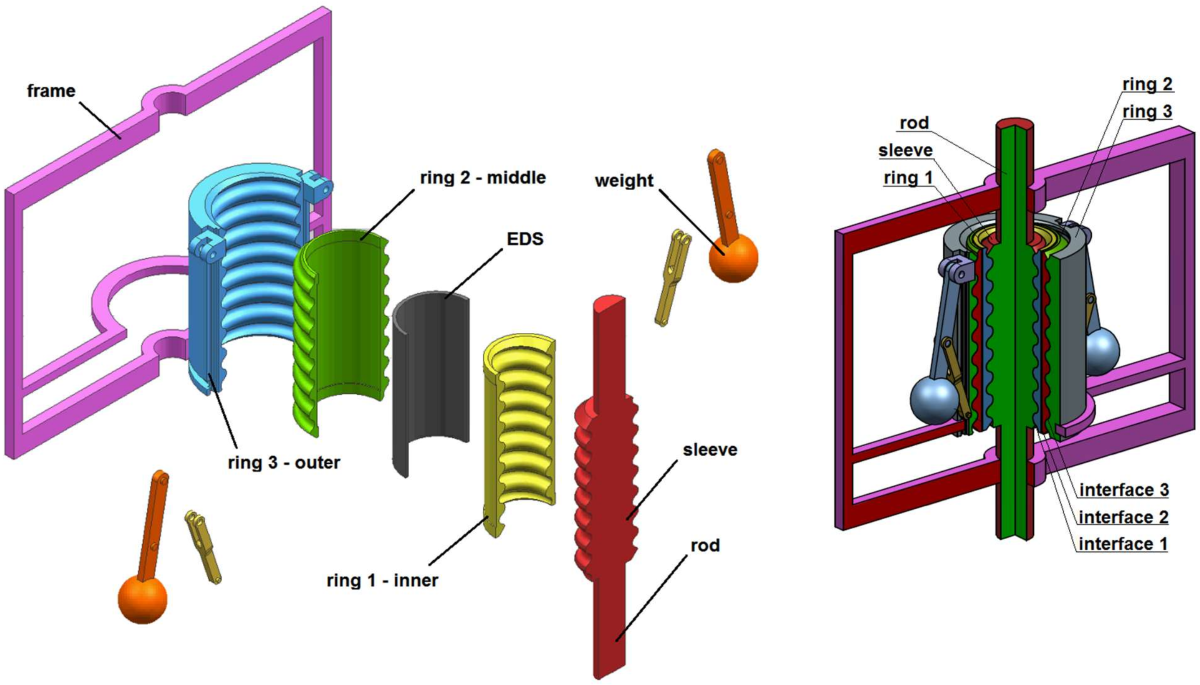


Figure 6. Components of the adaptive inertial shock-absorber.

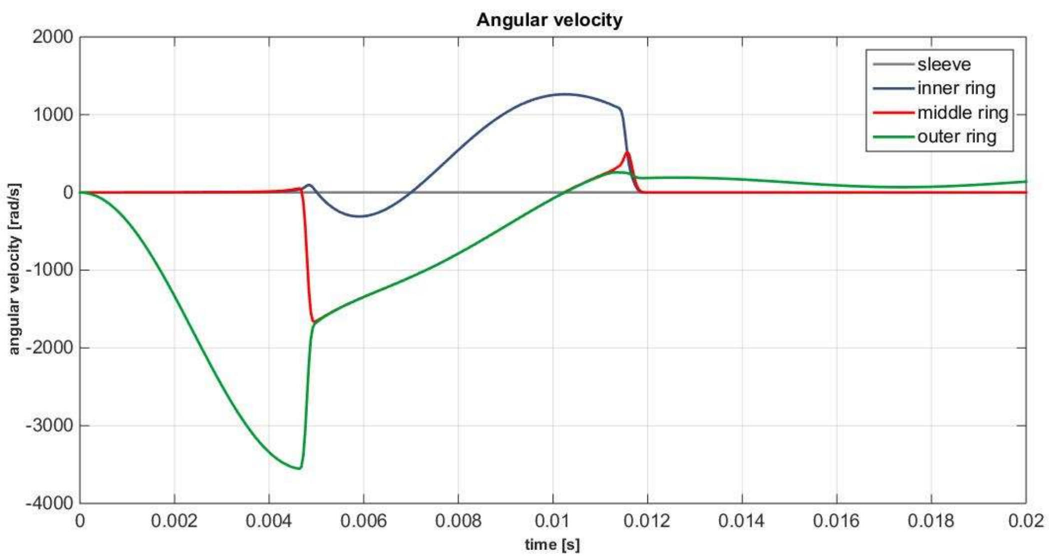
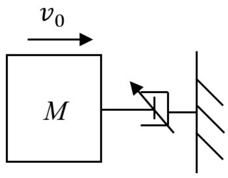


Figure 7. Angular velocity of inertial rings during realization of exemplary control strategy.

Table 5. Summary of the content of article C.

Adaptive inertial shock-absorber – paper C	
Solved problem	Find *combination of 'on/off' actions* performed on blockers between interfaces such that $\int_0^d F_{abs}\,\mathrm{d}x = E_{imp}$ and $\max_{t\in[0,T]} F_{abs}(t)$ is minimal 
Assumptions for system modelling	• Inertial system composed of three rings with two opposite-oriented threads. • Viscous interface between inner and middle ring. • Every connection between rings can be blocked and released in 0.2 ms. • Friction forces are negligible.
Requirements for the system	• Mitigation of the impact excitation and dissipation of entire impact energy. • Possibility to implement different semi-active control strategies.
Control and adaptation technique	• The kinetic energy of the amortized object is converted into energy of rotating parts by the change of translational movement of the screw into rotational movement of threaded rings. • In first phase of system operation the certain part of impact energy is accumulated as the kinetic energy of outer inertial ring. • Initiation of the second phase: when deflection of the contact interface decrease to zero the outer ring is combined with the middle ring and simultaneously the threaded connection between screw and inner ring is released. • In the second phase of system operation the outer inertial ring rotates together with middle ring and simultaneously, the inner ring is driven by the screw in the opposite direction. As a result, a significant difference between velocities of inner and middle rings appear and efficient dissipation of impact energy is provided. • The successful impact absorption can be performed based on the measurements of the contact interface deflections, which are calculated for identified impact conditions and absorber parameters.
Contributions	• The concept of controllable inertial shock-absorber is proposed and applied for the problem of impact absorption. • The model of system is implemented using multibody dynamics. • Exemplary control strategy ensuring successful impact mitigation is proposed and verified numerically. • Influence of different parameters on the system operation is analyzed. • Possibility of system adaptation is investigated and exemplary retuning of the system to other impact conditions is presented.
Supplementary materials	• Appendix 4 is a patent description of the adaptive inertial shock-absorber and it reveals exemplary technical implementation of the device.

4.3. Control systems for self-adaptive performance of the shock-absorber

The final proposal of this thesis is the novel control method, which was introduced in paper D and was further developed in paper E. The method has been named the Hybrid Prediction Control (HPC), what relates to two types of prediction performed during impact absorption process. The first is the Control Mode Prediction (CMP), which is based exclusively on the measurements of the working element kinematics and simultaneously eliminates necessity of identification of the object mass M or impact force F_{ext}. It is achieved by reformulation of the problem (4) to the state-dependent path-tracking:

$$\text{Find } u(t) \geq 0 \text{ such that } \int_d \overrightarrow{F_{abs}} \mathrm{d}\vec{s} = E_{imp} \text{ and } \int_0^T \left(F_{abs}(u(t), t) - F_{abs}^{opt}(t)\right)^2 \mathrm{d}t \text{ is minimal.} \quad (6)$$

Further derivation presented in paper E reveals the final form of the solved path-tracking problem:

$$\text{Find } u(t) \geq 0 \text{ such that } \int_d \overrightarrow{F_{abs}} \mathrm{d}\vec{s} = E_{imp}$$

$$\text{and } \int_{t_i}^{t_{i+1}} \left(F(u(t)) - F(t_i) - M\left(\frac{\dot{x}(t_i)^2}{2(d - x(t_i))} + \ddot{x}(t_i)\right)\right)^2 \mathrm{d}t \text{ is minimal,} \quad (7)$$

where: F is the absorber reaction force without internal disturbances, e.g., pneumatic force of pneumatic damper, viscous force of hydraulic damper or inertia force in case the inerter is considered.

The second type of prediction included in HPC is the Inverse Dynamics Prediction (IDP), which allows to maintain constant, optimal level of reaction force until the end of absorber stroke. As shown in Fig. 8. the CMP is the master loop, which decides if the absorber reaction force is below (Mode 1), above (Mode 3) or at the actually appropriate level (Mode 2) ensuring deceleration of the piston to zero and dissipation of entire impact energy. In case the Mode 2 is detected the IDP is activated in order to provide continuous control of the system parameter, which in case of pneumatic shock-absorber is the valve opening area. Such approach allows to avoid identification of the impacting object mass or external impulsive force and guarantees robust performance of the shock-absorber. Moreover, according to results presented in paper E system is able to sub-optimally mitigate the series of impacts, even if they are bi-directional. It should be highlighted that reception of repetitive impacts was not considered in the field of Adaptive Impact Absorption.

Simultaneously, the HPC method can be successfully applied for different types of absorbers. Moreover, it can be extended to the problems when kinematic condition implemented for Control Mode Prediction is represented by other types of inequalities imposed on the working element acceleration and other kinematically defined paths that are used for Inverse Dynamics Prediction.

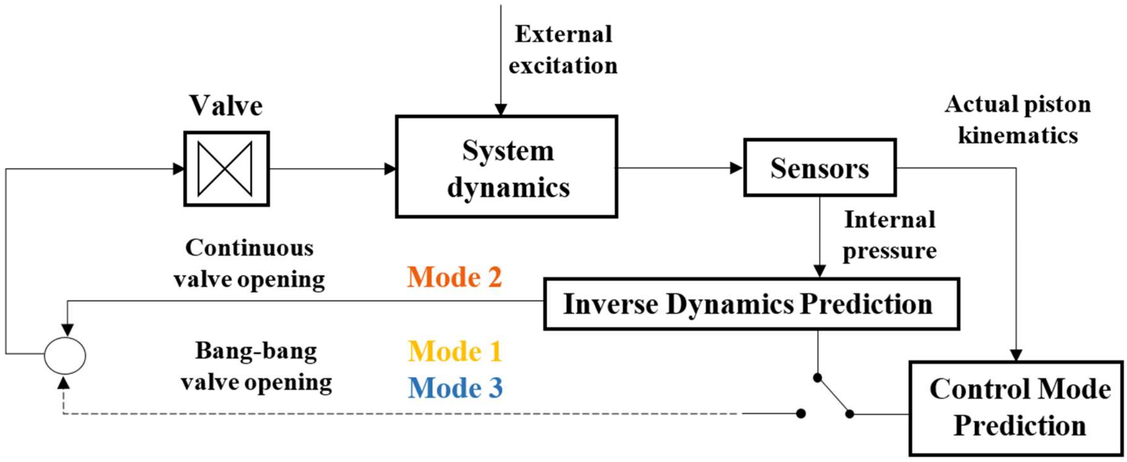


Figure 8. Block diagram of the control system for pneumatic shock-absorber based on the Hybrid

The most important aspects of the article D and E are summarized in Tab. 6 and 7, respectively. They include considered problem formulations, assumptions for system modelling, requirements for the system and description of applied control and adaptation method. Finally, original contributions are indicated.

Table 6. Summary of the content of article D.

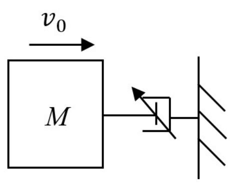
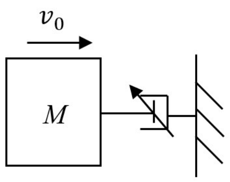
Kinematic feedback control for self-adaptive performance of shock-absorbers – paper D	
Solved problem	Find feasible change of reaction force $F_{feas}^{opt}(x)$ such that $\int_0^d F_{feas}(x)\mathrm{d}x = E_{imp}(M, v_0)$ and $\max_{t\in[0,T]} F_{feas}(x(t))$ is minimal, then: Find $A_v(t) \geq 0$ such that $\int_0^d F_{abs}\big(A_v(t)\big)\mathrm{d}x = E_{imp}(M, v_0)$ and $\int_0^T \Big(F_{abs}(A_v(t)) - F_{feas}^{opt}(x(t))\Big)^2 \mathrm{d}t$ is minimal. 
Assumptions for system modelling	• Single-chamber pneumatic absorber with outflow of gas to environment. • Non-deformable absorber walls. • Ideal gas, adiabatic process, isentropic model of gas outflow. • The friction force is negligible.
Requirements for the system	• Absorption and dissipation of the entire impact energy. • Minimization of the dynamic loading under impact excitation. • Robustness to force disturbances and gas leakages.
Control and adaptation technique	• Replacement of the standard AIA approach, which is based on identification or prediction of the excitation, calculation of the optimal feasible change of reaction force and calculation of control using inverse dynamics approach, by the kinematic feedback responsible for adaptive path planning. • Operation of the system divided into three processes: path finding, path tracking and path update; various possible implementations of the path tracking process are considered in the paper.
Contributions	• Assessment of the influence of imprecise impact identification on the performance of standard Adaptive Impact Absorption systems. • Assessment of the influence of disturbances (additional unknown forces, sudden gas leakage) on the performance of standard Adaptive Impact Absorption systems. • Proposal of control method based on kinematic optimality condition and adaptive path planning, and analyses of three variants of this method. • Proof of self-adaptive performance of the proposed system – no need for identification of the mass of impacting object and high robustness to disturbances of different kind. • Proposal of algorithm for real-time implementation of the control method.
Supplementary materials	• Appendix 5 discusses the influence of size of tolerance interval for kinematic condition and frequency of control recalculation, the adaptive tolerance is proposed and response of the system in case of harmonic excitation is examined.

Table 7. Summary of the content of article E.

Hybrid Prediction Control for self-adaptive performance of shock-absorbers – paper E	
Solved problem	Without knowledge about M and $F_{ext}(t)$: find $A_v(t) \geq 0$ such that $\int_d \overrightarrow{F_{abs}} \mathrm{d}\vec{s} = E_{imp}(M, v_0, F_{ext}(t))$ and $\int_{t_i}^{t_{i+1}} \left(F_{pneu}(A_v(t)) - F_{pneu}(t_i) - M \left(\frac{\dot{x}(t_i)^2}{2(d - x(t_i))} + \ddot{x}(t_i) \right) \right)^2 \mathrm{dt}$ is minimal. 
Assumptions for system modelling	• Double-chamber pneumatic absorber with controlled gas transfer between chambers. • Non-deformable absorber walls. • Ideal gas, adiabatic process, isentropic model of transfer between chambers. • The friction force is negligible. • Impact excitation defined by initial velocity v_0 or external force $F_{ext}(t)$ in a form of short-lasting, high value impulses.
Requirements for the system	• Absorption and dissipation of the entire impact energy without knowledge about the mass of impacting object M and. • Minimization of the dynamic loading under impact excitation. • Robustness to force disturbances and gas leakages. • Automatic re-adaptation of the system under series of impacts. • Successful mitigation of bi-directional excitations.
Control and adaptation technique	• Replacement of the standard AIA approach, which is based on identification or prediction of the excitation, calculation of the optimal feasible change of reaction force and calculation of control using inverse dynamics approach, by the Hybrid Prediction Control. • Operation of the system based on two types of prediction: Control Mode Prediction for finding the actually optimal path and Inverse Dynamics Prediction for maintaining optimal constant level of reaction force by using the flow model and kinematic quantities.
Contributions	• Further development of kinematic feedback control is presented, unknown impact mitigation problem is formulated and derivation of the Hybrid Prediction Control is shown. • Self-adaptive performance and robustness of the system to disturbances of various kind are demonstrated. • Analyses of system performance in case of series of impacts, including bi-directional excitations, are conducted and confirm its automatic re-adaptation capabilities. • System operation is analyzed also in case of ‘on-off’ type of the valve. • System responses are compared with passive, adaptive and optimal control methods.

4.4. Original achievements and directions of further research

The most important scientific achievements obtained within the thesis include:

- Proposal of original classification of shock-absorbers in terms of adaptive capabilities resulting from specific control methods.
- Elaboration and development of the adaptation technique based on single mechanical reconfiguration of the system:
    - Development of the concepts of adaptable pneumatic shock-absorber and adaptable inerter with variable thread lead for impact mitigation.
    - Mathematical modelling and numerical simulation of the proposed absorbers;
    - Determination of shock-absorbers' parameters, which provides optimal impact mitigation.
    - Design and experimental validation of the pneumatic absorber concept.
    - Demonstration that the optimal impact mitigation provided by both pneumatic and inertial absorber can be obtained without online control during impact mitigation process.
- Elaboration and development of the control systems ensuring self-adaptive performance of the shock-absorbers:
    - Elaboration of the adaptive inertial shock-absorber and development of control method based on the measurements of the contact interface deflections.
    - Development of Hybrid Prediction Control method, which enables solving the state-dependent path tracking method without knowledge of the excitation, implementation on the method using illustrative example of fluid-based absorbers.
    - Analysis of the robustness of the proposed HPC method in case of unknown disturbances of various kinds and subsequent excitations.
    - Comparison of the proposed method with adaptive and optimal control solutions.

- Proposal of the real-time implementation of the proposed control method.

Plans of further research primarily include practical implementation of the proposed control methods and extension of proposed concepts for the use in multi-dimensional systems. Another field of planned research will be adjustment of control methods to the vibration mitigation problems.

5. Original articles

Article A

Article

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Adaptable pneumatic shock absorber

Rami Faraj, Cezary Graczykowski and Jan Holnicki-Szulc

Abstract

Recent progress in the field of sensors, actuators, and smart materials allows the construction of more and more efficient controllable pneumatic dampers for shock absorption. Typically, such devices apply online semi-active control techniques, which utilize electromagnetic, piezoelectric, or magnetostrictive valves. As a result, they are characterized by a high efficiency of impact absorption, but simultaneously by a complicated construction and a specialized electronic system. The alternative solutions are semi-passive absorbers that ensure a similar performance by using a much simpler, low-cost construction and a less complicated adaptation mechanism. This paper introduces an adaptable semi-passive single-chamber pneumatic shock absorber, SOFT-DROP, which provides the optimal impact absorption and energy dissipation after a single reconfiguration performed at the beginning of the process. The high effectiveness of the proposed concept is proved in numerical and experimental investigations of the device. Moreover, the proposed semi-passive damper is also compared against already known pneumatic absorbers that utilize semi-active control methods. Ultimately, the device might be used in an airdrop system for delivery of light packages.

Keywords

Adaptable, semi-passive, impact absorption, optimal design, pneumatic shock absorber

1. Introduction

Pneumatic shock absorbers of diverse designs are successfully applied as energy-dissipating devices in various branches of engineering. The most basic class of such devices are passive pneumatic dampers, in which neither the internal pressure nor the size of the release orifice can be modified. As a consequence, they cannot adapt to the actual impact excitation and they are designed for a typical impact condition. As a result, their efficiency significantly decreases when the impact loading substantially changes.

The other group of pneumatic shock absorbers are semi-active dampers controlled in real time. They usually utilize fast-operating controllable valves based on advanced functional materials. Semi-active devices have the ability to adapt to the actual impact loading through internal pressure control; thus, their average efficiency substantially exceeds the efficiency of passive pneumatic shock absorbers. Semi-active shock-absorbing dampers have been considered in several patents and research papers. In particular, the concepts of double-chamber pneumatic shock absorbers, which exploit the idea of variable or controlled flow connection, were introduced in early patents (Duran, 1977; Hiramoto et al., 1999). Absorbers with controlled gas release to the environment and with controlled flow between chambers (Khamitov et al., 2009a, 2009b) were considered for efficient damping of vibrations, as well as for dynamic response mitigation in seismic conditions (Khamitov et al., 2008). Moreover, the influence of an additional external chamber on the operation of the air spring applied as a shock absorber in heavy vehicles was investigated by Roebuck et al. (2010). Further research concerned double-chamber adaptive pneumatic absorbers equipped with an online controlled piezoelectric valve. Their effectiveness was proved experimentally and application in a landing gear of small airplane was considered (Mikułowski et al., 2009).

Mathematical modeling of pneumatic absorbers was described by Barber (1997), Graczykowski (2011), and Mikułowski and Wiszowaty (2016). Moreover, various

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types of adaptive pneumatic structure were modeled by Graczykowski (2016). Models of pneumatic vibration isolators in the frequency domain were applied by Erin and Wilson (1998) and Lee and Kim (2007). Furthermore, control procedures were developed in order to obtain the desired characteristics of the pneumatic actuators. The development of a feedback controller with saturation was presented by Ilchmann et al. (2006), whereas an unconventional acceleration-feedback control system with time-delay minimization was applied by Wang et al. (1999). The development of an efficient control strategy aimed at saving energy submitted to a pneumatic actuator by application of an inter-chamber valve was proposed by Shen and Goldfarb (2007). The design of a robust L_2 gain state derivative feedback controller for an active suspension system of a vehicle was described by Yazici and Sever (2017a), whereas the application of a robust linear quadratic regulator for active control of a landing gear system equipped with an oleo-pneumatic shock absorber was proposed and compared against other control methods by Yazici and Sever (2017b).

Although the theoretical efficiency of semi-active pneumatic shock absorbers is relatively high, their adaptability is restricted by the speed of controllable valve operation. Moreover, such devices are characterized by a complex construction, which makes them expensive, prone to damage, and insufficiently reliable in impact conditions. Thus, when the excitation conditions are identified, the semi-active pneumatic absorbers can be replaced by semi-passive absorbers. Such pneumatic absorbers utilize a single control action and afterward operate in the passive manner. They have the same favorable mechanical characteristics as semi-active absorbers, but they use a less complicated and thus more reliable construction.

This paper presents detailed numerical and experimental investigations of the semi-passive pneumatic shock absorber, SOFT-DROP, which differs significantly from the adaptive pneumatic shock absorbers presented in previous research works. Although its operation is still based on preliminary impact identification and determination of the optimal system path, the path-tracking process is executed in an entirely different manner. The optimal outflow of the gas is not executed by a real-time valve control, which is replaced by a single control action aimed at proper shaping of the outflow vents. After system reconfiguration, the absorber operates in a purely passive manner and provides dissipation of the impact energy with the minimal level of deceleration. Therefore, the application of complicated fast-operating valves is totally eliminated. The proposed system achieves an efficiency similar to semi-active systems, but is much more reliable, owing to its simplified construction.

2. Adaptable pneumatic shock absorber

2.1. Motivation and requirements

The SOFT-DROP shock absorber was invented as a suspension for an airdrop system designed for the fast delivery of small packages. Let us assume that the entire airdrop system has a total mass of 5 kg and that the touchdown velocity reaches a value of 2.5 m/s. The following results will refer to these impact conditions. General technical requirements for the shock-absorbing system can be formulated as follows:

- Mitigation of low energy impacts—dissipated energy at a level of 20 J;
- At least 80% impact absorption efficiency and small absorber reaction force;
- Fast system reconfiguration before cargo touchdown;
- High reliability during harsh impact conditions;
- Small self-weight of shock-absorbing system;
- Low cost of device manufacture.

2.2. General concept of adaptive impact energy absorption

The process of adaptive impact absorption (Holnicki-Szulc, 2008) is aimed at impact energy dissipation and ensuring the smallest-possible generated reaction force. The process consists of three main stages. The first stage is impact detection and identification, which includes a recognition of the mass and velocity of the impacting object (Sekuła et al., 2013). The second stage is the determination of the optimal control scenario for the reaction force and the corresponding kinematics of the system. Finally, the third stage is the realization of the previously determined adaptation scenario. In pneumatic absorbers, the minimization of generated force is provided by an initially closed valve, resulting in the fastest possible increase of internal pressure and then maintaining a possibly constant reaction force. At the end of the absorber stroke, the valve is fully opened and the force is suddenly reduced. Successive stages of the process and the typical corresponding change of the generated force are presented in Figure 1.

2.3. Concept of an adaptable device based on a single reconfiguration technique

The proposed SOFT-DROP shock absorber is composed of two cylinders: an outer cylinder, including narrow slots, and an inner cylinder, including corresponding vents of a variable shape. The SOFT-DROP follows the general concept of adaptive impact absorption without using a release valve controlled in real time. The characteristic feature of the device is that the valve is formed by overlapping slots and vents. The general scheme of the SOFT-DROP shock absorber's construction, together with the release valve area, ensuring optimal impact energy dissipation, is presented in Figure 2. When the top cylinder moves downwards, the gas inside the absorber is compressed until the lower edges of the top cylinder vents and the upper edges of the lower cylinder slots start overlapping. The overlap area forms a valve, through which the gas is released. Such a strategy allows achieve the fastest possible increase of absorber reaction force to be achieved through the use of the pneumatic spring effect and then a constant value of the generated force to be maintained until the end of the absorber stroke. To prevent a rebound of the top cylinder, at the end of the available stroke, the gas must be finally exhausted to reduce the overpressure inside the chamber.

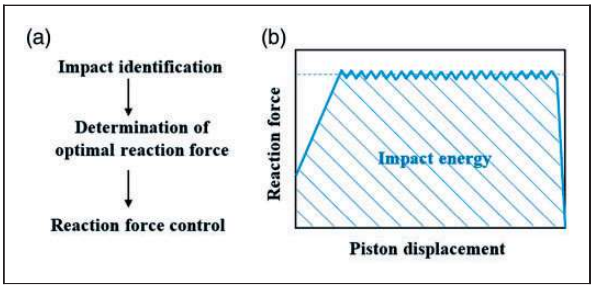

Figure 1. (a) General operation scheme of adaptive impact-absorbing system, (b) Typical reaction force scenario in a pneumatic impact-absorbing system.

The compression distance and the shape of the vents have to be selected precisely and reconfigured for each identified excitation condition. Adaptation to the impact excitation is performed by a single reconfiguration of the system, including a change in the initial relative position of the cylinders and a change in the vent shape. As a result, control actions are required only at the beginning of the shock-absorption process. After system adaptation, SOFT-DROP operates as a passive device.

According to the final planned application of the device, we decided to apply air under atmospheric pressure as the operational gas inside the shock absorber. On the one hand, this slightly decreases the ability of impact energy dissipation, but on the other hand it eliminates the important problem of gas leakages and the requirement of a precise sealing of the absorber.

To demonstrate the attractiveness of the proposed solution, a comparison of the SOFT-DROP device with an optimally designed passive pneumatic shock absorber with a constant valve opening is shown in Figure 3, which shows graphs of the absorber reaction force as a function of time and as a function of the displacement of the top cylinder. The assumed values of the most important design parameters for the exemplary SOFT-DROP shock absorber are collected in Table 1.

In the presented example, the efficiency of the optimally designed typical passive absorber is less than 63% (typically, it is even lower (Currey, 1988)), whereas the SOFT-DROP absorber with the same dimensions achieves an efficiency of 81%. In reference to the pneumatic actuators controlled in real time, a higher

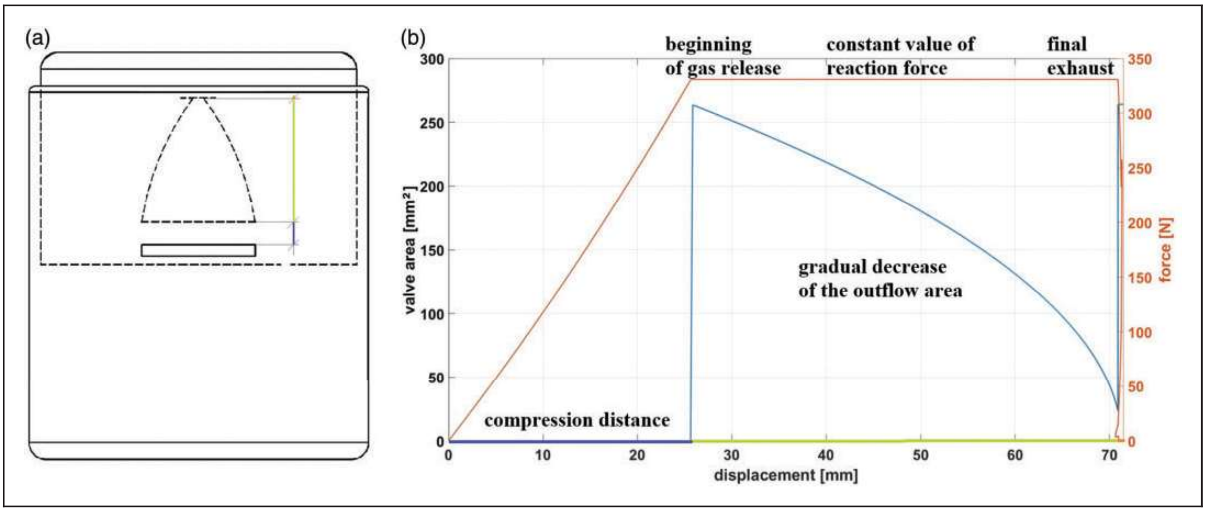

Figure 2. Characteristic features of the SOFT-DROP system: (a) construction scheme; and (b) valve area and force response in reference to three phases of absorber operation.

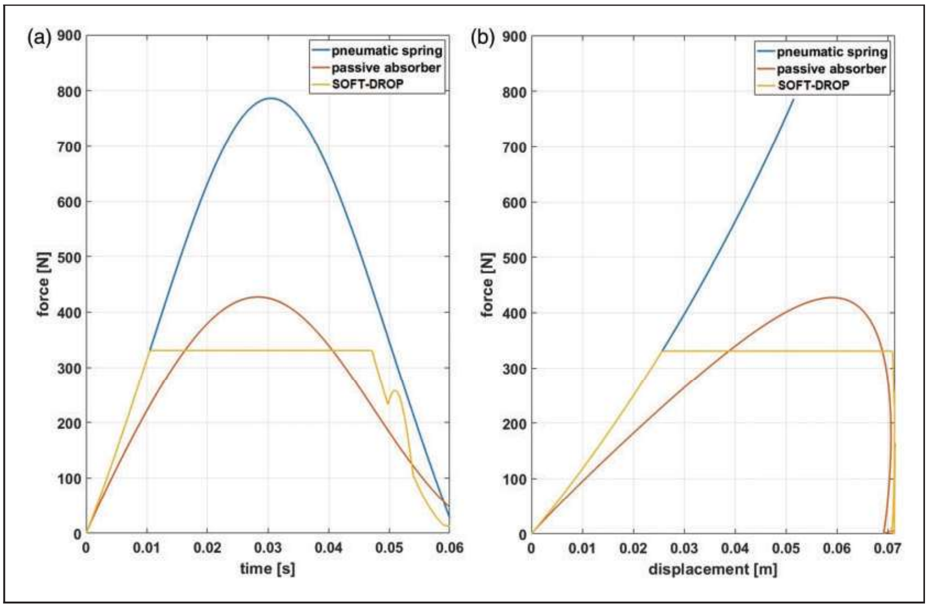

Figure 3. Reaction force as a function of (a) time and (b) displacement for pneumatic spring, passive pneumatic absorber, and the SOFT-DROP device.

Table 1. Exemplary design parameters of the SOFT-DROP shock absorber.

Initial internal pressure (kPa)	Operational gas	Absorber diameter (m)	Available stroke (m)	Suspended mass (kg)	Initial velocity of the mass (m/s)
101.3	Air	0.15	0.0725	5	2.5

efficiency can be achieved only if an initial overpressure in the system is applied.

The system should be reconfigured before the dissipation of the impact energy; this can be performed with the use of miscellaneous techniques. To adjust the compression distance, the initial relative position of the cylinders has to be changed. This can be achieved using a small ratchet system equipped with a servo-mechanism or light linear actuator. The next action that has to be performed in order to obtain the optimal response of the system is re-shaping of the top cylinder vents. This can also be realized using a mechanical ratchet system, allowing for the shape change of each vent. Conversely, the SOFT-DROP reconfiguration can also be ensured by manufacturing a set of different vents, which are selected by means of a relative rotation of the cylinders around their axis or by covering vents that are inappropriate for the actual impact conditions. For a fast and reliable reconfiguration of the system, both the compression distance and the re-shaping of the vents can be coupled and conducted concurrently.

The single reconfiguration technique introduced here allows a very efficient force response to be obtained in a wide range of different impact conditions and provides a performance similar to the performance of the fully controlled pneumatic absorbers, utilizing real-time control.

3. Optimal reconfiguration of the absorber

The concept introduced in the previous section allows an optimal change in the generated reaction force to be obtained using a single reconfiguration of the system at the very beginning of the impact absorption process. The assumed strategy of force control involves a first stage, when the force increases to an appropriately selected value, then a second stage, when the force is maintained at a constant level for a large part of the cylinder stroke, and a final stage, when the force is reduced. Such a strategy ensures absorption and dissipation of the entire impact energy with the minimal level of the force exerted on the landing object. Consequently, the minimal value of the object deceleration is provided and its rebounds are avoided.

In this way, the control of the generated force is obtained by a proper release of the compressed gas through a valve created by the overlapping slots and vents. Therefore, the main task related to the design of the adaptable absorber is to determine the proper locations and shapes of the vents for the assumed set of impact conditions. In this section, the design procedure is briefly presented by the use of a simple mathematical model of the adaptable pneumatic cylinder, in which the impact excitation is modeled by the mass M with initial velocity V_0. We briefly recall the model of the pneumatic absorber, propose a method to determine the optimal actual size of the outflow area $A_{\mathrm{valve}}(t)$ by using the inverse dynamics approach, and finally prove that such an optimal change can be achieved by overlapping narrow rectangular slots and properly shaped vents.

Further discussed experimental results and numerical simulations indicate that the construction and operation principles of SOFT-DROP allow the use of a relatively simple mathematical model for design and optimization purposes. Let us introduce the following simplifying assumptions:

- Both cylinders are nondeformable.
- No uncontrollable leakages of the gas occur.
- The friction force in the system is negligible.
- Idealized conditions of object contact with the ground are assumed (a one-degree-of-freedom model is applied).

The SOFT-DROP device is modeled using the equation of motion of the top cylinder (equation (1)) and the thermodynamic equation of energy balance, which involves the internal energy of gas, the work done by gas, and the change in enthalpy (equation (2)). The mass flow rate of gas through the valve $\dot{m}(t)$ is described by the model of isentropic flow (equation (3)). The operational gas is modeled by the ideal gas law (equation (4)), combining gas pressure $p(t)$, its temperature $T(t)$ and volume $V(t)$

$$M\ddot{u}(t) - Mg + (p(t) - p_{\mathrm{out}}(t))A = 0, \quad \dot{u}(0) = V_0 \tag{1}$$

$$\dot{m}(t)c_v T(t) + m(t)c_v \dot{T}(t) + p(t)\dot{V}(t) = \begin{cases} \dot{m}(t)c_p T_{\mathrm{out}}(t) & \text{for } p(t) \le p_{\mathrm{out}}(t) \\ \dot{m}(t)c_p T(t) & \text{for } p(t) > p_{\mathrm{out}}(t) \end{cases} \tag{2}$$

$$\dot{m}(t) = \begin{cases} A_{\mathrm{valve}}(t)C_{\mathrm{dis}}\sqrt{2}\sqrt{\dfrac{k\cdot\left(q^{\frac{2}{k}} - q^{\frac{k+1}{k}}\right)}{k-1}}\dfrac{p_{\mathrm{out}}(t)}{\sqrt{RT_{\mathrm{out}}}} & \text{for } p(t) \le p_{\mathrm{out}}(t) \\ -A_{\mathrm{valve}}(t)C_{\mathrm{dis}}\sqrt{2}\sqrt{\dfrac{k\cdot\left(q^{\frac{2}{k}} - q^{\frac{k+1}{k}}\right)}{k-1}}\dfrac{p(t)}{\sqrt{RT(t)}} & \text{for } p(t) > p_{\mathrm{out}}(t) \end{cases} \tag{3}$$

$$p(t)V(t) = m(t)RT(t). \tag{4}$$

In the equation of the isentropic flow, the variable q indicates the ratio of pressure inside the cylinder $p(t)$ to the external pressure $p_{\mathrm{out}}(t)$, which does not exceed the threshold at which a chocked flow occurs. If the outflow area $A_{\mathrm{valve}}(t)$ and the discharge coefficient C_{dis} are known, this model allows the response of the absorber to be simulated. The discharge coefficient is typically determined by experimental testing and indicates the difference between the theoretical and real mass flow rates, resulting from viscosity, turbulence, etc. Here, the discharge coefficient is assumed as equal to one.

In the design process, we determine the change of the outflow area, which minimizes the force generated by the absorber. The solution of the corresponding dynamic optimization problem indicates that the internal pressure and generated force must initially increase at the highest possible rate, then they should remain constant, and finally they should be reduced to zero. Moreover, the change in generated force should provide absorption and dissipation of the entire impact energy within the entire available stroke of the absorber. The corresponding mathematical model results from integration of the equation of cylinder motion, separately for the first stage of the process (displacement from 0 to u_x), when adiabatic compression of gas is assumed, and for the second stage of the process (displacement from u_x to $u_{\max}$), when a constant reaction force is generated

$$\begin{aligned} \frac{1}{2}MV_0^2 + Mgu_{\max} = {} & p_0 A h_t^k (h_t - u_x)^{\frac{-k+1}{k-1}} - p_0 A\frac{h_t}{k-1} \\ & - p_{\mathrm{out}}Au_x + \left(p_0\frac{h_t^k}{(h_t-u_x)^k}A\right) \\ & \times (u_{\max} - u_x) - p_{\mathrm{out}}A(u_{\max} - u_x) \end{aligned} \tag{5}$$

The solution of this equation allows determination of the displacement u_x and the corresponding gas parameters (pressure p_x and the temperature T_x), which indicate the limit between the two stages of the process. To determine the optimal shape of the outflow area, the so-called inverse dynamics approach is applied. In this method, we assume both the value of the force generated by the absorber and the corresponding kinematics of the system in order to determine the required mass flow rate of the gas $\dot{m}(t)$ and the opening of the valve $A_{\mathrm{valve}}(t)$. The mass of the gas $m(t)$ is calculated from the equation of energy balance (equation (2)) combined with the ideal gas law (equation (4)), yielding

$$\frac{p(t)V(t)^k}{m(t)^k} = \text{const.} \tag{6}$$

During the second stage of the process, the internal pressure must remain constant; hence, the mass of the gas $m(t)$ is proportional to the actual volume of the gas $V(t)$

$$m(t) = m_0 \frac{V(t)}{V_x} = m_0 \frac{u_{\max} - u(t)}{u_{\max} - u_x} \tag{7}$$

Consequently, the mass flow rate of the gas can be expressed in terms of the actual velocity of the cylinder $v(t)$

$$\dot{m}(t) = -\frac{m_0}{u_{\max} - u_x} v(t) \tag{8}$$

or, alternatively, as a function of the cylinder displacement $u(t)$

$$\dot{m}(t) = -\frac{m_0}{u_{\max} - u_x} \sqrt{-2a_x(u_{\max} - u(t))} \tag{9}$$

Further, the equation defining the isentropic model of the flow (equation (3)) can be applied to obtain an analytical formula that defines the required change of the outflow area

$$A_{\text{valve}}(t) = \frac{m_0}{u_{\max} - u_x} \sqrt{-2a_x(u_{\max} - u(t))}/C_{\text{dis}} \times \sqrt{2}\sqrt{\frac{k(q^{2/k} - q^{(k+1)/k})}{k-1}} \frac{p_x}{\sqrt{RT_x}} \tag{10}$$

Let us recall here that the parameters u_x, p_x, and T_x are determined by the use of (equation (5)) and depend on identified impact parameters, i.e. mass M and the initial velocity of the impacting object V_0. Thus, equation (10) reveals the dependence of the optimal outflow area $A_{\text{valve}}(t)$ on the identified impact conditions. In other research works, adaptation to impact conditions was achieved through the use of a real-time control system. Herein, we effectively use the fact that the optimal area of the gas outflow is a function of the cylinder displacement and we provide a semi-passive method of changing the outflow area along the cylinder stroke. For this purpose, we find the overall width of the vents d_{vent} located at the upper cylinder, which provides the optimal area of the valve during overlapping with the rectangular slot of height h located at the lower cylinder. The corresponding integral equation reads

$$\int_u^{u+h} d_{\text{vent}}(\bar{u})\mathrm{d}\bar{u} = A_{\text{valve}}(u) \tag{11}$$

A simplified solution of this equation can be found by assuming an infinitesimally small value of the orifice height and a constant value of the width of the vent in this region. It defines the change of the optimal width of the vent along the cylinder stroke

$$d_{\text{vent}}(u) = \frac{A_{\text{valve}}(u)}{h} \tag{12}$$

This formula remains valid for the considered range of slot heights, which remains relatively small. The optimal change of the valve shape and the resulting force response in terms of the displacement are presented in Figures 4 and 5, respectively.

The conducted numerical simulations indicate that the optimal location, size, and shape of the orifice depend considerably on the mass and the velocity of the impacting object. An increase in the mass of the damped body results in an extension of the compression distance determined by equation (5) and in a decrease in the maximal orifice area determined by equation (10) (Figure 4(a)). In turn, a higher value of the initial velocity does not cause any significant change in the maximal orifice size (Figure 4(b)). The obtained force–displacement characteristics also depend on the mass and velocity of the impacting object (Figure 5(b)). Nevertheless, the optimality of the dissipation process is always provided.

3.1. Comparison with semi-active solution

To demonstrate the effectiveness of the semi-passive impact absorber, a comparison of the proposed adaptable pneumatic shock absorber with an adaptive pneumatic shock absorber that utilizes a semi-active control strategy was conducted. Both absorbers enable realization of the optimal system path, which is determined at the beginning of the process. The comparison of the force–displacement characteristics for the adaptable SOFT-DROP device and a shock absorber controlled using a bang–bang strategy (proposed in Mikułowski et al., 2009) is presented in Figure 6. In the theoretical case, the optimal system path is followed exactly (Figure 6(a)). In practice, in the adaptable pneumatic absorber, the accuracy of path following is affected by the accuracy of the identified model of the gas flow, whereas, in the adaptive pneumatic absorber, it depends on the frequency limits of the valve control.

The on–off valve control was performed at a high frequency of 4 kHz, which ensures realization of the optimal system response (maximum pneumatic force of 331 N) with an accuracy of ±25 N (Figure 6(b)). By contrast, the operation of the proposed adaptable absorber was disturbed by a decrease in the discharge coefficient by 10%. This causes the maximum pneumatic force generated during the second stage to exceed the optimal value by 28 N (Figure 6(c)).

It can be concluded that for a reasonable flow model and identification accuracy and a feasible control frequency of the semi-active valve, the efficiencies of both

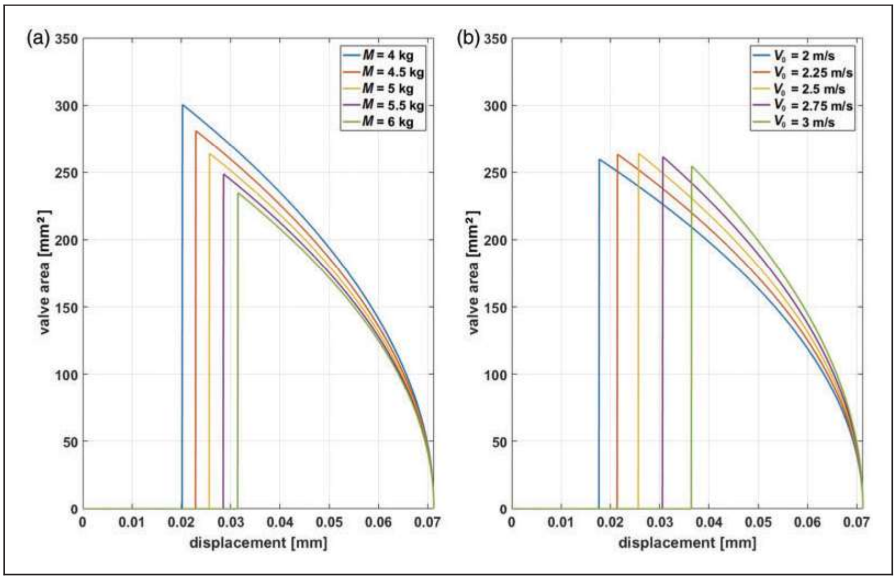

Figure 4. Optimal valve area as a function of the top cylinder displacement: (a) different masses; and (b) different initial velocities of the damped body.

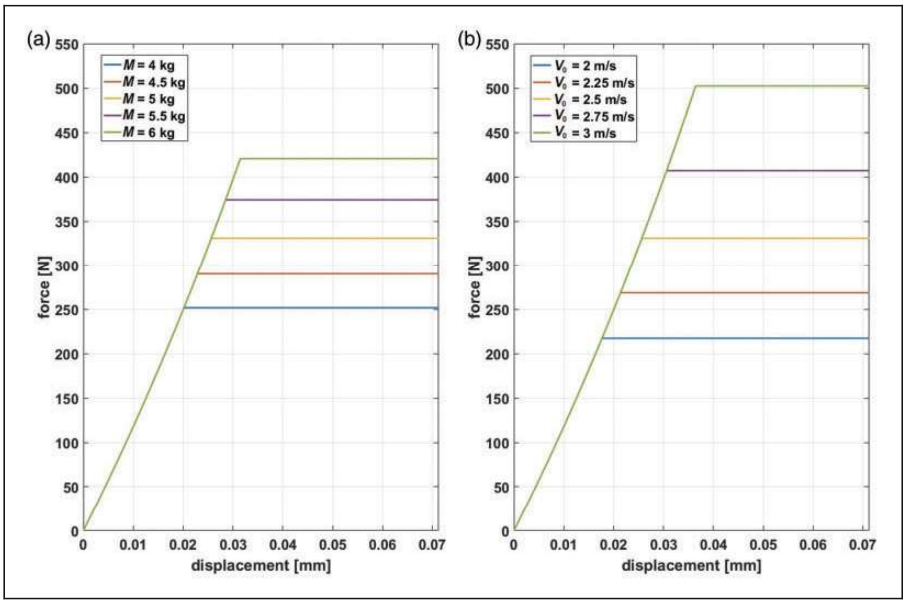

Figure 5. Reaction force as a function of the top cylinder displacement: (a) different masses; and (b) different initial velocities of the damped body.

systems are comparable. The superiority of the proposed adaptable system with the semi-passive control is revealed in its simpler construction and thus a higher reliability.

3.2. Design process and influence of manufacturing inaccuracies

To design a reliable demonstrator of a real shock absorber, the effects of friction force, uncontrolled leakages, and imprecise vent shaping were included in the mathematical model. According to the graph (Figure 7(b)), too small a valve area results in a stroke of the absorber that is not fully utilized, leading to a certain rebound of the top cylinder. In turn, undesired leakages in the system lead to the situation when the impact energy is not entirely dissipated within the assumed stroke. Thus, the influence of friction and system leakages can be taken into account during the design phase to achieve a possibly optimal operation of the shock absorber. Moreover, considering both these effects allows a fully optimal response of a nonideal system to be obtained.

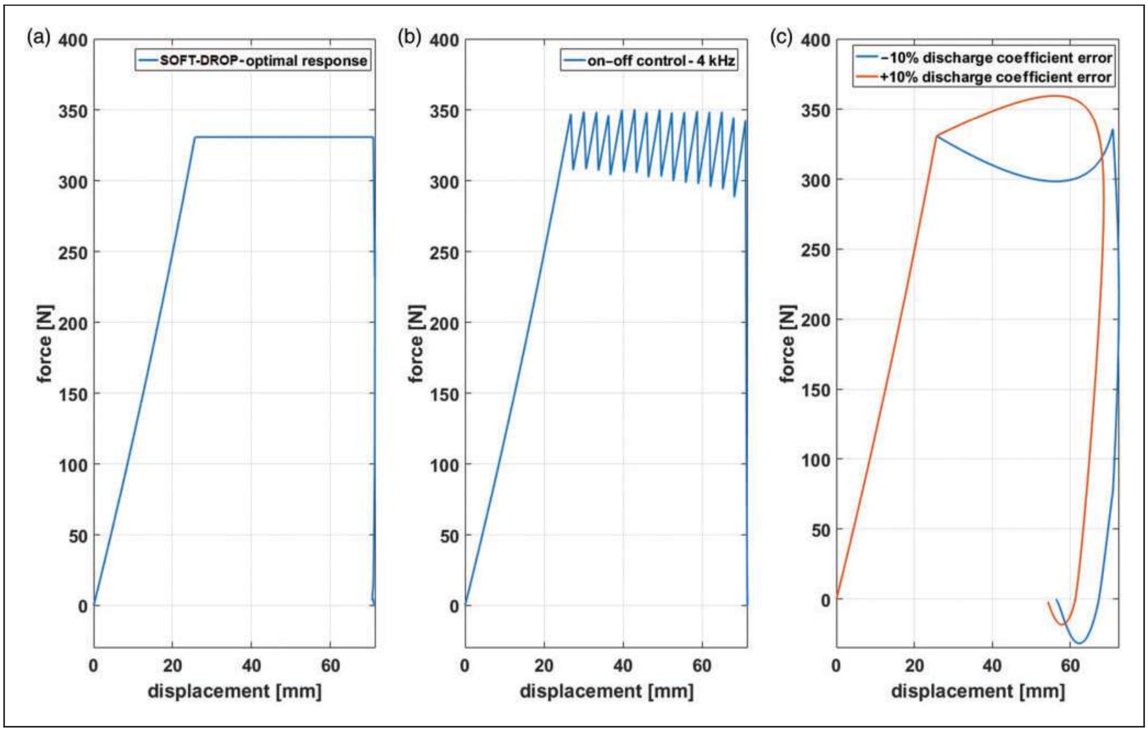

Figure 6. Force–displacement response to a given impact loading: (a) preliminarily determined optimal system path; (b) response of the adaptive absorber with a bang–bang control executed at a frequency of 4 kHz; and (c) response of the adaptable absorber with an inaccurate flow model identification.

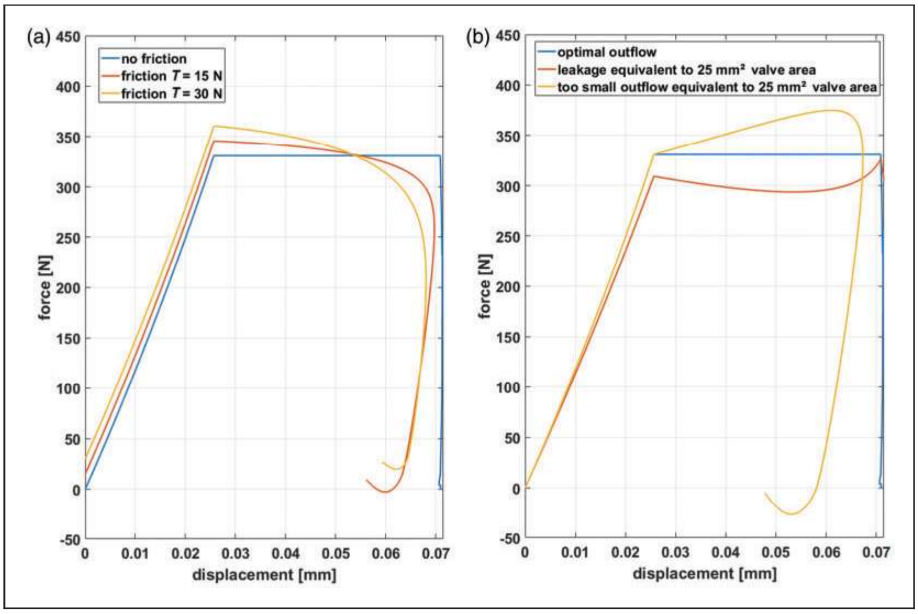

Figure 7. (a) Influence of high friction in the system. (b) Influence of leakages or insufficient valve area.

4. Experimental verification of the concept and model validation

To examine the discussed concept and validate the derived mathematical model, a demonstrator of the shock absorber was designed and manufactured using a ZORTRAX M200 3D printer, according to the specifications presented in Table 2. The plastic construction of the SOFT-DROP demonstrator was strengthened by 2 mm thick aluminum plates placed

Table 2. Specifications of SOFT-DROP demonstrator.

Internal and external diameters of top cylinder (mm)	Internal and external diameters of bottom cylinder (mm)	Depth of cylinders (mm)	Number of slot–vent pairs	Dimensions of each slot (mm)
145, 150	150, 155	150	2	64×2
Impact conditions	Corner radius (mm)	Material of the cylinders	Thickness of the layer (mm)	Infill of cylinder walls (%)
$M = 5\,kg$ $V_0 = 2.5\,m/s$	4	Z-HIPS high-impact polystyrene	0.14	100

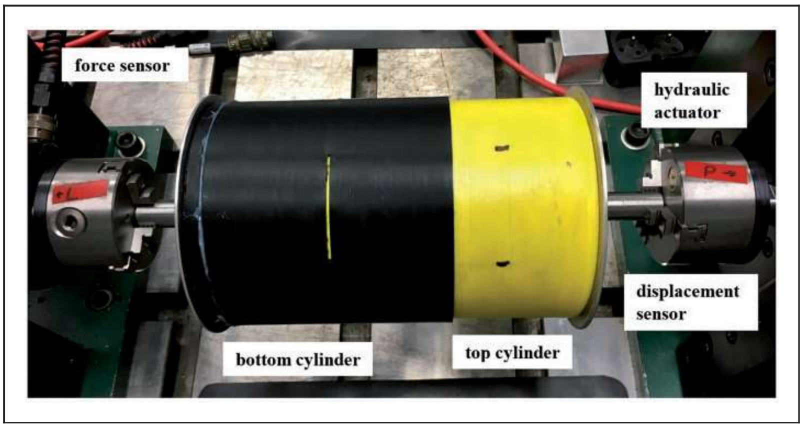

Figure 8. Laboratory stand used for SOFT-DROP testing.

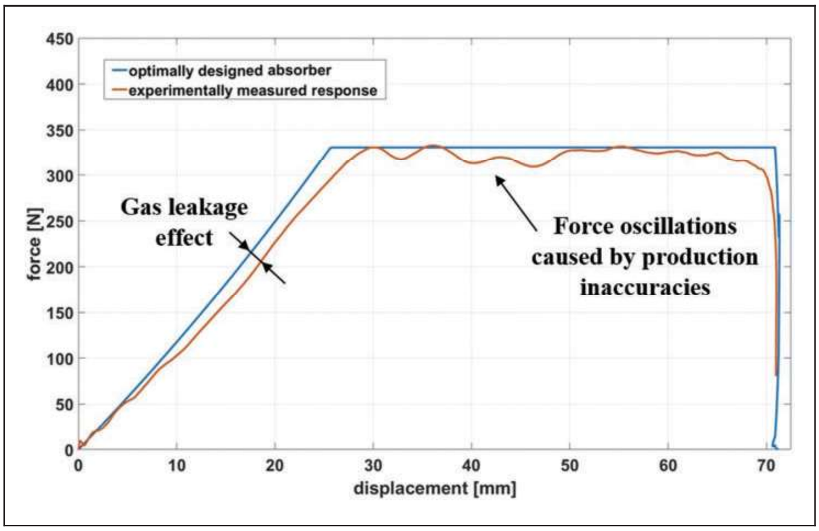

Figure 9. Comparison of the force response of the model and the demonstrator.

at the bottom and top of the cylinders. Precise grinding of the cylinder walls and the use of a lubricant allowed an easily moveable and sufficiently air-tight connection of the cylinders to be obtained. The valve for gas release was realized by two slots in the bottom cylinder and two appropriately shaped vents cut in the top cylinder.

The SOFT-DROP demonstrator mounted in the laboratory test stand is shown in Figure 8.

Table 3. Comparison of proposed semi-passive solution with other classes of shock absorbers.

	Passive	Semi-passive SOFT-DROP	Semi-active	Active
Adaptation	Not applicable	Single reconfiguration before impact reception	During impact	During impact
Efficiency	Moderate, depends on excitation conditions	High, comparable with semi-active	High	Highest
Weight	Smallest	Slightly higher than for passive device	Moderate	Moderate or high
Energy consumption	Not applicable	Small	Moderate	High
Operation in the case of electronics failure	Not applicable	Passive suboptimal response	Passive response	Possible destabilization of the system

The test rig was equipped with a fast hydraulic actuator, a displacement sensor, and a force sensor. The cylinders were mounted concentrically between the actuator and the force sensor in order to register the force response of the system under kinematic excitation.

The SOFT-DROP device was tested for kinematic excitation resembling the optimal kinematic response of the absorber under assumed impact conditions. The corresponding (unfiltered) force response as a function of the top cylinder displacement is presented in Figure 9. The difference between the results of numerical simulation and the experimental tests during the first phase of system operation is associated with a small gas leakage. In the second stage, the force is maintained at an almost constant level. This confirms the concept of system operation and the correctness of the model applied for the optimal passive design. Small fluctuations of the reaction force probably correspond to slight geometric asymmetry or elastic effects associated with deformations of cylinders.

The gas leakage during the compression phase corresponds to a gas release area of 25 mm^2. This results from the fitting accuracy of the cylinder diameters at a level of 0.11 mm, which can be significantly reduced. For instance, using fitting H6h5, the difference in diameters can be reduced to 0.04 mm or less, which results in a gas release area of 10 mm^2. To provide low friction and simultaneous precise fitting of the cylinders, one can be manufactured from aluminum and one from plastic. Moreover, such a solution will decrease elastic deformations of the top cylinder and corresponding force oscillations. Finally, to eliminate asymmetries in the vent shapes, the cylinder can be manufactured by injection molding.

5. Conclusions

The proposed concept of an adaptable pneumatic shock absorber replaces previously applied semi-active devices controlled in real time. The conducted numerical study revealed that a constant value of generated reaction force can be obtained using overlapping slots and vents of decreasing width along the cylinder length. It was proved that the system is efficient even in the case of inaccurate determination of the vent shape due to imprecise identification of the flow model. In addition, it was shown that the system is robust to friction and gas leakages resulting from manufacturing inaccuracies. The experimental tests proved the feasibility of the SOFT-DROP concept, the relevance of the presented mathematical model, and the correctness of the proposed design methodology. Finally, it can be concluded that the novel semi-passive technique of adaptation to impact excitation was successfully validated. To highlight the attractiveness of the proposed semi-passive device, a general comparison with passive, semi-active, and active solutions was conducted; the results are presented in Table 3.

Planned future research will be dedicated to the development of robust mechanisms, providing a single system reconfiguration at the beginning of the impact absorption process. The ultimate goal of this research is an application of the shock absorber in an inexpensive airdrop system for the delivery of light packages, which will provide adaptation to various weights of cargo and drop heights. The shock absorber designed in this case study provides deceleration below 7 g and can therefore be applied in an airdrop system for the emergency delivery of fragile electronic devices, sensors, computers, and medical equipment.

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Article B

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Can the inerter be a successful shock-absorber? The case of a ball-screw inerter with a variable thread lead

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Abstract

This paper investigates an application of a ball-screw inerter for mitigation of impact loadings. The problem of impact absorption is to provide a minimum reaction force that optimally decelerates and eventually stops an impacting object within the available absorber stroke. It significantly differs from vibration mitigation problems which are typical application of inerters. The paper demonstrates that the optimum absorption can be achieved by fully passive means. For known values of the object mass and inerter parameters, the obtained solution is independent of the impact velocity. The optimum passive absorption is achieved by employing a variable thread lead. As a result, two force components emerge, the typical inertance-related force and a damping-like term, and sum up to provide the optimum constant deceleration force. This result is relatively unique: conventional absorbers do not provide a constant force even with complex active control systems. Finally, an optimization problem is formulated to reduce the influence of process uncertainties (range of possible mass values, unknown friction). The results are verified and analyzed in a numerical example.

1. Introduction

Since inerter was introduced in [1] by prof. Malcolm Smith, the research in inerter-based structural control has been increasingly flourishing [2]. The spectrum of considered

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applications grows very rapidly and includes earthquake protection of buildings [3–6], mitigation of excitations in various suspensions systems (trains [7], road vehicles [8–12], aircraft landing gears [13–15] or even vessels [16]), mitigation of wind and wave induced loads in offshore wind turbines [17], gust load alleviation in truss-braced wings [18], improving walking performance of humanoid robots [19–21], motorcycle compensators [22], etc. Moreover, inerters are studied as a part of energy harvesting systems, as proposed and analyzed in [23,24]. Besides case studies and reports on practical applications of inerters, there is also a large number of fundamental researches on the influence of the device on vibrating structures [25] and simple systems [26], principles of system tuning [27] and optimal design [28–30] or on effects of imperfections on performance of inerter-based systems [31].

Typically, an inerter is implemented using a rack-pinion or a ball-screw mechanism, but there are reports on other inerter implementations, e.g., a fluid-based inerter is considered in [32,33] and a pivoted flywheel inerter based on living-hinges is proposed in [34]. In [35], a design based on a lightweight planetary flywheel is described and proved to have basically the same characteristics as the traditional rack-pinion inerters. In comparison to the typical mass component, the significant advantage of using an inerter is the fact that a similar performance can be obtained with a much smaller total mass of the system.

Most of the publications on inerters consider only passive devices, that is inerters with a specific, pre-designed inertance. Recently, there is a growing number of reports and researches on adaptive inerters. A device that employs a continuously variable transmission in order to enable modifications of the inertance is discussed and experimentally tested in [36–38]. Semi-actively controlled inerters are presented in [39,40]. Such adaptive inerters can be especially useful when the loading conditions are uncertain at the design stage or vary during the operational cycle.

The most common applications of inerters are various systems for mitigation of vibrations. In contrast, this paper considers an application of a passive ball-screw inerter for impact absorption. An earlier attempt to elaborate an inertial shock-absorber was described in [41], where a different concept of a controllable system was introduced and analyzed in detail. The addressed problem is to provide the minimum reaction force of a possibly flat profile that optimally decelerates an impacting object within the available absorber stroke. A typical passive inerter with a constant inertance cannot be directly used for such a purpose, on its own, for two important reasons:

1. The contact of the impacting object with the inerter terminal is a collision with a mass equivalent to the inertance of the device, which results in a detrimental impulsive momentum transfer instead of the intended steady deceleration with the lowest and possibly constant possible force. This necessitates an application of various impact interfaces in order to temporarily accumulate a large part of the impact energy and release it to the inerter. These interfaces, commonly springs [41], usually do not provide the optimum constant deceleration force.
2. After the collision, the object and the inerter terminal move forward together. The inertance is positive, so that providing a decelerating force of a proper sign requires an acceleration of the terminal, which contradicts the intended deceleration of the object.

Here, these problems are addressed without any interface and in a fully passive way by considering a ball-screw inerter with a variable thread lead:

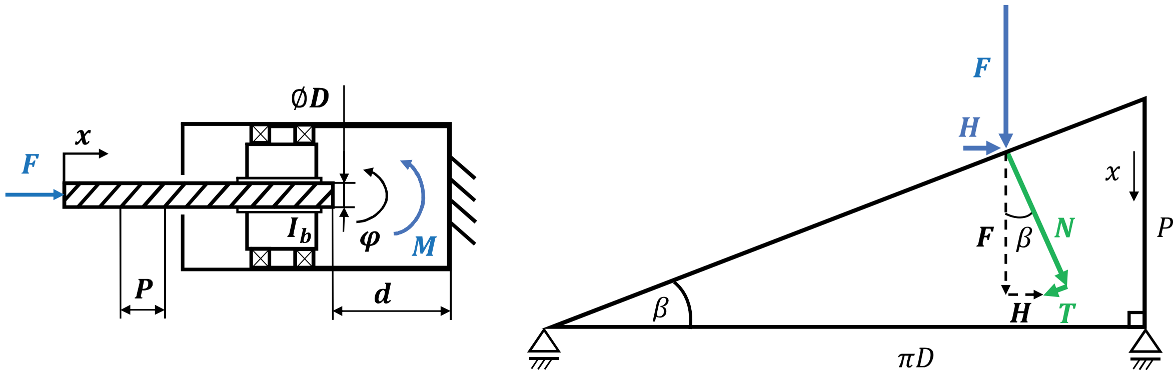

Fig. 1. The considered ball-screw inerter: a general scheme (left); the geometry of the thread line and the balance of forces (right). Notice that the right-hand side terminal of the inerter is fixed.

1. At the beginning of the stroke, the thread is parallel to the screw axis and the effective inertance vanishes, which avoids the problem of the impulsive momentum transfer.
2. As a result of the thread variability, two force components emerge: an inertance-related term and a damping-like term as a byproduct. These components sum up to provide the globally optimum constant deceleration force of the proper sign.

The obtained solution is independent of the impact velocity. However, variations of the object mass and inerter parameters (e.g., dry friction coefficient) change the deceleration profile. A formulation is thus proposed to optimize the absorption process when these process parameters are uncertain.

The paper is structured as follows: the following Section 2 discusses the ball-screw inerter with a variable thread lead and/or a variable inertia and derives its constitutive equation. Section 3 considers the problem of impact absorption, derives the nonlinear equation of motion of the decelerated object and applies it to determine the optimum thread lead; then an approach is proposed to optimize the thread lead when selected process parameters are uncertain. Finally, the approach is verified in the numerical example presented in Section 4.

2. Ball-screw inerter with a variable thread lead

For the sake of clarity, the considered inerter is discussed using a relatively simple 1 DOF model shown in Fig. 1; note that the second terminal of the device is fixed. The inerter is composed of a screw which is moving forward (under impact excitation as discussed in Section 3), a nut and a flywheel that is connected to the nut. The nut and the flywheel rotate together in accordance with the thread geometry defined by the thread lead $P(x)$, where x is the axial displacement of the screw. The flywheel and the nut act as an accumulator of mechanical energy. The inertance $b(x)$ might change with the screw displacement x due to two reasons:

1. the variable moment of the flywheel inertia $I_b(x)$ and/or
2. the variable thread lead $P(x)$.

This section derives the constitutive equation of such an inerter. Such an equation typically states that the force F is proportional to the acceleration $\ddot{x}$. Here, due to the effects of the variable thread lead and/or inertia, the equation turns out to involve also a damping-like term that depends on the velocity $\dot{x}$.

2.1. Basic formulas

Geometric relations. The inclination angle β of the thread is related to the lead $P(x)$ as follows:

$$\tan\beta = \frac{P(x)}{\pi D}, \tag{1}$$

where D is the diameter of the ball-screw connection.

Kinematic relations. The thread lead $P(x)$ defines a kinematic relation between the linear velocity $\dot{x}$ of the screw and the angular velocity $\dot{\varphi}$ of the nut and the flywheel. The infinitesimal screw displacement dx is proportional to the infinitesimal change of the flywheel angle dφ,

$$\frac{\mathrm{d}x}{P(x)} = \frac{\mathrm{d}\varphi}{2\pi}. \tag{2}$$

Division of Eq. (2) by the infinitesimal time interval dt leads to the formula that binds the angular velocity of the flywheel $\dot{\varphi}$ with the screw velocity $\dot{x}$,

$$\dot{\varphi} = \frac{2\pi\dot{x}}{P(x)}. \tag{3}$$

By the chain rule $\dot{P}(x) = \dot{x}P'(x)$, and differentiation of Eq. (3) yields the angular acceleration of the flywheel,

$$\ddot{\varphi} = \frac{2\pi}{P(x)}\left(\ddot{x} - \dot{x}^2\frac{\mathrm{d}}{\mathrm{d}x}\ln P(x)\right). \tag{4}$$

Angular momentum and the torque. The angular momentum of the flywheel is equal to the product of the flywheel inertia $I_b(x)$ and its angular velocity $\dot{\varphi}$. The time derivative of the angular momentum is equal to the unbalanced torque M that acts on the flywheel,

$$M = \frac{\mathrm{d}}{\mathrm{d}t}\Big(I_b(x)\,\dot{\varphi}(\dot{x}, P(x))\Big). \tag{5}$$

Inserting Eq. (3) into Eq. (5) and using Eq. (4), the following formula for the torque is obtained:

$$M = \frac{2\pi}{P(x)}\left(I_b(x)\,\ddot{x} + \left(\frac{\mathrm{d}I_b}{\mathrm{d}x} - I_b(x)\frac{\mathrm{d}}{\mathrm{d}x}\ln P(x)\right)\dot{x}^2\right). \tag{6}$$

Friction force. The dry friction model is assumed for the ball-screw connection. The friction force T is thus given by

$$T = \mu N, \tag{7}$$

where μ is the dry friction coefficient and N is the normal force, see Fig. 1 (right).

Force balance and the torque. The torque M acting on the flywheel is caused by the horizontal force H, see Fig. 1 (right), which acts on the arm $D/2$. The force H can be determined by using the conditions of equivalence of the primary external loads (F and H) and the forces acting in the normal and tangential direction (N and T), which read:

$$H = N \sin\beta - T \cos\beta, \tag{8a}$$

$$F = N \cos\beta + T \sin\beta. \tag{8b}$$

Thus, the torque M caused by the horizontal force H equals:

$$M = \frac{1}{2} DH = \frac{1}{2} D(N \sin\beta - T \cos\beta). \tag{9}$$

By combining Eqs. (7), (8b) and (9), one can obtain the following formula:

$$M = \frac{1}{2} DF \frac{\sin\beta - \mu \cos\beta}{\cos\beta + \mu \sin\beta}. \tag{10}$$

2.2. Constitutive equation

In the previous section, two formulas for the torque M have been derived: Eq. (6) based on the angular momentum and Eq. (9) based on the balance of forces. By comparing the two formulas and using the geometric relation Eq. (1), the following constitutive equation is obtained for the inerter with a variable thread lead, variable moment of the flywheel inertia and dry friction:

$$F = \left(I_b(x)\, \ddot{x} + \left(\frac{\mathrm{d}I_b}{\mathrm{d}x} - I_b(x) \frac{\mathrm{d}}{\mathrm{d}x} \ln P(x) \right) \dot{x}^2 \right) K(P(x), \mu), \tag{11a}$$

where, for notational simplicity,

$$K(P(x), \mu) \;=\; \frac{4\pi}{D} \frac{\pi D + \mu P(x)}{P^2(x) - \mu\pi D P(x)}. \tag{11b}$$

By analyzing Eq. (11a), one can notice that the reaction force of the considered inerter is composed of a component proportional to the terminal acceleration (an inerter term) and a component proportional to the square of the terminal velocity (a damping-like term):

$$F = b(x)\ddot{x} + f(I_b(x), P(x))\dot{x}^2. \tag{12}$$

The first component is the typical reaction force of an inerter, even though it depends here also on the displacement x and the dry friction coefficient μ. The second component arises due to the change of the angular momentum; this force might be referred to as "inertial damping" because it results from the inertial effects and has a damping-like influence on the motion of the impacting object. However, it can be nonzero even in a conservative system, which can be confirmed by setting $\mu = 0$ and $\mathrm{d}I_b/\mathrm{d}x = 0$ in Eq. (11a). Conversely, if the flywheel momentum $I_b(x)$ is proportional to the thread lead $P(x)$, the damping-like term will vanish even with a nonzero friction.

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3. Ball-screw inerter as a shock absorber

3.1. Impact absorption problem

The general aim of an impact-absorbing system [42] is to entirely decelerate an impacting object of mass m within the available absorber stroke d, simultaneously ensuring the lowest possible value of the reaction force F (or the deceleration $-\ddot{x}$, which are equivalent for a single specific value of m). The integral of the force with respect to the absorber displacement x must be equal to the initial kinetic energy E_{imp} of the decelerated object. Therefore, the general aim is to provide the following constant reaction force during the entire process,

$$\bar{F} = \frac{E_{\text{imp}}}{d}. \tag{13}$$

Without any other constraints imposed by the absorber design and dynamics, the problem can be formalized as the following optimization problem, in which two simple conditions are used to ensure the optimal Eq. (14) and successful Eq. (15) operation of the system:

$$\text{minimize rms}\, F(t) \tag{14}$$

with respect to F

$$\text{subject to} \quad \int_0^d F\, \mathrm{d}x = E_{\text{imp}}. \tag{15}$$

The deviation of the decelerating force $F(t)$ from the optimum flat profile Eq. (13) is quantified in the root mean square (rms) terms,

$$\text{rms}\, F(t) \;:=\; \left(\frac{1}{T} \int_0^T \left(F(t) - \bar{F} \right)^2 \mathrm{d}t \right)^{\frac{1}{2}}, \tag{16}$$

where T is the duration of the process. In comparison to the alternative formulation based on the maximum value, the rms-based objective function Eq. (14) allows the effects of possible short-time peaks of the force to be properly accounted for: even if they do not convey much momentum, such peaks can have a significant detrimental effect on the optimization based directly on the maximum value. Notice also that, in the absence of other constraints, both formulations are equivalent and yield the same optimum Eq. (13).

In any practical setup, the evolution of the reaction force $F(t)$ is additionally constrained to the absorber dynamics and depends on the values of the system parameters, so that obtaining a constant and globally optimal reaction force Eq. (13) is impossible with most absorber designs. For example, in the considered system, the force $F(t)$ depends on the moment of inertia $I_b(x)$ and the thread lead $P(x)$. This section (i) introduces these additional constraints for the case of the considered inerter, (ii) demonstrates that they do not prevent the inerter from achieving the global optimum Eq. (13) and finally (iii) proposes a general formulation to be used when some process parameters are uncertain.

3.2. Absorber dynamics

In the considered 1D movement setting shown in Fig. 1, a point mass approximation is used for the decelerated object, so that

$$E_{\text{imp}} = \frac{1}{2} m v_0^2, \tag{17}$$

where m is the mass of the object (together with the mass of the screw) and v_0 is its initial velocity. The balance of forces is formulated as follows:

$$F + m\ddot{x} = 0. \tag{18}$$

Combined with the constitutive equation of the ball-screw inerter Eq. (11a), Eq. (18) yields the nonlinear equation of motion of the decelerated object,

$$\ddot{x}\left(\frac{m}{I_b(x)} + K(P(x), \mu)\right) + \dot{x}^2 K(P(x), \mu)\frac{\mathrm{d}}{\mathrm{d}x}\ln\frac{I_b(x)}{P(x)} = 0, \tag{19a}$$

or if the flywheel inertia I_b is constant, $I_b = \text{const}$,

$$\ddot{x}\left(\frac{m}{I_b} + K(P(x), \mu)\right) - \dot{x}^2 K(P(x), \mu)\frac{\mathrm{d}}{\mathrm{d}x}\ln P(x) = 0, \tag{19b}$$

where $K(P(x), \mu)$ is defined in Eq. (11b) and used here for notational simplicity. The equation of motion is augmented with the following initial conditions:

$$x(0) = 0 \qquad \dot{x}(0) = v_0. \tag{20}$$

By analysis of Eq. (19b) three important facts can be noticed:

- The impact velocity v_0 affects the solution only by changing the time scale of the displacement process, which can be easily verified by a substitution $x(t) \leftarrow x(ct)$. Consequently, the results of any optimization performed with respect to the maximum deceleration or the maximum force are independent on v_0.
- The mass m and the flywheel inertia I_b occur in Eq. (19b) only in the form of the ratio m/I_b. Consequently, given the threaded screw, the inerter can still be tuned to the expected range of the masses by modification of the flywheel inertia only.
- The dry friction coefficient μ and the diameter D of the ball-screw connection occur in Eq. (19b) only in the form of their product μD. Therefore, at the design stage of the absorber, any probable uncertainties in the dry friction coefficient can be partially offset by a proper choice of the diameter of the ball-screw connection.

Consequently, for the purpose of optimization aiming at the reduction of the influence of parameter uncertainties, most of the process parameters can be assumed constant. One should allow variations of the impact mass and of the dry friction coefficient (which might be difficult to be accurately determined and maintained at the constant level through the lifetime of the absorber),

$$m \in S_m \qquad \mu \in S_\mu, \tag{21}$$

where the sets S_m and S_μ will in practice most likely take the form of intervals.

Energy balance. Integration of the equation of motion Eq. (19a) with respect to the displacement yields the equation of energy balance:

$$\frac{1}{2}m\left(\dot{x}^2 - v_0^2\right) + 2\pi^2\left(\frac{I_b(x)}{P^2(x)}\dot{x}^2 - \frac{I_{b_0}}{P_0^2}v_0^2\right) + \int_0^x\left(K(P(\bar{x}), \mu) - \frac{4\pi^2}{P^2(\bar{x})}\right)I_b(\bar{x})\ddot{\bar{x}}\,\mathrm{d}\bar{x}$$
$$+\int_0^x\left(\frac{4\pi^2}{P^2(\bar{x})} - K(P(\bar{x}), \mu)\right)\frac{I_b(\bar{x})}{P(\bar{x})}\frac{\mathrm{d}P}{\mathrm{d}\bar{x}}\dot{\bar{x}}^2\,\mathrm{d}\bar{x} + \int_0^x\left(-\frac{2\pi^2}{P^2(\bar{x})} + K(P(\bar{x}), \mu)\right)\frac{\mathrm{d}I_b}{\mathrm{d}\bar{x}}\dot{\bar{x}}^2\,\mathrm{d}\bar{x} = 0. \tag{22}$$

In the above equation, the first two terms denote the change of the kinetic energy of the impacting object and the flywheel, whereas the subsequent terms indicate the sources of system energy changes. In particular, the third term indicates the dissipation caused by friction itself, while the fourth term indicates the energy dissipation caused jointly by the variability of the thread lead and by friction. Notice that the third and fourth terms vanish when there is no friction in the system, since then $K(P, 0) = 4\pi^2/P^2$, see Eq. (11b). The fifth term denotes the change of system energy caused by the variability of the flywheel inertia, and it remains nonzero even when friction is negligible. The energy balance Eq. (22) confirms that (i) a system, in which the flywheel inertia changes, is non-conservative, and that (ii) a system with a variable thread lead remains a conservative system, provided the friction is zero and the flywheel inertia is constant.

3.3. Optimum absorption for a specific impact and friction

This section shows that the proposed absorber can achieve the global unconstrained optimum of the absorption process Eq. (13) for any specific values of the process parameters m and μ. According to Eq. (13), the globally optimum deceleration process is achieved when the deceleration $-\ddot{x}$ generated by the inerter is constant and stops the object within the entire available absorber stroke d. Therefore, the optimum acceleration and velocity of the object can be expressed by simple equations:

$$\ddot{x} = -\frac{v_0^2}{2d}, \qquad \dot{x}^2 = v_0^2\left(1 - \frac{x}{d}\right). \tag{23}$$

By substituting the above optimal kinematics into the equation of motion Eq. (19a), one obtains the following differential equation:

$$\frac{m}{I_b(x)} + \left(1 - 2(d-x)\frac{\mathrm{d}}{\mathrm{d}x}\ln\frac{I_b(x)}{P(x)}\right)K(P(x), \mu) = 0. \tag{24}$$

which can be used to determine either the thread lead $P(x; m, \mu)$ or the flywheel moment of inertia $I_b(x; m, \mu)$ that provide the optimum constant reaction force for the assumed specific values of the process parameters. In each case, the other unknown function (P or I_b) needs to be assumed. However, if both $I_b(x)$ and $P(x)$ are variable and unknown, an additional relation has to be introduced, such as, e.g., a cost function to be minimized.

Constant flywheel inertia. In the following, the optimum variable thread lead $P(x; m, \mu)$ is determined, while the moment of the flywheel inertia is assumed constant ($I_b = \text{const}$). This is the desired case in which the optimum constant value of the decelerating force is achieved in a fully passive manner, by simply engraving a variable thread. The procedure of determining the variable moment of inertia is analogous and not presented: notice that in practical implementations, it might not be possible to force $I_b(x)$ to vary in the desired manner by fully passive means and a more complex control system might be required for this purpose.

By assuming that $I_b = \text{const}$, Eq. (24) is simplified to the following form:

$$\frac{m}{I_b} + \left(1 + 2(d-x)\frac{\mathrm{d}}{\mathrm{d}x}\ln P(x)\right)K(P(x), \mu) = 0. \tag{25}$$

Even if strongly nonlinear, the differential Eq. (25) can be solved numerically to yield the thread lead $P(x; m, \mu)$ that optimally decelerates the object of mass m, under the assumption

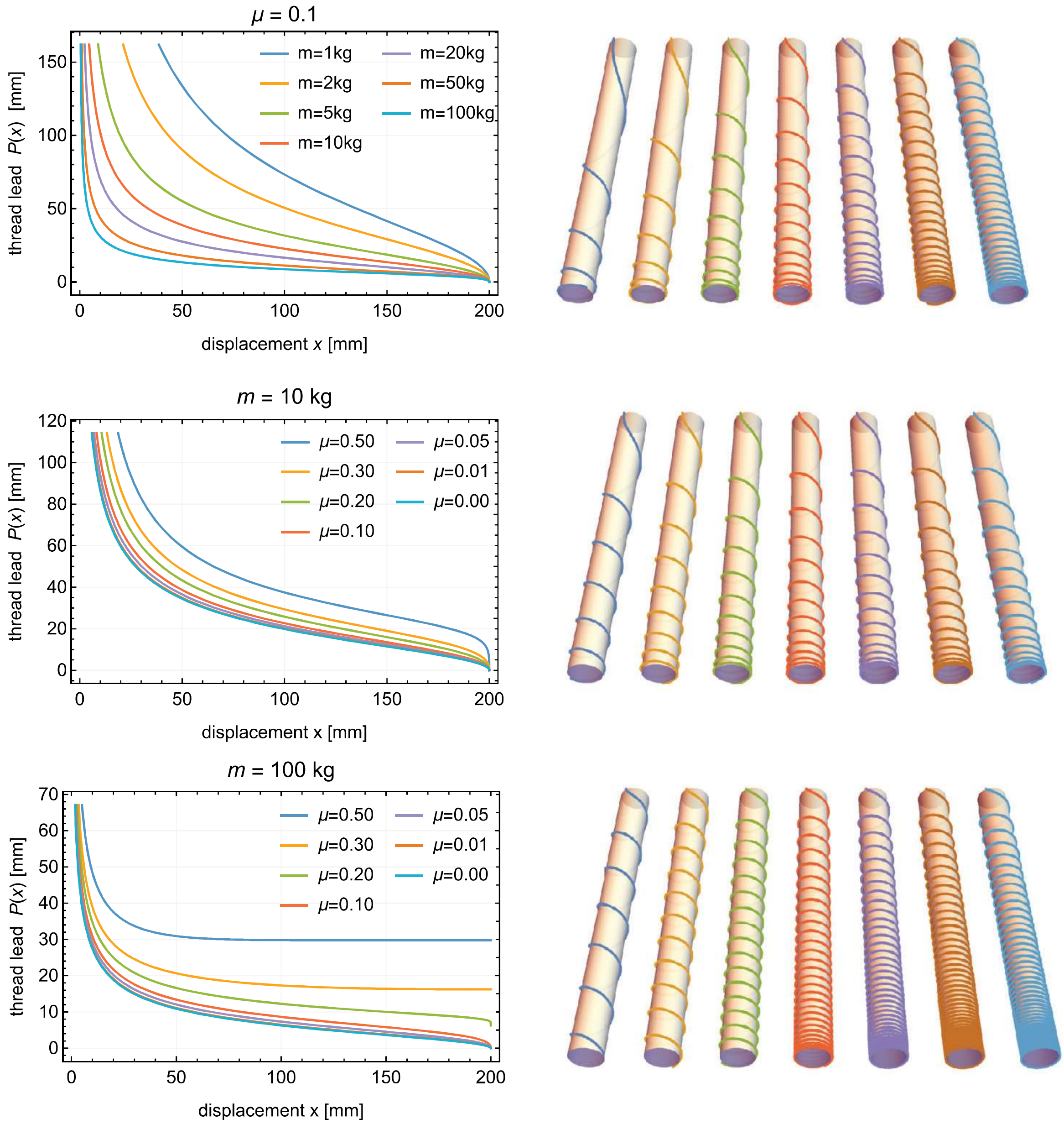

Fig. 2. Numerical example: optimum thread lead (top row) in dependence on the mass of the decelerated object for $\mu = 0.1$; (middle row) in dependence on the dry friction coefficient for $m = 10$ kg; (bottom row) in dependence on the dry friction coefficient for $m = 100$ kg.

that the coefficient of friction equals μ. Eq. (25) is augmented with the following initial condition:

$$P(0) = \infty. \tag{26}$$

Although of a slightly untypical from the mathematical point of view, such an initial condition is justified from the engineering point of view: it states that in the initial phase of the movement the thread is parallel to the screw axis. As a result, a smooth spin-up of the flywheel is ensured, which helps to avoid any inelastic impact and the related impulsive momentum transfer, as well as any friction-related screw locking that could appear at low initial values of the thread lead. Examples of threads that satisfy Eq. (26) are illustrated in Fig. 2.

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In general, Eq. (25) needs to be solved numerically for $P(0)\to\infty$. However, in the specific case without friction, $\mu=0$, the equation has a simple analytical solution that directly describes the optimum thread lead as a function of the displacement x:

$$P(x; m, \mu=0) = 2\pi\sqrt{\frac{I_b}{m}}\sqrt{\frac{d-x}{x}}. \tag{27}$$

Notice that Eq. (25) and thus the optimum thread lead are independent of the initial velocity v_0 of the object. It is a promising result in terms of practical applications: once the mass and the coefficient of friction are known, a single thread is optimal for a range of impact velocities (e.g., different drop heights in cargo discharge operations).

3.4. General formulation

Optimization objective. If an entire range of masses $m\in S_m$ is considered, the two objectives mentioned in Section 3.1 (minimization of the reaction force F and of the peak deceleration $-\ddot{x}$) are no longer equivalent. For notational simplicity, they are replaced by a single objective of minimization of

$$H_{\mathrm{rms}} := \mathrm{rms}\, H(t), \tag{28}$$

where either

$$H(t) := F(t) \quad \text{or} \quad H(t) := -\ddot{x}(t). \tag{29}$$

In practice, the choice between the force and the deceleration might depend on the specific technical problem: basing the optimization on the reaction force is reasonable in applications that require protection of the absorber or the fuselage (such as landing gears [43,44]), while the deceleration should be selected in applications where the protection of the passengers or the cargo is the top priority [45]. Note also that the force-based criterion will most probably focus the optimization on large masses only, so that in case of a smaller mass the (smaller) reaction force might be very far from the global optimum Eq. (13).

The absorption process, and thus the reaction force F and the objective function H_{rms}, depend via the equation of motion Eq. (19b) and the balance of forces Eq. (18) on the object mass m and the dry friction coefficient μ, as well as on the thread lead $P(x)$,

$$F = F(m, \mu, P(x)), \qquad H_{\mathrm{rms}} = H_{\mathrm{rms}}(m, \mu, P(x)). \tag{30}$$

Optimization problem. The general optimization problem can be stated as follows:

$$\begin{aligned} &\text{minimize} && \sup\nolimits_{m\in S_m} \sup\nolimits_{\mu\in S_\mu} H_{\mathrm{rms}}(m, \mu, P(x)),\\ &\text{with respect to} && P(x),\\ &\text{subject to} && \int_0^d F(m, \mu, P(x))\,\mathrm{d}x = E_{\mathrm{imp}} \quad \text{for all } m \in S_m \text{ and } \mu \in S_\mu. \end{aligned} \tag{31}$$

Note that the optimization problem Eq. (31) is a difficult problem of variational optimization: the thread lead $P(x)$ is not a number or a vector, but rather a function of the screw displacement x, so that the search domain is a certain functional space. In the following, the search domain of $P(x)$ is restricted to yield a more standard optimization problem.

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Table 1
Numerical example: parameters of the screw inerter and the impact.

	Symbol	Unit	Value/range
Length	d	[mm]	200
Diameter	D	[mm]	20
Flywheel moment of inertia	I_b	[kg mm^2]	100
Dry friction	μ	–	[0, 0.5] *
Impacting mass	m	[kg]	[1, 100]*
Impact velocity	v_0	[m/s^2]	10

* [0, 0.1] for the purpose of optimization.
** [5 kg, 15 kg] for the purpose of optimization (10 kg ± 50%).

Restriction of the domain. In Section 3.3, it is shown that Eq. (25) provides the global optimum for any specific values of the process parameters m and μ. However, the general formulation Eq. (31) takes into account the entire ranges S_m and S_μ of these parameters. The approach described in Section 3.3 cannot be directly used in such a case, as the optimum thread lead $P(x; m, \mu)$ depends on m and μ. To avoid variational analysis of Eq. (31), it is proposed here to restrict the search domain to the set of the optimum functions $P(x; \bar{m}, \bar{\mu})$, and to solve the problem (31) in the restricted domain. More specifically, the search domain is restricted to the following two-parameter family of the thread leads:

$$\left\{P(x; \bar{m}, \bar{\mu}) \,|\, \bar{m} \in S_m, \bar{\mu} \in S_\mu\right\} \tag{32}$$

In this way, the general problem of variational optimization Eq. (31) is simplified to the following classical problem of parametric optimization with respect to $(\bar{m}, \bar{\mu}) \in S_m \times S_\mu \subset \mathbb{R}^2$:

$$\begin{aligned} &\text{minimize} && \sup_{m\in S_m} \sup_{\mu\in S_\mu} H_{\text{rms}}(m, \mu, P(x; \bar{m}, \bar{\mu})), \\ &\text{with respect to} && (\bar{m}, \bar{\mu}) \in S_m \times S_\mu \subset \mathbb{R}^2, \\ &\text{subject to} && \int_0^d F(m, \mu, P(x; \bar{m}, \bar{\mu}))\,\mathrm{d}x = E_{\text{imp}} \quad \text{for all } \bar{m} \in S_m \text{ and } \bar{\mu} \in S_\mu. \end{aligned} \tag{33}$$

The optimum thread lead found in the restricted search domain might not coincide with the general optimum found in variational analysis, but it can be intuitively expected to be relatively close to it because of the way the restricted domain Eq. (32) is constructed: it is the set of the thread leads that are optimum for certain specific values $\bar{m}$ and $\bar{\mu}$.

4. Numerical case study

4.1. The absorber and the impact parameters

For the purpose of the numerical study, specific values of parameters are assumed for the considered screw inerter and the impact, see Table 1. The parameters μ and m are varied within the listed ranges. The following section uses impractical, wide ranges of parameters in order to investigate and illustrate the general characteristics of the device, while for the purpose of optimization, the last section uses narrower ranges that are more likely to occur in practice.

As noticed at the end of Section 3.3, the crucial Eq. (25) used to determine the optimum thread lead depends on μ and D only through their product μD. Similarly, it depends on m

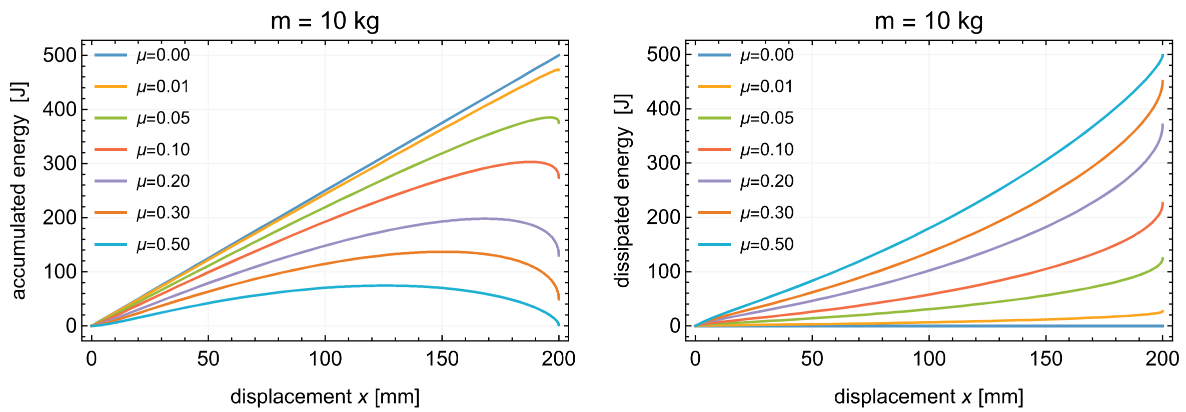

Fig. 3. Numerical example for $m = 10$ kg and $v_0 = 10$ m/s^2: (left) energy accumulated in the flywheel; (right) energy dissipated by friction.

and I_b only through their ratio m/I_b. Consequently, the relative influence of these parameters on the resulting optimum thread lead is closely related, and in the following variations of the diameter D and the flywheel inertia I_b are neglected.

4.2. Optimum thread lead and energy balance

Eq. (25) is used to compute the optimum thread lead $P(x)$ for the specific values of the parameters listed in Table 1. The left column of Fig. 2 shows the dependence of the optimum leads on the process parameters: mass m (at $\mu = 0.01$) and dry friction coefficient μ (at $m = 10$ kg and $m = 100$ kg). The respective threads are visualized in the right column.

Notice that for high enough masses and dry friction coefficients, the end value of the optimum thread lead is nonzero, $P(d) > 0$. On the other hand, the impact energy must be completely absorbed to fulfill Eq. (15), so that the axial velocity at the end of the stroke is always zero, $\dot{x}(d) = 0$. Consequently, in these cases the angular velocity at the end of the stroke vanishes, $\dot{\varphi}(d) = 0$, see Eq. (3), and at the end of the process no energy is accumulated in the form of flywheel rotation: the entire energy of the impact is dissipated by friction. In contrast, for the idealized case of $\mu = 0$, there is obviously no dissipation at all and the entire impact energy is accumulated in the rotational motion of the flywheel.

Fig. 3 shows the accumulated energy (left) and the dissipated energy (right) in function of the displacement x for different levels of the dry friction. Both plots illustrate the case $m = 10$ kg. The initial velocity v_0 affects only the scaling of the vertical axes, in this particular example a relatively high value of 10 m/s^2 is used. Without friction, no dissipation occurs and the entire absorbed energy is accumulated in the flywheel. For intermediate values of the friction coefficient, a part of the energy is dissipated and a part is accumulated in the rotating flywheel. Finally, in the case of a high friction, the entire impact energy is dissipated and the device effectively operates like a friction damper; nevertheless, it still provides optimal deceleration of the object independently of the impact velocity, which is usually not possible in typical passive dampers. Fig. 4 plots the total dissipated energy as a percentage of the entire impact energy in dependence on the mass m and the dry friction coefficient μ. As evident in Fig. 4, the considered absorber operates in between the two following extreme modes of operation:

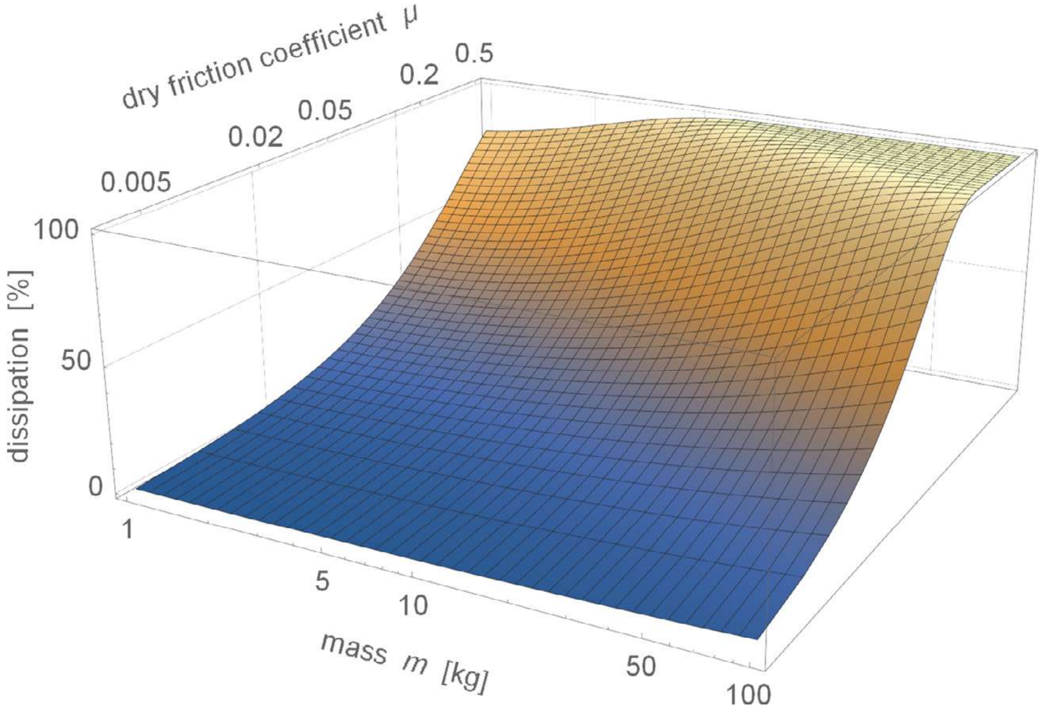

Fig. 4. Numerical example: the total dissipated energy as a percentage of the total impact energy in function of the mass m and the friction coefficient μ.

1. Inertial accumulation of the entire impact energy in the rotational motion of the flywheel and no dissipation. This case occurs for negligible friction and relatively low impacting masses.
2. Frictional dissipation of the entire impact energy, which occurs for relatively high impacting masses and a significant friction level. The inertia of the flywheel is used only for temporary accumulation of the energy in order to ensure that a constant level of the deceleration force is maintained during the process.

In practice, ball-screw mechanisms can be designed to have very small values of the friction coefficient, so that the first option (inertial accumulation) might be considered as feasible in practice. The second option (frictional dissipation) has three important drawbacks: (i) the system should be designed to be wear-resistant and (ii) efficiently transfer the heat outside, which for higher masses and velocities is an engineering challenge itself, and moreover, (iii) if the friction coefficient is overestimated and/or the mass underestimated, the actual impacts might not be fully absorbed, so that the system would violate the crucial constraint Eq. (15). Notice also that the mode of operation depends actually on the product μD and the ratio m/I_b, so that the influence of possible uncertainties in μ and m can be partially decreased by a proper design of the diameter D and the flywheel inertia I_b, see the remarks at the end of Section 3.3.

4.3. Optimization for a range of impacts and friction coefficients

The previous section studies the proposed absorber for a range of process parameters m and μ, assuming that the thread is always designed optimally. However, in reality the thread must be decided in advance, while the actual impacting mass and friction might differ from the

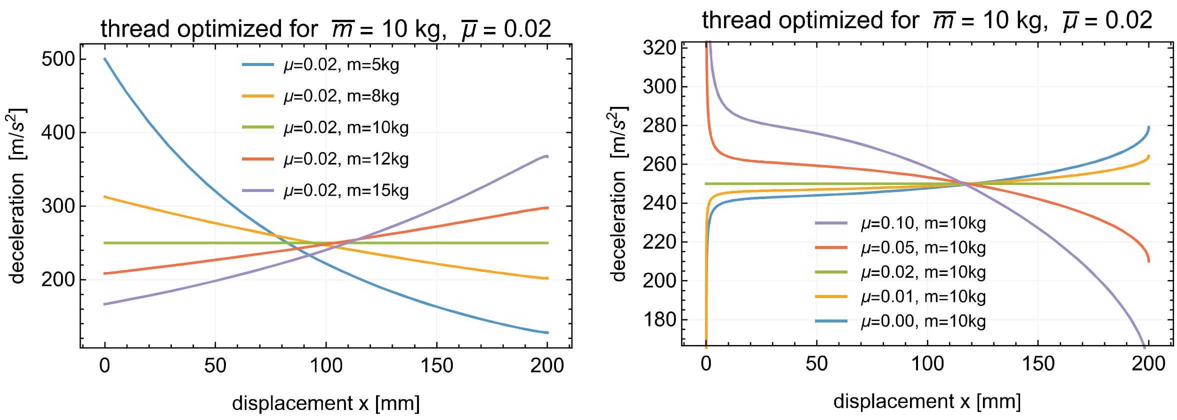

Fig. 5. Numerical example: deceleration profiles obtained for misestimated mass (left) and misestimated friction coefficient (right). The thread lead is optimum for $m = 10$ kg and $\mu = 0.02$; the initial velocity is $v_0 = 10$ m/s^2.

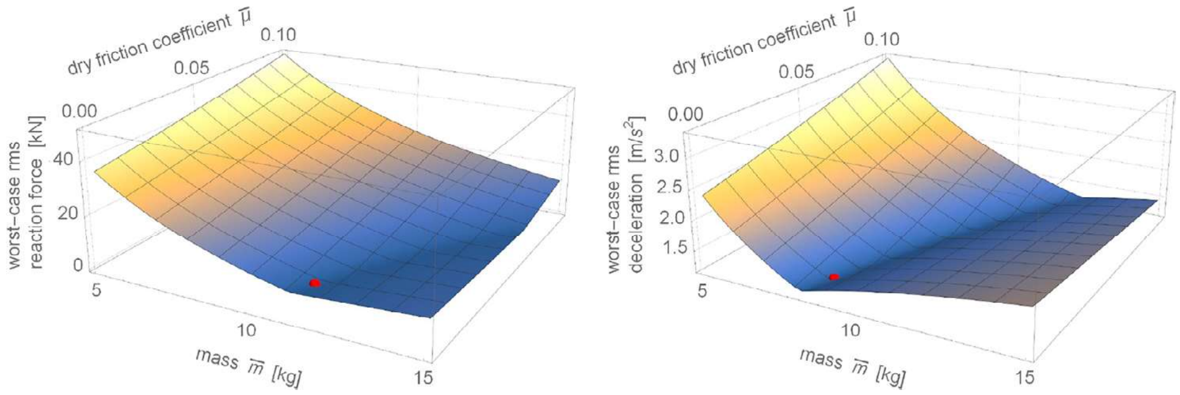

Fig. 6. Numerical example: the worst-case rms of the reaction force (left) and the worst-case rms of the deceleration (right) in dependence on the design values $\bar{m}$ and $\bar{\mu}$ that determine the thread $P(\bar{m}, \bar{\mu}, I_b)$. The minima are marked with dots.

design values. Fig. 5 assumes that the thread lead is optimum for $m = 10$ kg and $\mu = 0.02$ and shows the decelerations obtained for various actual values of the impacting mass and friction coefficients. It can be observed that any misestimation of the process parameters shifts the deceleration profile away from the globally optimum constant value and increases its rms value. In particular, if the friction is underestimated, an initial deceleration peak occurs. Such a peak is short and does not influence considerably the rms value; in a practical application, it can be relatively easily alleviated using a simple interface element, e.g., rubber [46].

For optimization purposes, it is assumed here that the impacting mass equals 10 kg with the uncertainty 50% (that is $m \in [5\,\text{kg},\ 15\,\text{kg}]$) and that the actual dry friction coefficient is not larger than 0.1, see Table 1. The optimization approach proposed in Section 3.4 employs a thread lead $P(x; \bar{m}, \bar{\mu})$ that is optimum for certain design values $\bar{m}$ and $\bar{\mu}$, which are treated as optimization variables. Fig. 6 plots the two considered objective functions, see Eqs. (28) and (29),

$$\sup_{m \in S_m} \sup_{\mu \in S_\mu} \operatorname{rms} H(t; m, \mu, P(\bar{m}, \bar{\mu}), I_b), \tag{34}$$

in dependence on the optimization variables $\bar{m}$ and $\bar{\mu}$. Notice that each point in the plots is obtained by performing a large number of simulations (each one for a specific pair (m,

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$\mu) \in S_m \times S_\mu$) and taking the worst-case (maximum) rms value of the deceleration or of the force. The optima are clearly recognizable in Fig. 6 and marked explicitly with red dots:

- The optimum with respect to the reaction force is achieved for the thread design values $\bar{m} = 11.5$ kg and $\bar{\mu} = 0.008$. The worst-case (maximum) rms of the reaction force is then 8.6 kN.
- The optimum with respect to the deceleration is achieved for the thread design values $\bar{m} = 8.5$ kg and $\bar{\mu} = 0.012$. The worst-case (maximum) rms of the deceleration is then 1.2 m/s^2.

The locations of the optima depend on the objective function in a predictable way: the reaction force increases with the impacting mass, hence to minimize its rms value one should focus on large masses; the reaction forces for low masses might be then highly non-flat, but they are low anyway and thus of no concern. On the other hand, the decelerations do not depend so strongly on the mass, and thus the optimum in the midrange can be expected. In both cases, the optimum design corresponds to the mostly accumulative mode of operation without significant friction and dissipation. Therefore, the thread lead might be designed using Eq. (27) and the mass m at approximately one third of the mass range (to minimize the worst-case rms deceleration) or two thirds of the mass range (to minimize the worst-case rms reaction force).

5. Conclusion

Theoretical and numerical analyses presented in the paper indicate that an inerter can be successfully applied for absorption of impact energy and mitigation of impact loadings. However, instead of a classical inerter with a constant inertance, a device with a variable moment of inertia or a variable thread lead has to be used. The variable-thread inerter is a simpler solution since it can be constructed as a passive device, in which thread lead changes continuously along the screw.

For known process parameters (mass of the object and the dry friction coefficient), the optimal change of the thread along the screw can be precisely determined. Such an optimally shaped variable-thread inerter provides absorption of the entire impact energy with a constant level of the reaction force, independently on the initial velocity of the impacting object. In the case of a negligible friction, the proposed inerter provides a direct transfer of the impact energy into the rotational energy of the flywheel. If the friction is significant, the impact energy is only temporarily converted into the rotational energy of the flywheel and then eventually dissipated by friction. Finally, the proposed inerter can be designed in a way that provides an almost optimal impact absorption for a given set of process parameters. In such a case, the uncertainties in process parameters can be partially compensated by a proper choice of the variation of the thread lead along the screw.

In further research, it is planned to consider influence of nonlinearities such as elasticity of the thread and the backlash, and to progress towards a lab-scale demonstrator. Besides, applications of other types of inerters in problems of optimal impact absorption will be studied. In particular, semi-active inerters will be employed, such as inerters with a controllable moment of inertia and inerters based on magneto-rheological fluids, to utilize the paradigm of real-time adaptation and obtain the optimal mitigation of arbitrary impact loads.

Acknowledgment

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Adaptive inertial shock-absorber

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Abstract

This paper introduces and discusses a new concept of impact absorption by means of impact energy management and storage in dedicated rotating inertial discs. The effectiveness of the concept is demonstrated in a selected case-study involving spinning management, a recently developed novel impact-absorber. A specific control technique performed on this device is demonstrated to be the main source of significant improvement in the overall efficiency of impact damping process. The influence of various parameters on the performance of the shock-absorber is investigated. Design and manufacturing challenges and directions of further research are formulated.

S Online supplementary data available from stacks.iop.org/SMS/25/035031/mmedia

Keywords: adaptive impact absorption, adaptive inerter, semi-active control, shock-absorber

(Some figures may appear in colour only in the online journal)

1. Introduction

In recent years, mechanical structures with embedded active and adaptive absorbers or dissipators have received much attention in the area of vibration and impact control of structures. Large output bandwidth, efficient conversion or dissipation of energy and ease of integration of various time-oriented control algorithms make these devices very attractive for research. Active and adaptive methods of control for absorbers and dissipators can be used for example to emphasize and exploit their nonlinear characteristics. Owing predominantly to these features, mechanical structures can automatically adjust their response to compensate for external excitations or loads. Despite the growing body of research, it is still challenging to design appropriate absorbers and dissipators and to control structures with such devices.

At present, adaptive absorbers and dissipators are predominantly built based on the so-called smart materials such as magneto-rheological fluids, electro-rheological fluids, piezoelectric materials, shape memory alloys or granular materials. For example, in late 1990s, Carlson *et al* [11] developed a magneto-rheological damper applicable to on-and-off-highway vehicle suspension systems. It was experimentally demonstrated that magneto-rheological dampers can be applied for an effective control of the damping force in a vehicle suspension system. Many other researches (Mikułowski *et al* [6, 7, 13], Makowski *et al* [8], Bajkowski *et al* [12]) provided additional know-how in the field of design of effective active or semi-active suspension systems for vehicles, aircraft or other mechanical structures, based on various smart material technologies under proposed different optimization criteria.

A separate type of devices with embedded inertia elements used to accumulate and absorb energy of impact or vibration was proposed by Chen *et al* [3], Smith [9], Gomuła *et al* [10] and Lazar *et al* [1, 2]. They discussed numerous ideas of application of so-called inerter devices in the area of protection of structures against impact and vibration. They proposed an application of the inerter as an absorber of vibrations (in the form of a tuned spring–mass system) attached to the main body of a mechanical structure, or in a vehicle suspension as a strut that replaces traditional struts with springs and dampers, or the use of the inerter to simulate an additional mass element in a structure, or as a device able to absorb an impact to which the structure is exposed.

The goal of this paper is to briefly describe the first results obtained during the development of a new shock-absorbing device and to discuss the considered semi-active

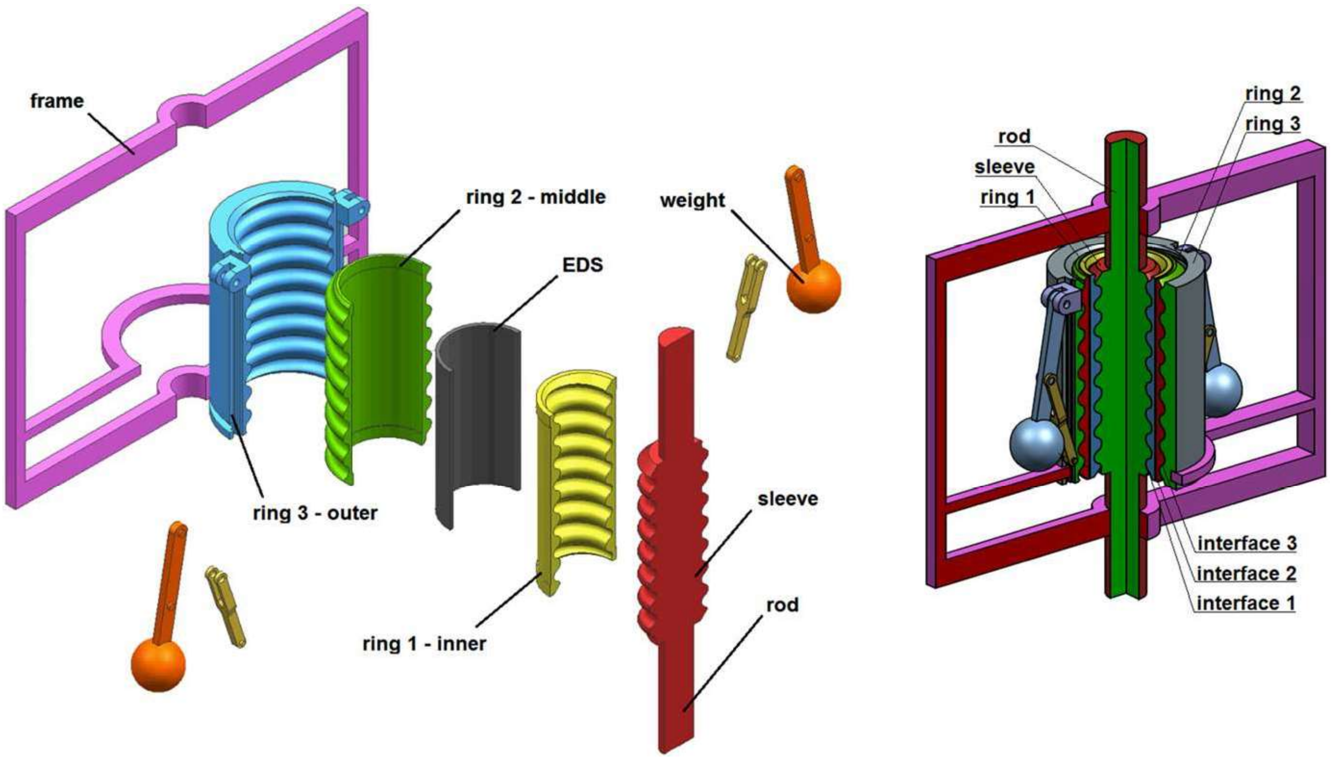

Figure 1. SPIN-MAN—the design of the shock-absorber.

control strategies for this device. The newly invented *adaptive inerter* system [4] is based on such a switchable device, which includes rotating discs with a controllable spinning orientation. The control of spinning or spinning management (SPIN-MAN) will be demonstrated to be the main source of significant improvement in the overall efficiency of impact damping process. The proposed impact absorption strategy is based on a controlled, on/off type of switching devices and it can be called *impact energy management*. Similarly, on/off type of switching devices have been used for effective damping of impact-born vibrations in the so-called prestress accumulation-release (PAR) technique [5], where a proper timing of instant release/attachment of structural joints leads to an extremely effective management of impact energy absorption.

Presented adaptive inertial shock-absorber is a concept of inerter device with two phases of operation. The first one, conversion of linear movement into rotation of inertial mass and the second, exploiting stored kinetic energy of rotating outer mass into damping process, due to initiation of inverse spinning of second, inner mass. The viscous interface between these two, oppositely rotating masses generates major damping effect. The SPIN-MAN device concept allows formulation of various objectives to be sought:

(a) absorption of impact load (subjected to various design constrains) with minimized maximal impact force during the loading process
(b) absorption of repetitive impacts (subjected to various design constrains) with minimized maximal impact force during the load series, and necessary restoration to the initial configuration of shock-absorber after each single impact
(c) absorption of harmonic loads (subjected to various design constrains), e.g. with minimal amplitude of deflections, which can be addressed e.g. to car suspension applications.

The first paper (in a planned series of publications) is devoted to problem (a) specified above and presents a quasi-optimal problem solution reached via a trial and error method.

2. SPIN-MAN as an adaptive inertial shock-absorber

Shock-absorber SPIN-MAN, shown in figure 1, converts a linear motion, caused by an impact acting on the cushioned structure (connected with the impact-absorber), into the rotary motion of its inertial parts. The SPIN-MAN device is composed of:

- a frame
- a rod with a threaded sleeve
- three rings (1—inner, 2—middle and 3—outer)
- energy dissipation surface (EDS)—dissipative connection of inner and middle ring
- two weights on arms
- three blockers of the relative rotation (sleeve/ring 1, ring 1/ring 2, ring 2/ring 3).

The rod is positioned in the frame, and the inner ring is screwed to the sleeve. The inner ring has, on its inner surface, a thread compatible with the thread on the sleeve; its outer surface is cylindrical. The middle ring has a cylindrical inner surface and a threaded outer surface. The outer ring has an inner thread compatible with the middle ring's thread and this thread provides a rotary motion in the direction opposite to the direction of motion provided by the thread on the sleeve. Between the inner ring and the middle ring, a connecting element, called the EDS, is placed. This part is a dissipative connection and it can be realized based on an elastomeric ring, a frictional connection or a connection filled with a magneto-rheological fluid, which viscosity (and thus the damping characteristics) can be changed using a magnetic field controller. The geometric and mass parameters of each part should be appropriately selected to achieve a good performance of the shock-absorber.

Weights connected with the outer ring are placed on the arms. Such a solution provides the possibility to control the position of the weights and hence the moment of inertia of the spinning mass. The connections of the main components have a significant influence on the SPIN-MAN's performance, because they enable the energy transfer from the impacting body to the inertial parts. Each connection was called an *interface* and they can be described as follows:

- interface 1—sleeve/ring 1, threaded connection; linear and angular displacements are possible;
- interface 2—ring 1/ring 2, dissipative connection (EDS); only angular displacements are allowed;
- interface 3—ring 2/ring 3, threaded connection (thread oriented contrary to the thread in the interface 1); the outer ring is allowed to rotate, the middle ring can rotate and move in the direction of the rod axis.

Each interface is equipped with a blocker of the relative rotary motion that can be separately turned on and turned off providing for a desired sequence of relative rotary motions.

When the impact acting on the amortized device is detected by the forward movement of the rod, the system of sensors should identify the characteristics of the shock—instant velocity growth and impact energy as an increase of the body kinetic energy. For known parameters of the impact, control system determines the required moment of inertia of rotating parts and the control strategy that will be performed with the blockers to ensure achievement of assumed objectives. Blockers on the first and the third interface are intended to control the energy transfer in the system, while the blocker on the second (middle) interface is used to activate the main process of energy dissipation. To increase the effectiveness of the system adapting to the impact conditions, the moment of inertia of outer ring can be changed continuously (by the position of the weights control) during the entire process of shock absorption and not only at its beginning. Appropriately selected parameters of the control strategy (switching instances of the blockers) lead to significant changes in the response of the damping device. Moreover, it is possible to additionally control the damping characteristics of interface 2, especially in the case of the connection filled with the magneto-rheological fluid. The large number of controllable parameters of the system enables one to change and adjust characteristics of the SPIN-MAN in order to obtain desired objectives that can be the optimal reduction of the maximum loads in the amortized system or the reduction of the damping time.

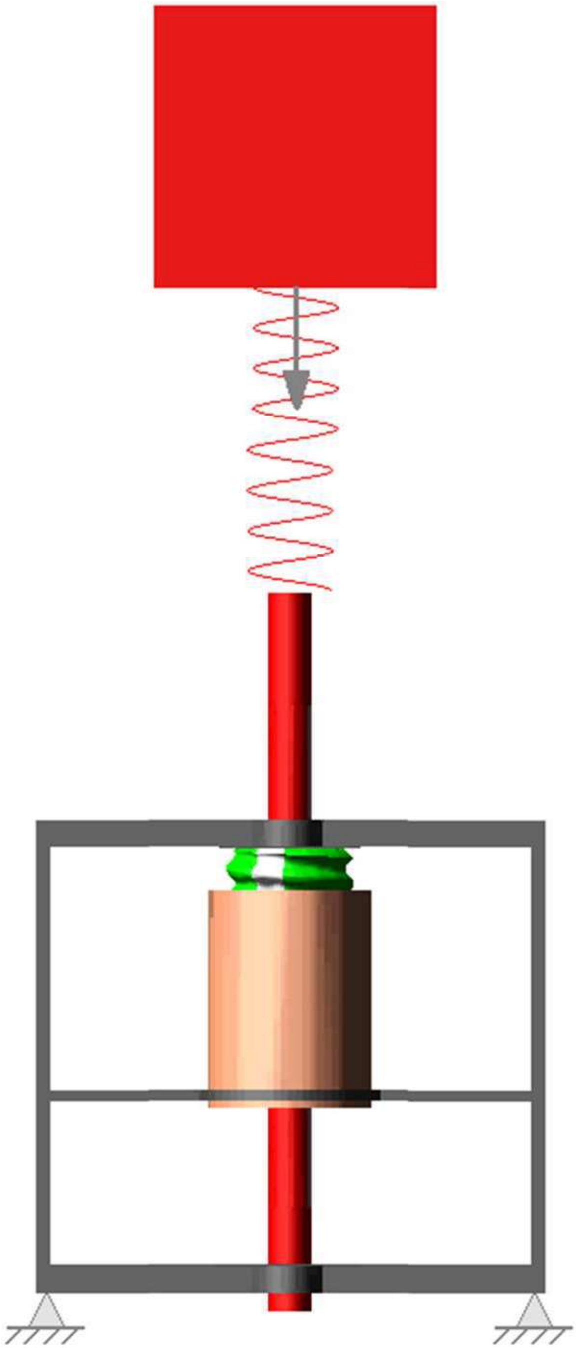

Figure 2. Scheme of the considered system.

3. Numerical simulation of the impact absorption

This section demonstrates an application of SPIN-MAN as a shock-absorber tuned for a smooth mitigation of a recognized impact, including its reception and dissipation. The adaptive device, placed in a fixed frame (figure 2), has been loaded by

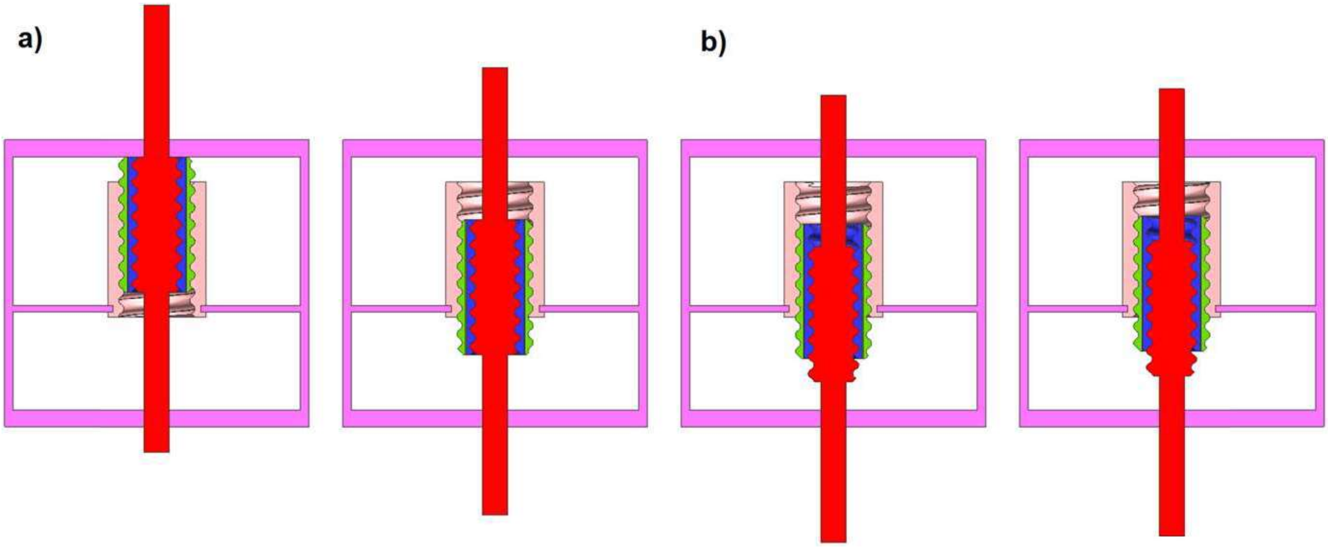

Figure 3. Sequence of the displacements of components: (a) phase 1, (b) phase 2.

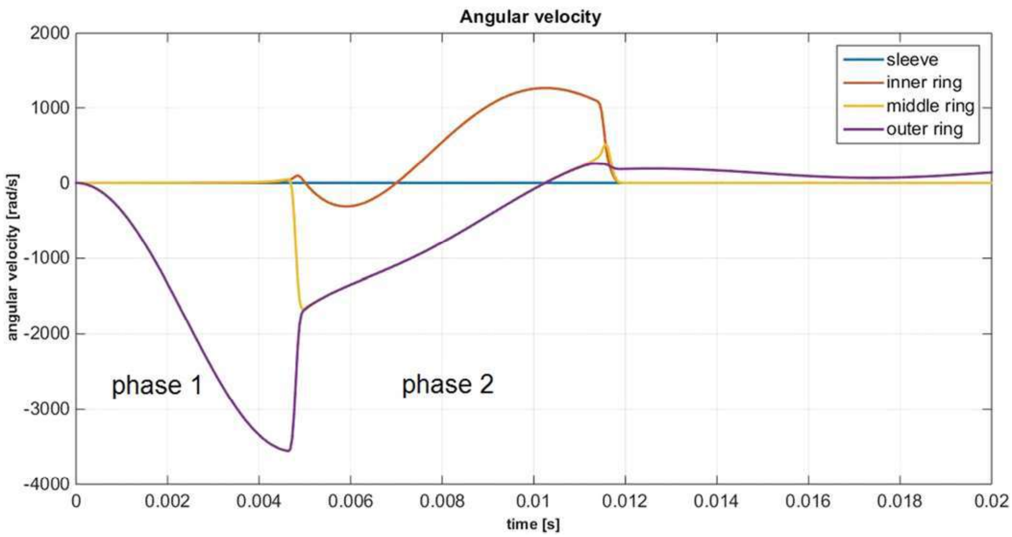

Figure 4. Angular velocity of SPIN-MAN's components.

Table 1. Main system parameters.

Cushioned body	Mass	10 kg
	Excitation—initial velocity	$10\ \mathrm{m\ s^{-1}}$
Spring	Spring constant	$1000\ \mathrm{N\ mm^{-1}}$
Threads	Thread diameter	26.5 mm; 42 mm
	Thread pitch	27 mm
EDS	Damping coefficient	$25\ \mathrm{N\ s\ mm^{-1}}$
Rings	Mass	0.1 kg
	Moment of inertia	$50\ \mathrm{kg\ mm^2}$

a dropping mass impacting it via a spring with an appropriately selected stiffness. The mass of 10 kg and with the initial velocity of $10\ \mathrm{m\ s^{-1}}$ has been assumed. The energy dissipation on the EDS was realized by the mechanism of viscous damping. Duration of the blockers switches was considered and turning on/off actions lasting 0.2 or 0.4 millisecond were assumed. A sample control strategy with two switching actions has been performed and simulated via the ADAMS software package to prove the potential of the SPIN-MAN concept. The absorption of the total impact energy (dropping mass 10 kg with initial velocity $10\ \mathrm{m\ s^{-1}}$) took less than 12 ms (while the period of harmonic excitation by the spring–mass system is 20 ms) and the force amplitude has been reduced by 50% (figure 6).

Parameters of the discussed simulation are presented in table 1.

The sequence of phases of SPIN-MAN operation is demonstrated in figure 3 and in the attached simulation movie (supplementary material, available at stacks.iop.org/SMS/25/035031/mmedia). Initially, interface 1 and interface 2 are blocked leading to the movement of the rod and the sleeve together with the inner and the middle ring (figure 3(a)). Displacement of them in the direction of the rod's axis provides the rotary motion of the outer ring. In a particularly

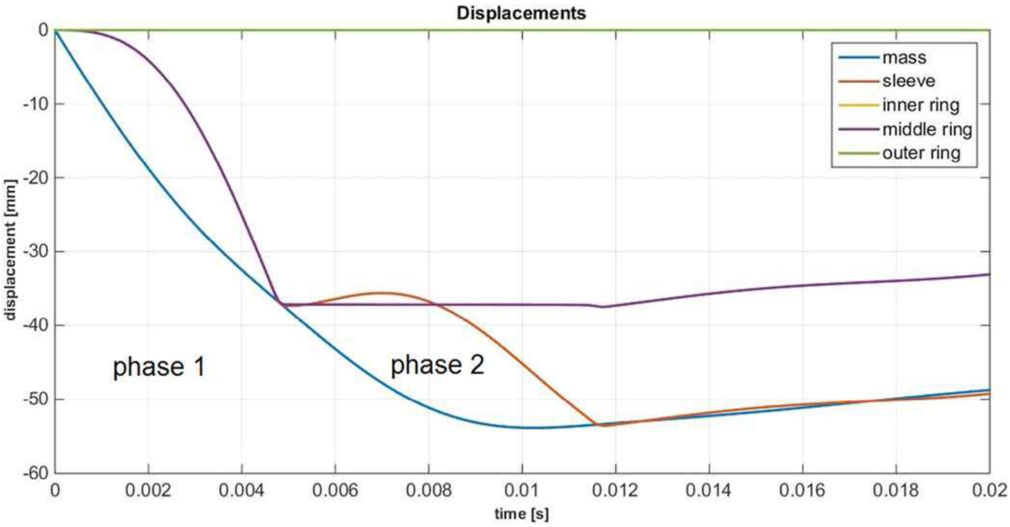

Figure 5. Displacements of SPIN-MAN's components.

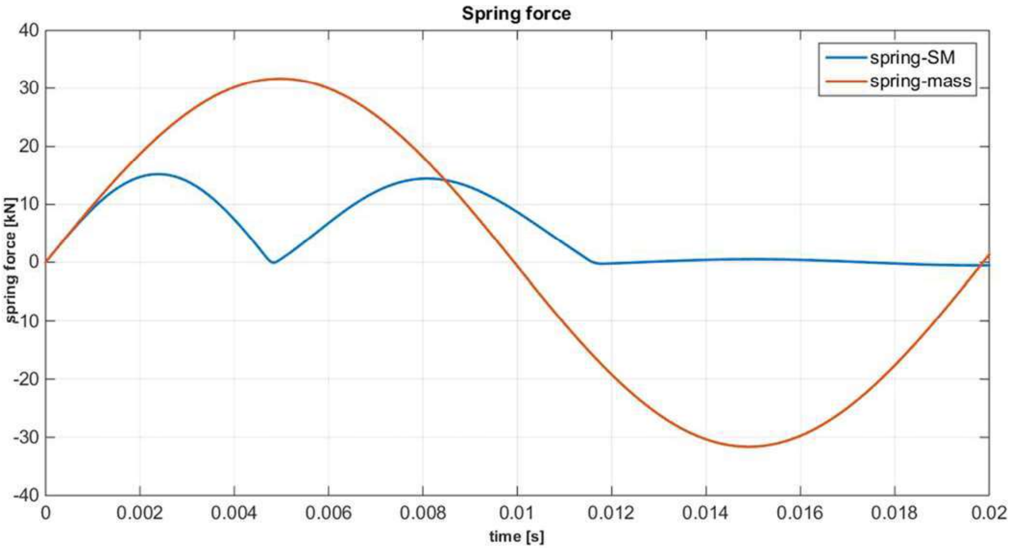

Figure 6. Impact force reduction by the use of the SPIN-MAN device.

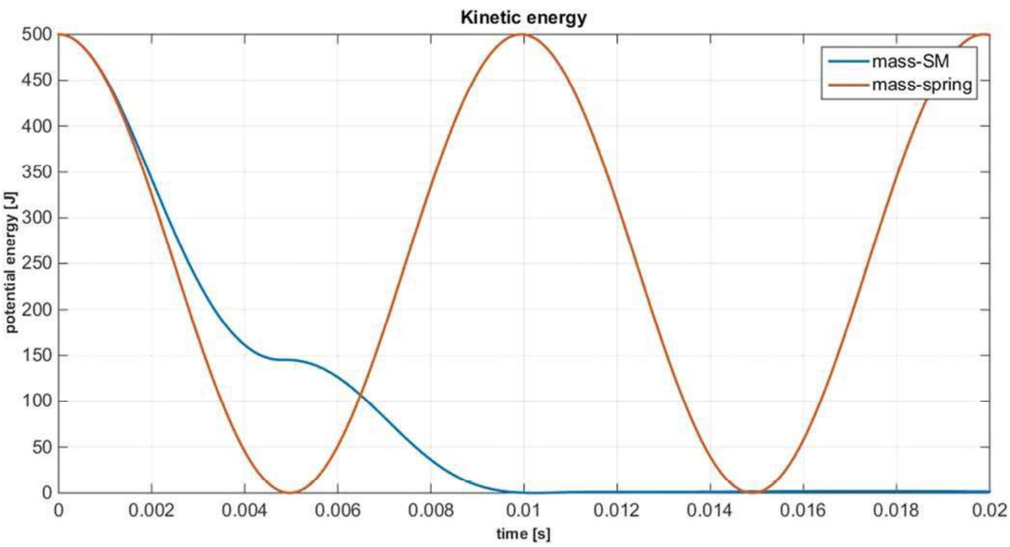

Figure 7. Kinetic energy of the mass (impact energy).

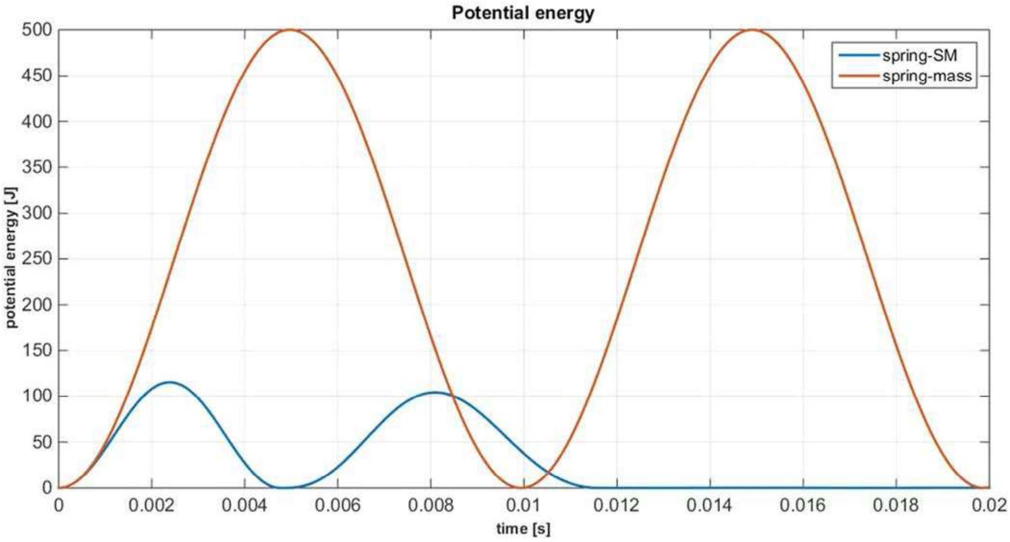

Figure 8. Potential energy of the spring.

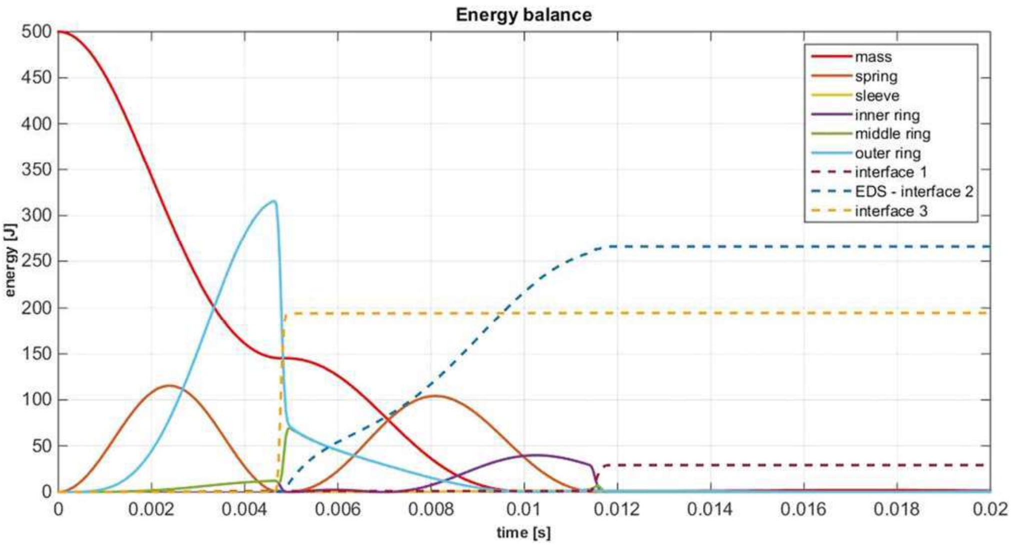

Figure 9. Energy balance—energy of the system components.

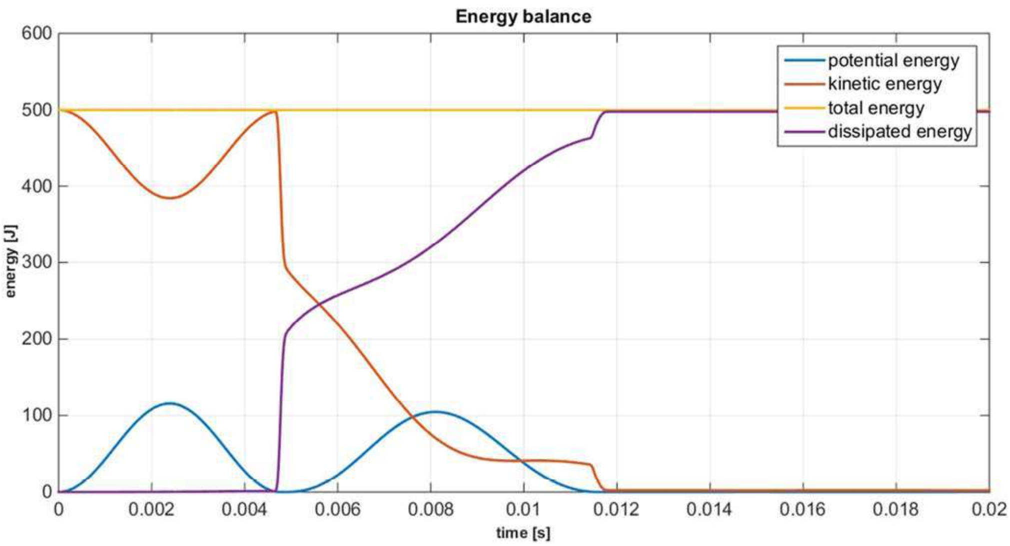

Figure 10. Energy balance for the whole system.

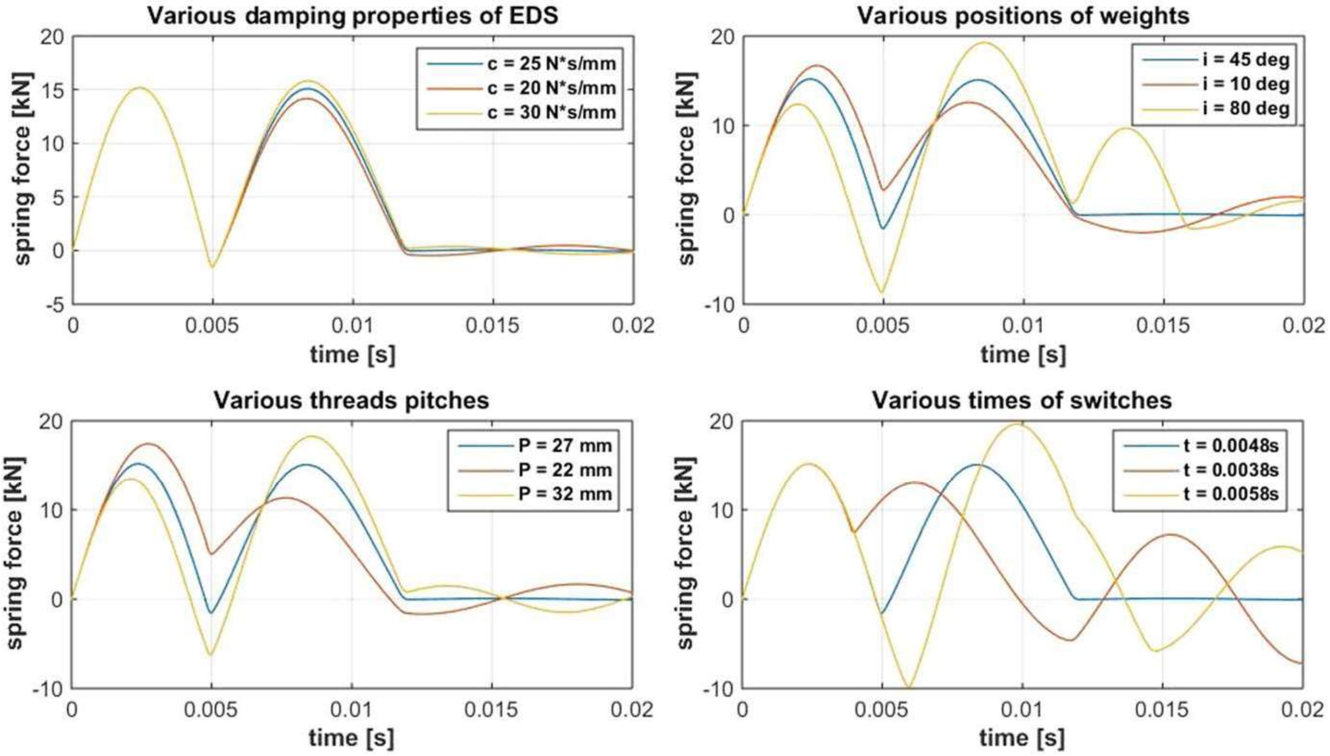

Figure 11. Influence of selected parameters on the performance of the SPIN-MAN.

determined time period, interface 1 and interface 2 are released in time intervals lasting 0.2 milliseconds. After that, interface 3 is blocked in 0.4 milliseconds, which is the beginning of the second phase of SPIN-MAN's operation. The switching of the blockers results in the rotation of the combined middle and outer rings, as well as in rotation of the inner ring (in the opposite direction). The velocity difference between the parts leads to a high energy dissipation on the EDS caused by the viscous connection.

Numerical simulation of the impact absorption process with the SPIN-MAN device has been performed in ADAMS multibody simulation software. Obtained results are presented below.

Angular velocities of rings resulting from the assumed control strategy are shown in figure 4. The switching actions clearly modify the rotation modes of the rings. Phase 2 mainly contributes to the energy dissipation process, due to the difference in angular velocities between ring 1 (inner) and ring 2 (middle).

At the beginning of phase 2, the inertia of the outer ring combined with the middle one is large enough to pull the inner ring and cause its rotary motion in the direction of the outer ring rotation. Such a situation corresponds to the return movement of the rod with the sleeve. In figure 5 vertical displacements are shown.

The residual motion that can be noticed on the graphs results mainly from the high sensitivity of the shock-absorber response to the geometric and mass parameters, whose proper selection is difficult. It is worth mentioning that in the case of friction on threads all residual motions are damped very fast.

As shown in figure 6, the proposed spin management technique leads to 50% reduction of the maximum impact force (compared to the passive reception by the spring–mass system). Also, it is important, that the implementation of the control allowed the total reduction of the force to be achieved in about 12 milliseconds after the impact occurrence.

Graphs of the kinetic energy of the mass (figure 7) and the potential energy of the spring (figure 8) show that the energy of the mass is transferred to the spring but not entirely. In the second phase, curves of the energy changes are smoother because of the damping on the second interface that makes the system response slower.

Besides accumulation in the spring, the kinetic energy of the mass is converted into the rotational energy of the outer ring in phase 1. In phase 2, the energy accumulated in the outer ring is used in the process of combining it with the middle ring and the remaining energy is dissipated in the EDS. Simultaneously, the residual kinetic energy of the mass is converted into the rotary motion of the inner ring which moves in the opposite direction to the outer ring.

Figure 9 presents a graph that shows the energy flows inside the SPIN-MAN. The participation of interface 2 in the energy dissipation process is dominant in the proposed control strategy. Nevertheless the energy dissipated in interface 3 during the process of combining the outer ring together with the middle ring is also significant.

In order to verify the accuracy of the conducted analysis, the energy balance for the whole system is computed and shown in figure 10. According to the graphs, over 60% of the impact energy is stored in the rotating ring during phase 1 and dissipated in phase 2 which lasts about 7 milliseconds. The

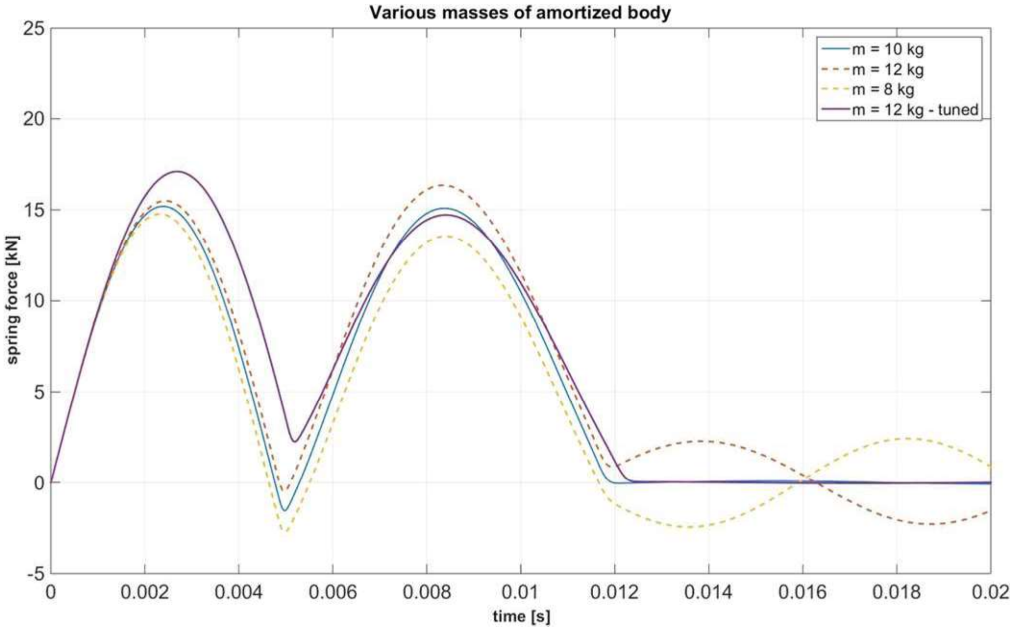

Figure 12. Influence of the mass of amortized body on the performance of the SPIN-MAN and its adaptation capabilities.

sharp jump of the dissipated energy in the fifth millisecond is related to the process of switching to the second mode of operation, which is needed to achieve the rotation of the elements in the opposite directions. For the presented solution, the construction of the blocker for interface 3, which should endure the absorption of such an amount of energy, would be a significant challenge. However, the results are very promising.

Mass distribution, geometry of threads and viscous damping coefficient were selected via a trial and error method to find the quasi-optimal solution in reference to the minimization of the maximum value of the impact force loading considered system. In order to evaluate the influence of particular parameters on performance of the SPIN-MAN device, a sensitivity study of the shock-absorber operation was performed. In a model of a tuned shock-absorber, damping, mass, geometry and time parameters were changed and influence of their changes on the impact force reduction was observed. Results of such simulations are shown as graphs in figure 11.

According to presented results, there are a number of parameters that can significantly affect efficiency of the device. Influence of change of one parameter can be compensated by repeated selection of other parameters. The thus manufactured shock-absorber composed of components with incorrect parameters can be tuned by changing the weights' position.

Adjustment of weights' position that changes the inertia moment of outer ring can also be used to adapt the SPIN-MAN to different impact excitations. To ensure the reader that changing of the inertia moment is really effective, simulations of the shock-absorber amortizing bodies of different masses were performed and results are shown in figure 12. Graphs present the system response of the device tuned for the mass of 10 kg and amortizing bodies of 8 kg and 12 kg, and also a response of the device after a change of weights' position to the 80° of inclination angle and change of moments of blockers' switches—tuning the shock-absorber for mass of 12 kg.

Presented results show that compensation of various parameters changes can be obtained by the appropriate selection of the time moments when blockers' switching on/off actions will be performed and by the adjustment of weights' position changing moment of inertia of the outer ring. In that way inaccuracies in manufacturing of the device as well as variable loading conditions may not significantly affect the performance of discussed shock-absorber.

4. Conclusions

This paper introduces a concept of impact energy management and demonstrates its application in a case-study using the SPIN-MAN device, which is a novel shock-absorber invented in the Institute of Fundamental Technological Research (IPPT PAN). The presented numerical results were carried out using the Adams multibody simulation software. The spin management as a control concept based on switching on/off actuators shows its potential for extremely effective shock absorption. The results of numerical simulations demonstrate that it is possible to reduce significantly the maximum loads caused by the impact. Moreover, it can be achieved in a very short time. Numerical modeling of the problem shows high sensitivity of simulation results to modifications of parameters such as mass distribution, pitch of threads, damping coefficients or timing of blockers' switching on/off actions. Therefore, development of effective numerical tools able to solve automatically optimal design problem can be challenging.

It should be mentioned that further investigations are necessary. They should be conducted in the fields of not discussed technical problems like time delays, stability of control algorithms, various criteria of optimization (e.g., minimization of mass accelerations) and methodology which should be used during phase of designing and adjusting device to external loads. The results of conducted simulations show that among the set of device parameters the crucial ones are thread pitches and moment of inertia of outer ring. While thread pitches, after manufacturing the device, cannot be changed, it is still possible to adjust properties of the device to external load by changing the moment of inertia of outer ring by repositioning additional weights. In further analysis we plan to investigate more deeply the influence of friction on threads whose consideration in this paper was omitted. Friction affects the efficiency of the device but it is assumed that it can be compensated by the adjustment of control strategy—timing of blockers' switches and/or by changes in the inertia moment of outer ring. The biggest challenge in preparation of the SPIN-MAN prototype seems to be design and manufacturing process of very fast and robust blockers and implementation of the control strategies performed on these blockers. In reference to the operation of blockers, friction apart from extending the time of accelerating the outer ring and as a result decreasing the amount of accumulated energy, may also cause a decrease of local loads on the components' connections during combining them by the blockers.

There are a large number of design and manufacturing challenges. Nevertheless, first attempts of constructing the components for the SPIN-MAN demonstrator (to be reported in the next paper) look optimistic.

Concluding, several challenging issues should be still addressed:

- Development of techniques for real time impact load identification (see [14]).
- Development of switchable actuators for the SPIN-MAN.
- Development of a mathematical model describing the SPIN-MAN functionalities.
- Formulation of optimal design problem for various applications of the SPIN-MAN and development of optimization algorithms.
- Adjustment of the SPIN-MAN construction allowing system restoration.
- Adjustment of the SPIN-MAN construction allowing effective mitigation of the system response for the harmonic excitations.
- Development of control strategies for system restoration after each impact of series of impacts or each period of harmonic excitations.
- Solving the problem of most effective reception of repetitive impacts.
- Solving the problem of smoothest reception of forced vibration.

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Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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Article D

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Development of control systems for fluid-based adaptive impact absorbers

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ABSTRACT

The paper presents development, evaluation and comparison of various control systems for adaptive fluid-based absorbers serving for absorption of the impact loading. The investigations concern two competitive approaches: i) standard control systems with single determination of the optimal system path based on identified impact conditions, and ii) newly-developed control systems with on-line determination and update of the system path during the process. It is revealed that low robustness of the standard control systems to imprecise impact identification and unknown disturbances results from the assumed path-determination approach and utilized simple path-tracking methods. The proposed solution to this problem is application of the innovative control systems, which utilize Automatic Path Finding and Automatic Path Update algorithms based on full kinematic feedback as well as Hybrid Path Tracking method dedicated for fluid-based absorbers. The introduced approach to absorber control is used to develop three different self-adaptive systems of increasing complexity and robustness. The favourable capabilities of proposed systems including no need for impact identification, high robustness against force disturbances and reduction of leakages influence are proved. Detailed discussion is presented using the illustrative example of single-chamber adaptive pneumatic shock-absorber mitigating impact loading.

1. Introduction

The concept of Adaptive Impact Absorption [1] had emerged as a response to unfavourable mechanical characteristics and dissipative properties of passive shock-absorbing systems operating in a wide range of impact loadings. Although typical passive systems are usually subjected to various impact excitations, their design process often assumes optimization for typical expected or average loads, e.g. by topological methods [2,3]. As a result, passive systems often cannot respond in a fully satisfactory way to all possible impact scenarios. In particular, in the case of low-energy impact the classical passive elastic–plastic rate-independent dissipater generates excessive resistance force, which results in too high value of impacting object deceleration, stopping the object by using only a small part of available stroke [4] and possible final rebound from the obstacle (the case of impact energy absorbed but not entirely dissipated). On the contrary, in the case of high-energy impact the standard passive absorber generates insufficient resistance force, which results in inadequate value of impacting object deceleration and failure in stopping the object before the end of available absorber stroke (the case of energy not fully absorbed and dissipated). Moreover, in case of passive impact absorbers with rate-dependent mechanical characteristics

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such as spring-dashpot systems, hydraulic or pneumatic dampers the dynamic response depends not only on impact energy but separately on the mass and velocity of the impacting object. Both parameters affect the time-history of generated resistance force and the impacting object deceleration, the utilized absorber stroke as well as the total amount of dissipated energy and the corresponding object rebound. Thus, totally passive hydraulic or pneumatic absorbers respond in a desired way exclusively to a single impact scenario defined by mass and velocity of the impacting object.

In contrast, the concept of Adaptive Impact Absorption [5–7] postulates design of dissipative systems which possess the ability of adaptation to a given impact loading, for instance impact of the object of given mass and initial velocity. The adaptation is executed either by preliminary adjustment or on-line control of the selected elements of impact absorbing system (structural fuses). The process is conducted in semi-active way, i.e. without direct introduction of external forces and corresponding additional energy into impact absorbing system. The objective of adaptation is to dissipate the entire impact energy by using the lowest value of generated force and the lowest value of the corresponding deceleration of the impacting object. Thus, from the point of view of impact-absorbing system the positive influence of successfully determined and implemented adaptation process is twofold. At first, the system gains the most favorable dissipative characteristics for typical expected impact loading. Secondly, this most beneficial characteristics is obtained for the entire wide range of possible impact loads.

The adaptation may be performed for both the single unidirectional dissipater and the complex energy-dissipating structure. In particular, in simple problems of impact absorption the most typical option is application of single- or double-chamber fluid-based dampers equipped with various types of controllable valves, i.e., hydraulic [8–11] or pneumatic [12–16] dampers equipped with fast piezo-electric or magneto-strictive mechanical valves as well as dampers based on magneto-rheological [17–19] or electro-rheological [20] fluids. In such case the control of valve operation determines forces generated by dissipater, its dissipative characteristics and corresponding system response to impact excitation. In turn, in more advanced problems of impact absorption the energy absorbing structure may take various complex forms such as skeletal structure equipped with controllable dampers of a proper type. In this case, the system of embedded dissipaters not only controls local values of generated forces and local dissipative characteristics but also significantly influences global response of the entire impact absorbing structure and its global dissipative properties [21,22]. Thus, introduced control allows to adapt the impact-absorbing structure to impact excitation of various mass, velocity, location and direction.

The separate attention should be focused on the development of adaptation methods and corresponding control strategies. So far, most of the research was focused on realization of the adaptation strategies to excitation known a priori at the beginning of the impact absorption process. In case of hydraulic or pneumatic dampers these strategies were divided into two separate actions. The first action was determination of feasible optimal response of the damper by using known parameters of the impact loading and limitations resulting from system construction ("path-finding problem"). It was executed only once, just before or at the very beginning of the impact and reference path was not updated during the energy dissipation process. The second stage of system operation was realization of previously determined optimal reference response of the damper by using proper control of valve opening and corresponding fluid flow during the impact absorption process ("path-tracking problem"). It was executed either by open-loop control systems with valve opening dependent directly on impact parameters and time [14] or closed-loop control systems with feedback signal based on actual level of generated force or actual level of impacting object deceleration [10,14,15,16]. Another recently developed control strategy providing optimal system response for identified impact conditions was proper reshaping of the orifice along the length of the cylinder. Such approach allows for reconfiguration of the system at the beginning of impact reception and its further passive operation providing high performance comparable with on-line controlled absorbers [23]. Similarly, passive damper operating in the way comparable with semi-active devices was developed and its velocity and displacement dependent performance was investigated in [24].

Previous numerical and experimental studies had proved that the above control approach works well and provides optimal adaptation of the impact absorbing system in case of precise identification of the impact loading and optimal prediction of the operational conditions. Despite the fact that impact identification process is often affected by measurement errors [25] and random system disturbances as well as changes of system parameters under impact loading are common, such effects were neglected in previous research works. The current paper fills this gap in the literature by presenting detailed evaluation of the influence of possible inaccuracies in identification of mass and velocity of the impacting object, disturbances in resistance force generated by the absorber (e.g. occurrence of additional elastic or damping forces) and disturbances of thermodynamic processes (e.g. possible fluid leakages). The investigations are conducted for standard open-loop and closed-loop control strategies combined with continuous and bang-bang control methods.

After recognition of the unsatisfactory robustness of the control systems based on single system path determination in the case of inexact impact identification and random disturbances, the authors propose utterly new approach to the problem of control of adaptive fluid-based impact absorbers. In this method the optimal reference response (system path) is initially determined and further updated at each control step by using full information about actual piston kinematics and the condition of the optimality for the remaining part of the process ("Automatic Path Finding" and "Automatic Path Update"). Further, determined optimal reference response is obtained by dedicated "Hybrid Path Tracking" technique, which combines precise control of the generated force resulting from pressure difference and bang-bang compensation of force disturbances. The above two-stage procedure is sequentially repeated for each control step. Since path-determination strategy is based on actual state of the system it automatically takes into account all previously occurring phenomena including unknown impact excitation and random disturbances. Thus, the proposed control system possesses build-in capability of automatic adaptation to all possible conditions and object equipped with such control system can be classified as **self-adaptive system**.

2. Adaptive Impact Absorption using fluid-based absorbers

2.1. General model of adaptive fluid-based absorber

For the sake of clarity we consider the simplest possible class of controllable impact absorbers, which contain single fluid chamber formed by rigid lateral walls and mobile piston (Fig. 1). The chamber is filled with hydraulic or pneumatic fluid and it is equipped with controllable valve enabling fluid outflow. The absorber is subjected to vertical impact loading of external rigid object, which causes downward piston motion, increase of fluid pressure and generation of time-dependent resistance force. During impact the opening of the valve is modified in order to influence the rate of fluid outflow and to control the value of generated force. The objective of the absorber is to dissipate the entire impact energy and to optimally mitigate the impact loading.

In order to precisely define and analyse the problem of Adaptive Impact Absorption we start with formulation of general mathematical model of the absorber. For this purpose we treat the problem as one-dimensional and we consider displacement of the impacting object and the piston as a single mechanical degree of freedom. Moreover, we assume that impact velocity is relatively low and it does not cause significant wave propagation effects in a given volume of fluid. Thus, for the thermodynamic modelling of fluid we apply the Uniform Pressure Method (UPM), which assumes that fluid pressure, temperature and density are uniform in the entire fluid chamber. Consequently, the corresponding thermodynamic model of the fluid will be governed by ordinary differential equations [26,27].

The mathematical model of the considered system is described by equation of motion of the impacting object, equation of mass or volume balance, equation of thermodynamic energy balance, algebraic equation of state of the fluid and algebraic equation defining the valve flow. The equation of motion of the impacting object takes the form:

$$M\ddot{u} - Mg + (p - p_{ext})A + F_{el} + F_{visc} + F_{fr} + F_D^{BOT} - F_D^{TOP} = 0 \tag{1}$$

where M is the mass of the impacting object, u is actual displacement of the object, g is the acceleration of gravity, p is pressure inside the cylinder, p_{ext} is external pressure, A is internal area of the piston, F_{el}, F_{visc} and F_{fr} are additional elastic, viscous and frictional forces generated by the absorber, F_D^{BOT} and F_D^{TOP} are bottom and top contact forces between piston and cylinder ends. The balance of fluid volume takes a classical form:

$$\dot{V} + \beta(p,T)V\dot{p} - \alpha(p,T)V\dot{T} = Q_V \tag{2a}$$

where V is the total volume of the fluid, $\beta(p,T)$ is the compressibility coefficient while $\alpha(p,T)$ is the thermal expansion coefficient and both of these coefficients depend, in general, on pressure p and temperature T of the fluid. Moreover, r.h.s. term Q_V is the volumetric inflow rate into the considered fluid chamber, which is negative in considered case of fluid outflow. The above equation of volume balance is usually used for fluids of small compressibility, while for strongly compressible fluids it is often replaced by balance of fluid mass, which takes a simple form:

$$\dot{m} = Q_m \tag{2b}$$

where m is mass of the fluid while Q_m is mass inflow rate into the fluid chamber. The equation of thermodynamic energy balance indicates the equality of added enthalpy, change of internal energy and work done by the fluid. In general case when both fluid inflow and outflow occur it takes a general form:

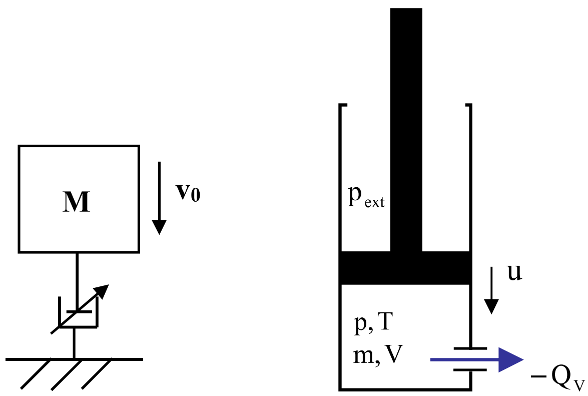

Fig. 1. Considered impact absorption problem involving adaptive shock-absorber.

$$\dot{m}c_pT_v + Q_Vp_v - Q_V\alpha T_vp_v = \dot{m}\left[c_pT - \alpha p\rho^{-1}T + p\rho^{-1}(\beta p - \alpha T)\right] + mc_p\dot{T} - \alpha pV\dot{T} + V\dot{p}(\beta p - \alpha T) + p\dot{V} \tag{3a}$$

where the index v indicates the parameters of the fluid flowing through the valve, which in case of fluid inflow are different than parameters of the fluid inside the considered chamber. The c_p is the constant pressure heat capacity, ρ is density of the fluid, while other quantities were already defined before. In case of no fluid inflow the equation takes the simplified form:

$$Q_Vp = \dot{m}p\rho^{-1}(\beta p - \alpha T) + mc_p\dot{T} - \alpha pV\dot{T} + V\dot{p}(\beta p - \alpha T) + p\dot{V} \tag{3b}$$

and, eventually, by utilizing the equation of volume balance is reduced to:

$$0 = \dot{m}p\rho^{-1}(\beta p - \alpha T) + mc_p\dot{T} - V\dot{p}\alpha T \tag{3c}$$

The system of differential Eqs. (1–3) has to be complemented by equation of state of the fluid which can be expressed either by mass and volume or, alternatively, by density of the fluid and it provides algebraic relation between its thermodynamic parameters in a general form:

$$f(p, T, m, V) = 0 \text{ or } f(p, T, \rho) = 0 \tag{4}$$

Finally, the last element of the model is definition of the volumetric or mass flow rate of the fluid through the valve, which typically depends on pressures on both sides of the valve, the temperature of the inflowing fluid, as well as, the actual area of the valve:

$$Q_{V/m} = f(p, p_{ext}, T_v, A_v(t)) \tag{5}$$

The above system of equations complemented by the definition of time-dependent area of the valve $A_v(t)$ and proper initial conditions allows to determine response of the impact absorber equipped with controllable valve to impact excitation.

2.2. Formulation of the AIA control problem for fluid-based absorber

Further we define the problem of Adaptive Impact Absorption for one degree-of-freedom mechanical system including rigid object and adaptive single-chamber fluid-based impact absorber with controllable valve (Fig. 2). Adaptation of the system to various masses and velocities of the impacting object is performed by the control of valve opening during the period of impact absorption. The objective of adaptive impact absorbing system is to dissipate the entire impact energy within available stroke of the absorber. Simultaneously, optimal process of impact mitigation is defined by minimization of the reaction force generated by the absorber and minimization of the corresponding deceleration of the impacting object.

Consequently, in the mathematical formulation of the AIA problem we search for the function describing change of valve opening in time $A_v(t)$, which satisfies equality condition imposed on the amount of dissipated energy and additional optimality condition based either upon minimization of the maximal value of force or maximization of the absorber efficiency. Thus, the classical AIA problem has two force-based formulations. The first mathematical formulation takes the form:

$$\text{Find } A_v\,(t) \text{ such that } \int_d \vec{F}d\vec{s} = E_{imp} \text{ and } \max F(t) \text{ is minimal} \tag{6a}$$

where F is the reaction force generated by the absorber, d is the stroke of the absorber, while E_{imp} is the value of impact energy (i.e. kinetic energy of the impacting object in case when gravitation force is omitted). In turn, the second formulation reads:

$$\text{Find } A_v(t) \text{ such that } \int_d \vec{F}d\vec{s} = E_{imp} \text{ and } \frac{\int_d \vec{F}d\vec{s}}{F_{max}d} \text{ is maximal} \tag{6b}$$

where the absorber efficiency is the ratio of dissipated energy to the hypothetical energy which can be dissipated by using the constant level of force of maximal obtained value. Let us note that the first formulation is a classical min–max problem, which involves maximal value of force in time and minimum searched over time-variable valve opening. In turn, the second formulation includes maximal value of force in denominator of the objective function and therefore it is a max-max problem. However, according to the equality condition of energy dissipation the numerator of the objective function is a constant and thus both formulations of the AIA problem are fully equivalent.

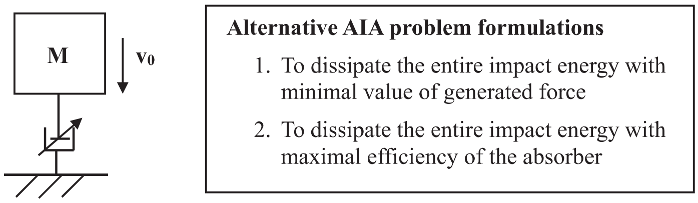

Fig. 2. Formulation of the Adaptive Impact Absorption problem for one degree-of-freedom system.

Theoretical, absolutely optimal solution of the AIA problem is obtained for the ideal absorber which generates constant force equal E_{imp}/d during the entire stroke d. In such case the equality condition concerning energy dissipation is identically met, the operations of finding the maximum over time and integration over stroke are trivial, and the second objective function indicating absorber's efficiency achieves its maximal theoretical value equal to one. Obviously the limitations resulting from properties of the fluid and characteristics of the valve cause that such favorable solution is not possible to be obtained.

2.3. Standard solution of the AIA control problem for fluid-based absorber

The standard solution of the above stated AIA control problem is based on the knowledge of impact parameters, i.e. mass and velocity of the impacting object, and it is divided into two separate steps. The first step is solution of the path-finding problem aimed at transition of the considered system from the initial state of dynamic equilibrium to the final state of static equilibrium using minimal value of generated force (determination of the optimal reference response). It is executed before the impact absorption process and it is based on finding optimal feasible change of reaction force in terms of displacement $F_{feas}(u)$, which satisfies previously formulated optimality criteria:

$$\text{Find } F_{feas}(u) \text{ such that } \int_d \vec{F}_{feas} d\vec{s} = E_{imp} \text{ and } \max F_{feas}(u) \text{ is minimal} \tag{7}$$

The above formulation includes feasible reaction forces $F_{feas}(u)$, which can be generated by the absorber of given mechanical properties. In particular, this constraint takes into account the maximal possible rate of force increase at the beginning of the process, which is obtained with a closed valve. As a result the optimal reference change of reaction force includes the initial stage of force increase and the subsequent stage when the force remains constant until the end of absorber stroke (cf. Sect. 3.2). Determined change of the reaction force unambiguously defines the corresponding optimal reference response of the system including acceleration, velocity and displacement of the impacting object. Let us stress the fact that in standard AIA approach the above path-finding problem is solved before impact and it is not updated during the energy dissipation process.

The second step is realization of the previously determined reference change of the reaction force, i.e. the solution of the path-tracking problem:

$$\text{Find } A_\nu(t) \text{ such that } \int_d \vec{F} d\vec{s} = E_{imp} \text{ and } \int_{t_0}^{t_k} (F - F_{feas}^{opt})^2 dt \text{ is minimal} \tag{8}$$

The path-tracking can be executed either with the use of feed forward or feedback control strategy. In case of open-loop system the change of valve opening can be determined by using the inverse dynamics approach, i.e. introduction of the calculated reaction force into the set of governing equations, determination of system kinematics from the equation of motion, calculation of the corresponding change of mass of the fluid from the equation of state and finally determination of the required opening of the valve from the flow equation (cf. Sect. 3.2). In turn, in case of closed-loop system based on bang-bang control method either actual level of pressure or actual level of generated force (or equivalently deceleration) can be used as a feedback signal.

The standard design of the AIA control system for fluid-based absorber directly corresponds to the above procedure of optimal valve control strategy determination. Consequently, the system has to be equipped with dedicated impact identification subsystem [25], which is activated at the beginning of the impact process in order to identify parameters and energy of the impact loading (Fig. 3). Moreover, it contains the subsystem which determines optimal change of the reaction force and the subsystem which controls valve opening in order to obtain reference system response. Let us indicate that from the point of view of the control system design the determination of the optimal reaction force is considered as external open loop, while control of the valve opening is considered as internal (either open or closed) loop.

3. Analysis of the standard AIA control strategies for pneumatic absorber

3.1. Basic assumptions of the analysis

In order to be more specific we will analyse single-chamber adaptive pneumatic shock absorber, which can be considered as a special case of the general fluid-based absorber defined in Section 2.1. The model of pneumatic absorber is obtained by replacement of general fluid state equation (4) by ideal gas law and by introduction of proper corresponding definitions of compressibility coefficient β and thermal expansion coefficient α:

$$V = \frac{mRT}{p}, \tag{9a}$$

$$\beta = -\frac{1}{V}\frac{\partial V}{\partial p} = p^{-1}, \tag{9b}$$

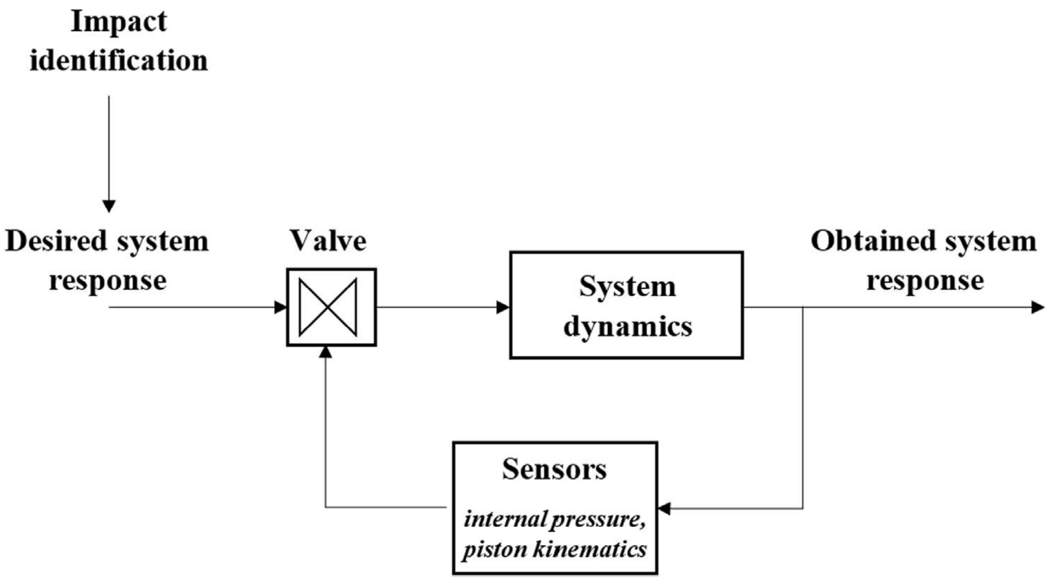

Fig. 3. Typical structure of control closed-loop system dedicated for AIA device.

$$\alpha = \frac{1}{V}\frac{\partial V}{\partial T} = T^{-1} \tag{9c}$$

Consequently, the general equation of energy balance and the simplified equation of energy balance for the case of fluid outflow take the form:

$$\dot{m}c_pT_v = \dot{m}c_vT + mc_v\dot{T} + p\dot{V} \tag{10a}$$

$$0 = mc_p\dot{T} - V\dot{p} \tag{10b}$$

The Eq. (10b) can be combined with ideal gas law (9a) and integrated analytically, which yields well-known adiabatic equations of state defining relations between thermodynamic parameters of the gas. Finally, it is assumed that during the entire process the ratio of pressure inside the cylinder and external pressure is relatively low and the valve flow can be described by the classical isentropic flow model.

Let us further consider horizontally located, frictionless single-chamber adaptive pneumatic absorber subjected to impact of the object of mass M and velocity v_0 and analyse exclusively the stage of impact absorption associated with gas outflow. The corresponding mathematical model consists of equation of motion of the impacting object, adiabatic equation of gas state and equation describing isentropic flow through the valve, cf. also [28,29]:

$$M\ddot{u} + (p - p_{ext})A + F_{el} + F_{visc} + F_D^{BOT} = 0 \tag{11}$$

$$\frac{pV^\kappa}{m^\kappa} = \text{const.}, \quad \frac{Tm^{1-\kappa}}{V^{1-\kappa}} = \text{const.} \tag{12}$$

$$\dot{m} = -A_v(t)\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa - 1}}\frac{p}{\sqrt{RT(t)}} \tag{13}$$

$$\text{IC}: u(0) = 0, \dot{u}(0) = v_0, m(0) = m_0, p(0) = p_0, V(0) = V_0 \tag{14}$$

The control of the impact absorption process will be executed by modification of the valve opening $A_v(t)$ during the impact process. The following standard control strategies based on preliminary identification of the impact conditions (i.e. mass and velocity of the impacting object) will be analysed:

- open loop system with optimally determined continuous change of valve opening at the very beginning of the process,
- closed loop system with pressure feedback and bang-bang control method,
- closed loop system with acceleration feedback and bang-bang control method.

Table 1
Parameters assumed in the numerical study.

Suspended mass [kg]	Initial velocity of the mass [m/s]	Initial internal pressure [kPa]	Operational gas	Absorber's diameter [m]
10	10	151.95 (1.5 atm)	compressed air	0.2
Length of the chamber [m]	Available stroke for system with continuous control [m]		Available stroke for system with bang-bang control [m]	
0.2	0.2		0.18	

For the sake of clarity of the discussion on the performance of different control strategies, the shock-absorber and impact excitation will be the same in all presented simulations, see Table 1. The reduction of the available stroke in case of bang-bang control strategy is caused by predicted unfavorable response of the pneumatic absorber at the final stage of impact absorption, when the piston approaches cylinder bottom. In such situation the volume of gas remaining inside the cylinder is small and opening of the valve for a single control period causes large drop of pressure and oscillation of the generated pneumatic force. Consequently, the pneumatic force cannot be maintained constant until the end of the stroke. For that reason it was assumed that control system based on bang-bang control method should dissipate energy within 90% of entire absorber's stroke. In order to provide complete scope of parameters applied in conducted simulations, it should be mentioned that minimal and maximal area of valve opening was assumed to be equal $0\mathrm{cm}^2$ and $15\mathrm{cm}^2$, respectively.

3.2. Description of standard control strategies

All standard control techniques start with determination of the optimal feasible change of the reaction force, which satisfies the condition of impact energy dissipation and minimization of the maximal force and deceleration value. In case of pneumatic absorber the main physical restriction is the limitation of maximal possible rate of generated force increase during initial gas compression, which is achieved when the valve remains closed. In turn, near the end of absorber stroke the maximal possible rate of pneumatic force decrease, depending on the maximal valve opening, is very high. Thus, the corresponding optimal control strategy includes the first stage when pneumatic force gradually increases, the stage when pneumatic force is maintained at possibly constant level until the end of absorber stroke and the final stage when it is instantly decreased to zero value.

The condition of dissipating the entire impact energy allows to calculate the required duration of the subsequent stages of the process. The corresponding mathematical condition results from the integration of the equation of cylinder motion separately for the first stage corresponding to piston displacement $(0, u_x)$ and the second stage corresponding to displacement (u_x, u_{max}) when force is maintained approximately constant:

$$\frac{1}{2}Mv_0^2 = p_0 A \frac{h_t{}^{\kappa}}{(\kappa-1)(h_t-u_x)^{\kappa-1}} - p_0 A \frac{h_t}{\kappa-1} - p_{ext}Au_x + p_0\frac{h_t^{\kappa}}{(h_t-u_x)^{\kappa}}A(u_{max}-u_x) - p_{ext}A(u_{max}-u_x) \tag{15}$$

The above equation enables finding piston displacement, velocity and acceleration (u_x, v_x and a_x) as well as corresponding gas pressure and temperature (p_x and T_x), which indicate the end of the stage of force increase and beginning of the stage of force remaining constant. Let us stress the fact that in such approach the system path is determined for given impact parameters. Thus, the control system is expected to work correctly exclusively in the case of precise identification of the mass and velocity of the impacting object.

The applied control systems utilize either open- or closed-loop strategies for maintaining approximately constant level of force during the second stage of the process. In case of **open loop control system**, the optimal change of the outflow area will be determined by using the so called inverse dynamics approach. In this method, constant value of generated pneumatic force and zero values of elastic and viscous forces treated as system disturbances are assumed. Thus, the equation of motion of the piston motion (Eq. (11)) allows to determine optimal kinematics of the piston during the second stage of the process in terms of time:

$$a_{opt} = -\frac{A}{M}(p_x - p_{ext}), \quad v_{opt}(t) = v_x - \frac{A}{M}(p_x - p_{ext})(t - t_x) \tag{16a, b}$$

$$u_{opt}(t) = u_x + v_x(t - t_x) - \frac{A}{2M}(p_x - p_{ext})(t - t_x)^2 \tag{16c}$$

Then, we use adiabatic equation of state (Eq. 12a), which indicates that change of the mass of gas $m(t)$ has to be proportional to the change of its volume $V(t)$ and equal:

$$m(t) = m_0\frac{V(t)}{V_x} = m_0\frac{u_{max} - u_{opt}(t)}{u_{max} - u_x} \tag{17}$$

so the mass flow rate through the valve depends linearly on piston velocity:

$$\dot{m}_{opt}(t) = -\frac{m_0}{u_{max} - u_x} v_{opt}(t) \tag{18}$$

Moreover, adiabatic equation of state (12b) indicates that temperature of the gas T(t) remains constant and equal T_x. Thus, required change of valve opening can be determined directly from (Eq. (13)) where optimal change of piston velocity is introduced:

$$A_v(t) = \frac{\frac{m_0}{u_{max}-u_x} v_{opt}(t)}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_x}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\frac{p_x}{\sqrt{RT_x}}} = \frac{\frac{m_0}{u_{max}-u_x}\left[v_x - \frac{A}{M}(p_x - p_{ext})(t - t_x)\right]}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_x}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\frac{p_x}{\sqrt{RT_x}}} \tag{19}$$

Let us note that the above formula depends explicitly on the mass of the impacting object, while dependence on impact velocity is embedded in the variables v_x, p_x, T_x, which are determined from the Eq. (15) involving both the mass and the velocity of the impacting object. The above change of valve opening is maintained until the entire stroke of the absorber is utilized and the entire impact energy is absorbed. Then, at the final stage of the process the valve is fully opened in order to reduce generated pneumatic force to zero.

The above derivation clearly indicates that the above control strategy requires mathematical model of the flow through the valve, which can be treated as disadvantage since such model is not always exact for the whole range of operational conditions. Moreover, the open-loop path-tracking strategy includes idealization of operational conditions, i.e. the lack of disturbances of generated force and perfect chamber sealing. Thus, the proposed path-tracking method is expected to work correctly only in case when generated force is purely pneumatic and unexpected gas leakages do not occur.

In case of **closed loop control system with pressure feedback** and bang-bang control method the approximately constant level of generated force during the second stage of the impact absorption process is obtained by switching the valve between two extreme positions. In the simplest control strategy the valve is closed when actual pressure of gas is lower than optimal pressure p_x and is opened when actual pressure is higher than optimal pressure p_x. In a more advance control scheme the level of pressure tolerance $(p_x - \Delta p^{tol}, p_x + \Delta p^{tol})$ is introduced and the control $A^i_v(t)$ defined in time period $(t_i, t_{i+1}\rangle$ is conducted according to the following rule, also known as Schmidt regulator:

$$\begin{aligned}
&\text{if } p(t_i) \le p_x - \Delta p^{tol} \text{ then } A^i_v(t) = A^{min}_v \\
&\text{if } p(t_i) \ge p_x + \Delta p^{tol} \text{ then } A^i_v(t) = A^{max}_v \\
&\text{if } p_x - \Delta p^{tol} < p(t_i) < p_x + \Delta p^{tol} \text{ then } A^i_v(t) = A^{i-1}_v(t)
\end{aligned} \tag{20}$$

Finally, when assumed part of the absorber stroke is utilized and the entire impact energy is absorbed, the valve is fully open in order to reduce generated pneumatic force. Let us note that bang-bang regulation method does not require the exact mathematical model of the valve flow but on the other hand its disadvantage is the requirement of commutative switching the valve between two extreme positions, which results in high oscillations of pneumatic force close to the end of cylinder stroke. Moreover, the closed loop path-tracking strategy includes idealization of operational conditions, i.e. lack of disturbances such as additional elastic or viscous forces. Thus, the control system is expected to work correctly only in case when generated force is purely pneumatic. In turn, in contrast to open-loop system, the small leakages of gas can be compensated by applied path-tracking method.

The **closed loop control system with acceleration feedback** and bang-bang control method is in principle similar to previously considered system with pressure feedback. In this case the level of generated force during the second stage of the impact absorption process is maintained constant by switching the valve between two extreme positions when actual piston deceleration is higher or lower than the optimal value a_x. In a more advance control scheme the level of acceleration tolerance $(a_x - \Delta a^{tol}, a_x + \Delta a^{tol})$ is assumed and the path-tracking is governed by the following rule:

$$\begin{aligned}
&\text{if } a(t_i) \le a_x - \Delta a^{tol} \text{ then} A^i_v(t) = A^{min}_v \\
&\text{if } a(t_i) \ge a_x + \Delta a^{tol} \text{ then } A^i_v(t) = A^{max}_v \\
&\text{if } a_x - \Delta a^{tol} < a(t_i) < a_x + \Delta a^{tol} \text{ then } A^i_v(t) = A^{i-1}_v(t)
\end{aligned} \tag{21}$$

Finally, at the end of the process the valve is fully open in order to reduce generated pneumatic force. The similarity to control method involving pressure feedback is revealed by requirement of commutative opening and closing of the valve, which results in high oscillations of pneumatic force close to the end of cylinder stroke. On the other hand, the current path-tracking method does not include any idealization of the operational conditions. Thus, the control system is expected to compensate small disturbances in generated force and gas leakage.

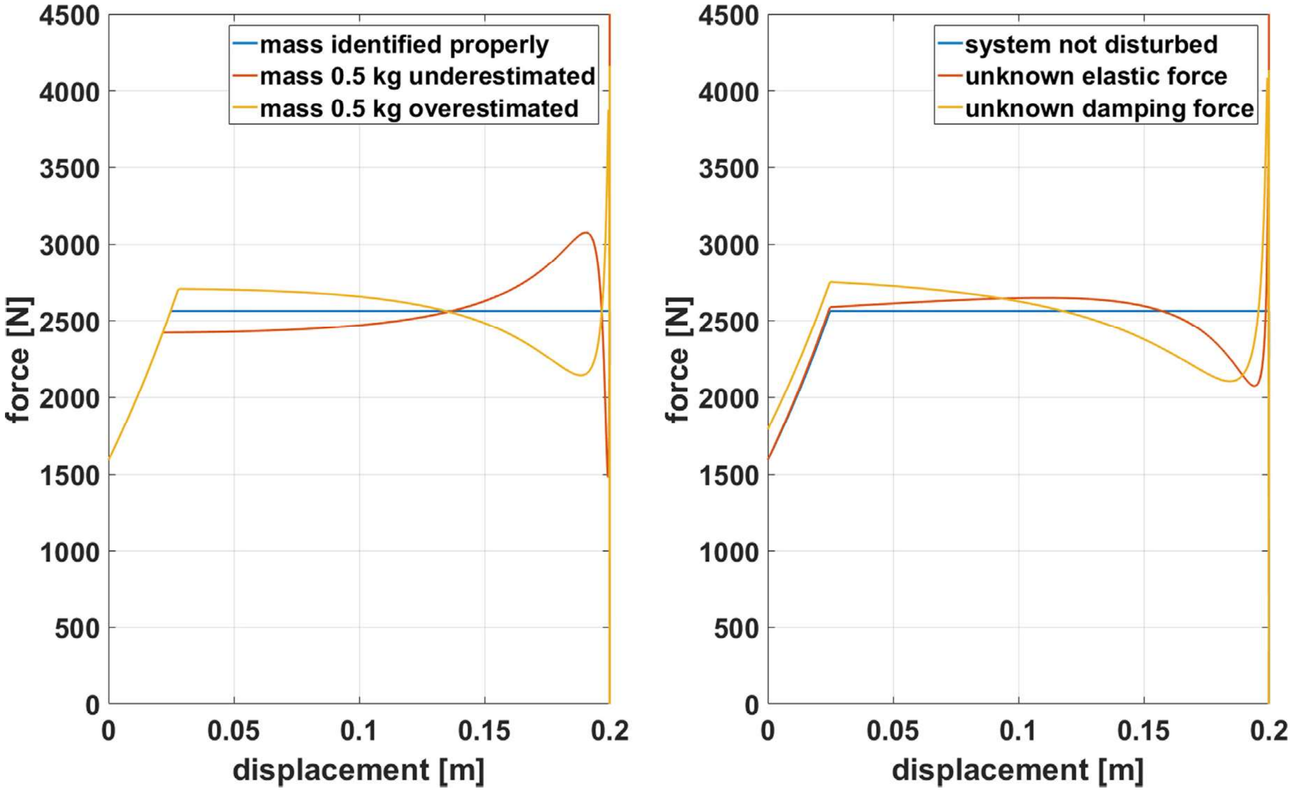

Fig. 4. Influence of inaccurate impact identification and unknown disturbance on dynamic response of adaptive pneumatic cylinder with open-loop control system.

3.3. Numerical evaluation of the robustness of the standard AIA control strategies

In order to test the robustness of the above control systems we will artificially introduce inaccuracies in identification of impact conditions, disturbances caused by additional elastic and damping forces as well as unpredicted leakage of the gas at certain period of the process.

At first, we will analyse the influence of inaccurate mass identification and unknown disturbances on the performance of adaptive pneumatic absorber with open-loop control system. The response of such system in case of 0.5 kg under- and overestimation of impacting mass is shown in Fig. 4a. The application of open loop control causes that initial stage of pressure increase is too short (mass underestimated) or too long (mass overestimated). During the second stage the path-tracking procedure is destabilized and non-monotonic change of generated pneumatic force is observed. Consequently, the value of resistance force is not optimally minimized. Moreover, the entire impact energy is not dissipated before the end of the absorber stroke, which results in impact against cylinder bottom and corresponding peak of generated force. Introduction of unknown disturbances such as additional elastic and viscous forces in the system also implies its non-optimal operation. The open-loop path-tracking system is destabilized and the absorption process is accomplished with failure. Similarly, as in previous case the piston is not stopped before the end of the stroke and its contact with the absorber bottom results in substantial peak of generated force (Fig. 4b).

Secondly, we investigate the influence of inaccurate mass prediction and unknown disturbances on the performance of adaptive pneumatic absorber equipped with closed-loop control systems with pressure or acceleration feedback (which in case of lack of disturbances operate in identical manner). Imprecise prediction of the mass in closed-loop system with pressure feedback or acceleration feedback also results in their non-optimal operation presented in Fig. 5. In contrast to open-loop system, the pneumatic force is maintained approximately constant during the second stage of the process. However, in case of mass overestimation the non-optimal, excessively large level of the reaction force leads to stopping of the object before the end of the absorber stroke and its final rebound. In turn, in case of mass underestimation the non-optimal, insufficient level of the reaction force does not allow to stop the object before the end of cylinder stroke, which results in impact of the piston against bumper placed before cylinder bottom (in system with bang-bang control the bumper is located at 90% of the absorber's stroke).

Introduction of unknown disturbances in the closed-loop system with pressure feedback implies non-optimal operation presented in Fig. 6a. In system with pressure feedback the total value of force during the second stage does not remain constant, i.e. it slightly increases in system with additional elastic force and decreases in system with additional viscous force. However, the total value of force is always higher than in undisturbed system, which results in stopping the impacting object

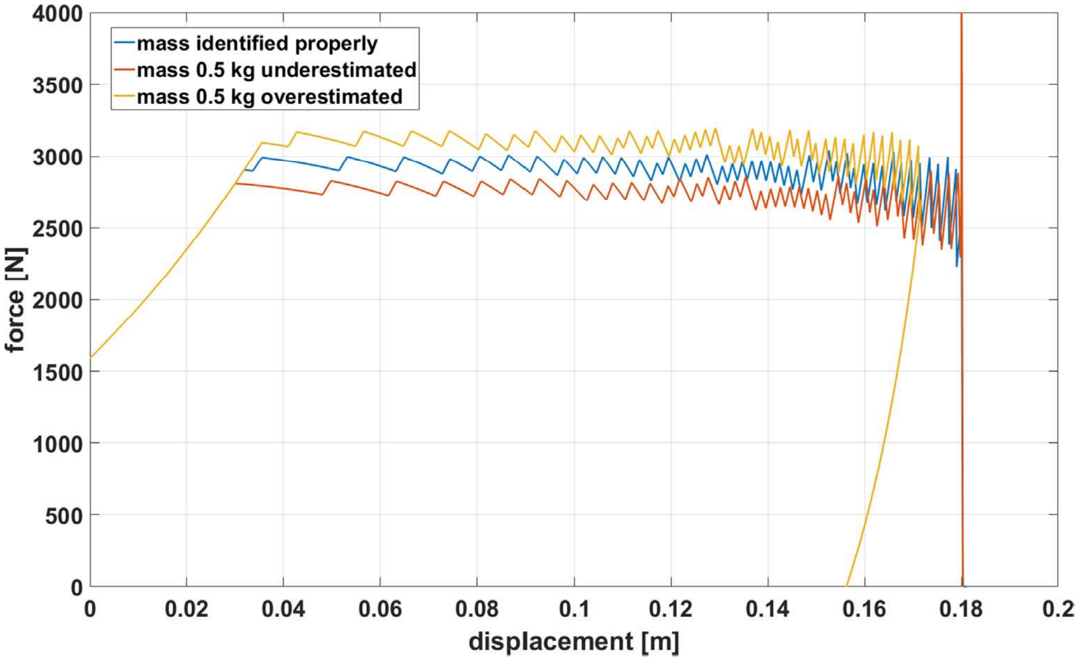

Fig. 5. Influence of inaccurate mass identification on dynamic response of closed loop system with pressure or acceleration feedback.

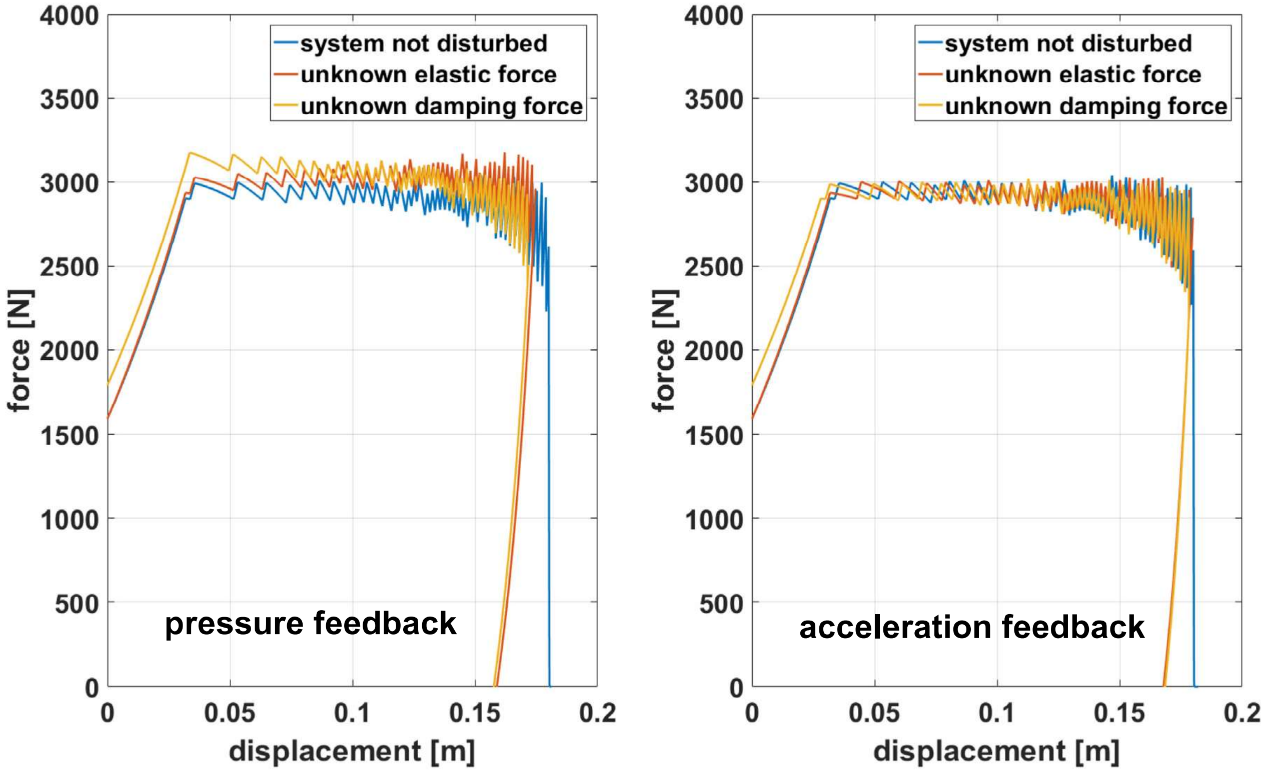

Fig. 6. Influence of imprecise identification on the closed loop system response with pressure and acceleration feedback.

before the end of cylinder stroke and its final rebound. The performance of the system with acceleration feedback is better because the total value of generated force is maintained at approximately constant level during the second stage of the process. Nevertheless, the final rebound of the impacting object is observed in case of both types of disturbances. In case of disturbance by viscous force the final rebound is the result of higher level of total generated force during the initial part of the process. In turn, in the case of disturbance by elastic force the rebound is caused by accumulated elastic energy.

4. Improved AIA control strategies for pneumatic absorber

4.1. Design of the proposed control system

The lack of robustness of the previously considered standard AIA control systems resulted from three basic facts. At first, determination of the optimal system path was based on identification of the mass and velocity of the impacting object and it was not modified during the process. Secondly, the applied path-tracking methods had assumed idealized model of absorber with a full knowledge about characteristics of generated force and thus the system could not compensate possible force disturbances. Thirdly, the path-tracking methods had involved two extreme approaches to the valve flow control: i) open-loop control based on idealized, analytical model of the flow resulting in immediate system destabilization in case of disturbance or ii) bang-bang control assuming complete lack of knowledge about the flow model resulting in high force oscillations close to the end of the stroke.

In order to omit the above disadvantages of the standard AIA systems the following strict requirements regarding the design of improved AIA control strategies were formulated:

- The control system does not use any external subsystem for identification of the parameters of the impacting object including its mass and velocity; the process of adaptation to impact excitation has to be performed without the knowledge of such parameters.
- The prevailing part of the force generated by the absorber is pneumatic force resulting from pressure difference, however the additional unknown elastic and viscous forces can also appear and they have to be compensated by the control system.
- The gas release is executed mainly by the valve and flow can be described by isentropic flow model, however disturbances of the flow intensity or additional gas leakages can also appear and they have to be compensated by the control system.

In order to provide the possibility of elaboration of the control system which fulfils the above requirements two important assumptions were adopted. It was assumed that the entire kinematics of the piston as well as the value of actual pressure are precisely measured at each stage of the process. Moreover, it was assumed that control of the valve flow is performed without any delay, which is obtained e.g. by using piezoelectric valve.

General scheme of the improved AIA control system that is able to meet the listed above requirements is presented in Fig. 7. The innovation in control system design is obtained by applications of two control loops, which are able to realize the tasks of Automatic Path Planning and Hybrid Path Tracking.

The **external loop** of the control system utilizes full information about actual system kinematics and the condition of global kinematic optimality for the remaining part of the process for the purpose of automatic planning of the system path. The Automatic Path Planning (APP) is performed at each control step and it can be divided into Automatic Path Finding (APF), which operates during the first stage of the process when generated force increases and Automatic Path Update (APU), which operates during the second stage when generated force is maintained approximately constant. The APF algorithm supersedes the impact identification procedure and allows to adjust duration of the first stage of the process and corresponding level of force generated during the second stage. In turn, the APU algorithm is responsible for automatic update of the reference system response in case of force disturbances or unexpected gas leakages.

The **internal loop** of the control system is aimed at realization of determined optimal system path with the use of dedicated Hybrid Path Tracking algorithm. The path-tracking is based on closed-loop valve control with feedback to actual system kinematics and pressure value and it operates differently when actual kinematic response is inside and outside the

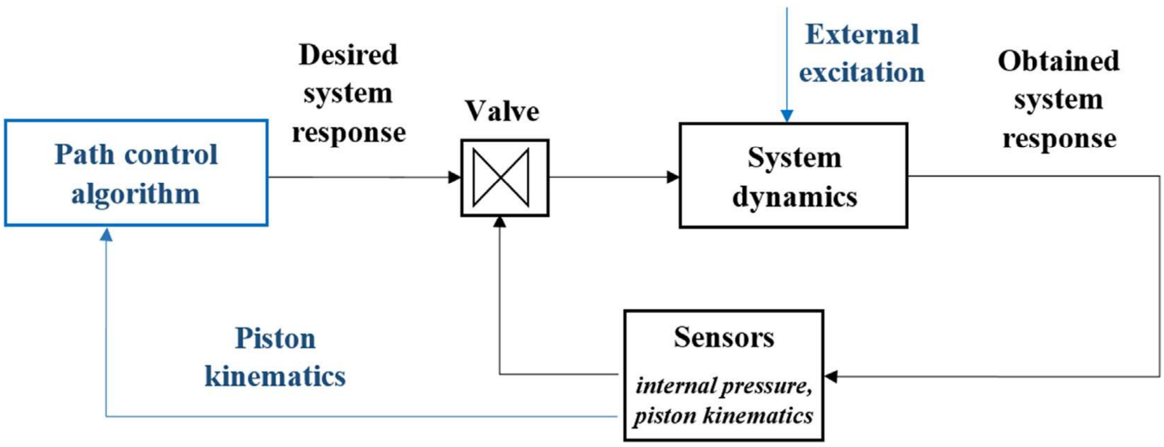

Fig. 7. General scheme of the proposed improved AIA control system with full kinematic feedback.

assumed range of tolerance of determined optimal system path. In case when the actual response is within the tolerance level the tracking method effectively utilizes definition of generated pneumatic force and assumed model of the valve flow in order to provide approximately constant value of impacting object deceleration. In turn, when actual kinematic response temporarily jumps outside the tolerance level the bang-bang control of the valve is applied in order to compensate actually occurring system disturbances.

Application of Automatic Path Planning enhanced by Hybrid Path Tracking provides self-adaptive performance of the impact absorbing system and ensures its response close to optimal even in case of harsh disturbances and change of system parameters. The self-adaptation is here understood as automatic sequential determination of the optimal system path at consequent stages of the process and its efficient realization by feed-forward control system. Let us note that from the point of view of control theory the proposed approach can be compared to sequential application of Optimal Control, Model Predictive Control [30] or rather combination of Model Predictive Control and feed forward control such as applied in [31]. On the other hand, the proposed method is significantly different than typical control methods applied for pneumatic systems, e.g. [32–34].

In further discussion, we will examine and compare the robustness of three different self-adaptive systems utilizing the concept of Automatic Path Planning:

- system with Automatic Path Finding (APF) utilizing Simplified Path Tracking (SPT) algorithm with continuous valve operation;
- system with Automatic Path Finding (APF) utilizing dedicated Hybrid Path Tracking (HPT) algorithm with continuous and bang-bang valve operation;
- system with Automatic Path Finding (APF) and Automatic Path Update (APU) utilizing dedicated Hybrid Path Tracking (HPT) algorithm.

4.2. Automatic Path Finding

The APF algorithm utilizes full kinematic feedback, i.e. measurement of the actual displacement, velocity and acceleration of the piston. The introduced condition of kinematic optimality of the remaining part of the process is based on comparison of the actual level of piston deceleration with the constant level of deceleration required to decrease actual piston velocity to zero using the remaining stroke of the absorber. The case of smaller value of actual deceleration indicates the requirement of continuation of the first stage of absorption process and thus increasing the value of generated pneumatic force. In turn, the case of larger value of actual deceleration indicates possibility of starting the second stage of absorption process and maintaining the constant level of force. Therefore, the kinematic condition checked at every control step of the first stage of the process takes the form:

$$\text{if}\,|a(t_i)| < a_x - \Delta a^{tol} \text{ then } A_v^i(t) = A_v^{min} \tag{22a}$$

$$\text{if}\,|a(t_i)| \geq a_x - \Delta a^{tol} \text{ then } A_v(t) \text{ determined by path tracking algorithm}$$

$$\text{where } a_x = \frac{v_x^2}{2(u_{max} - u_x)}, v_x = v(t_x), u = u(t_x)$$

$$t_x = t_i \text{ such that the condition } |a(t_i)| > \frac{v_x^2}{2(u_{max} - u_x)} \text{ is met for the first time.} \tag{22b}$$

The above kinematic condition has the similar function as the energy condition indicating dissipation of the entire impact energy (Eq.15) used in classical AIA control systems. It enables finding time t_x, piston displacement, velocity and acceleration (u_x, v_x and a_x) as well as corresponding gas pressure and temperature (p_x and T_x), when the second stage of the impact absorption process begins. Nevertheless, the application of the kinematic condition is much more robust and efficient. At first, the use of such condition does not require preliminary identification of the mass and initial velocity of the impacting object, which are compulsory for the application of the energy condition. Secondly, the kinematic condition takes into account additional forces and possible leakages of gas during the first stage of impact, while energy condition assumes generation of the ideal pneumatic force. As it is shown in Fig. 8 the Automatic Path Finding allows to determine constant level of force generated by the absorber, which provides dissipation of the entire impact energy and utilization of the entire absorber stroke.

4.3. Automatic Path Finding with Simplified Path Tracking

The first and the simplest analysed control system utilizes **Automatic Path Finding** and **Simplified Path Tracking (SPT)**. The SPT algorithm is aimed at maintaining approximately constant level of acceleration a_x, by controlling exclusively the corresponding pneumatic force. It is based on closed-loop control strategy with feedback to initial level of pressure and actual kinematics of the piston. In order to determine the corresponding control law we utilize the adiabatic equation of state

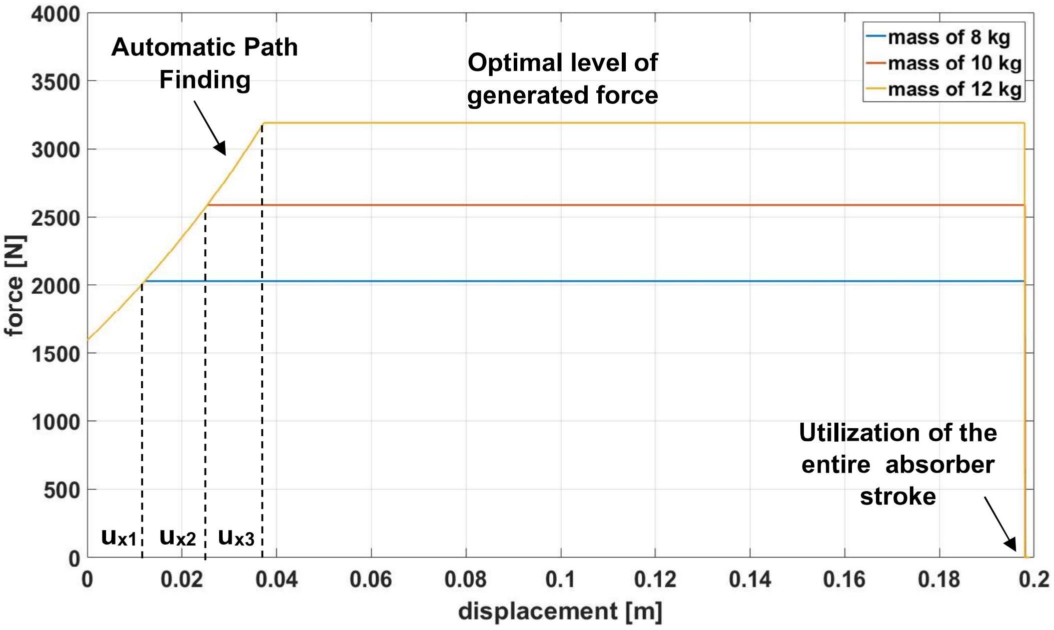

Fig. 8. Automatic Path Finding with unknown mass and velocity of the impact in case of no additional disturbances in the system.

(Eq. 12a) defined for the instant t_x when condition (22b) is fulfilled and second stage of the impact absorption process begins. Obtained equation indicates that during the second stage the mass of gas m(t) has to be proportional to its actual volume V(t):

$$m_{opt}(t) = m_0 \frac{V(t)}{V_x} = m_0 \frac{u_{max} - u(t)}{u_{max} - u_x} \tag{23}$$

while the mass flow rate of gas through the valve has to be proportional to actual piston velocity:

$$\dot{m}_{opt}(t) = -\frac{m_0}{u_{max} - u_x} v(t) \tag{24}$$

Moreover, the adiabatic equation of state (Eq. 12b) indicates that temperature of the gas T(t) remains constant and equals T_x. Thus, the required change of valve opening in terms of actual velocity of the piston can be determined directly from Eq. (13):

$$A_v(t) = \frac{\frac{m_0}{u_{max}-u_x} v(t)}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_x}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\frac{p_x}{\sqrt{RT_x}}} \tag{25}$$

where $T_x = T_0(p_0/p_x)^{1/\kappa-1}$. The actual velocity of the piston during each control step for time period $\langle t_i,\ t_{i+1}\rangle$, can be approximated either by constant or linearly decreasing velocity defined by kinematic values determined at time t_i:

$$\overline{A^i_v}(t) \cong \frac{\frac{m_0}{u_{max}-u_x} v_i}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_x}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\frac{p_x}{\sqrt{RT_x}}} \quad \text{or} \quad \overline{A^i_v}(t) \cong \frac{\frac{m_0}{u_{max}-u_x} [v_i + a_i(t - t_i)]}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_x}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\frac{p_x}{\sqrt{RT_x}}} \tag{26a, b}$$

The formula (26a) indicates constant opening of the valve at each control step, while the formula (26b) indicates linear decrease of the valve opening at each control step. Let us note that although the above closed-loop control strategy seems similar to the previously presented open-loop strategy, they result in the same optimal response of the system only in case when no disturbances are present. The main advantage of discussed closed-loop system is the fact that disturbance by additional elastic or viscous force does not affect the process of maintaining constant pneumatic force.

Graphs visualizing influence of unknown force disturbances and sudden gas leakage on the performance of control system with APF and SPT are shown in Fig. 9a and Fig. 9b, respectively. Elastic and damping forces appearing in the system cause that total force generated by the absorber is not constant, the entire stroke is not utilized and dissipation process is not optimal. On the other hand, it can be clearly observed that total generated force is a sum of constant pneumatic force and disturbing force, which indicates that control system was not destabilized. In turn, the appearance of sudden leakage results in improper operation of the control system. After significant decrease of pneumatic force caused by leakage the

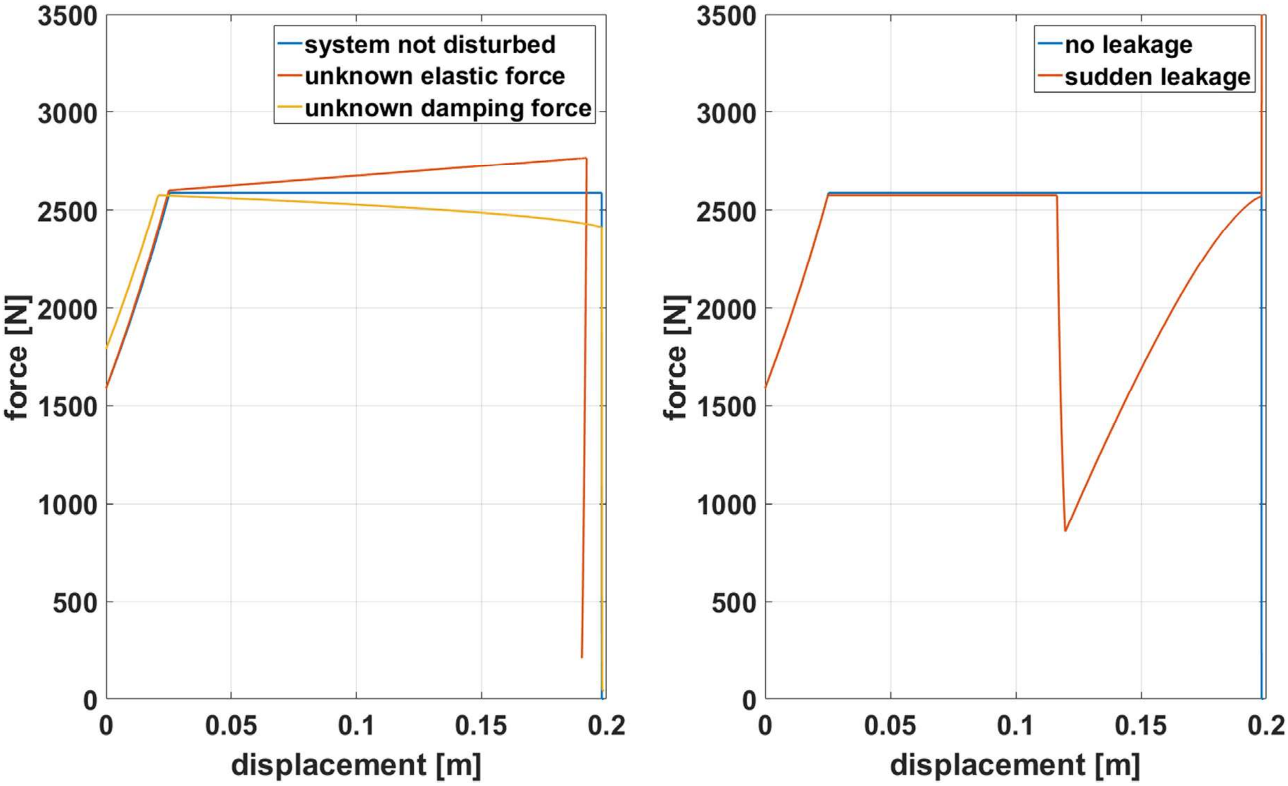

Fig. 9. Response of the system controlled with APF and SPT in case of unknown force disturbances and sudden gas leakage.

system slowly strives to obtain previously determined value of pneumatic force, which ends up with hitting the bumper near the absorber bottom because of insufficient impact absorption capability.

Eventually, it can be concluded that control system with Automatic Path Finding and Simplified Path Tracking algorithm eliminates two drawbacks of standard AIA systems - the requirement of impact identification and destabilization of the control system in case of force disturbances. Influence of fluid leakage is not sufficiently reduced so control system development is still required.

4.4. Automatic Path Finding with Hybrid Path Tracking

The second analysed control system, which can be considered as improved version of APF-based system discussed in previous section, is based on **Automatic Path Finding** and **Hybrid Path Tracking (HPT)**. The HPT algorithm utilizes both the continuous and bang-bang valve control in order to keep acceleration of the impacting object within the assumed range defined by critical accelerationa_x determined by APF algorithm (Eq. (22)) and assumed tolerance level Δa^{tol}. The proposed version of hybrid path-tracking utilizes the combination of:

- continuous control of valve opening applied within the range of acceleration tolerance, aimed at maintaining constant level of pneumatic force,
- bang-bang control applied when piston acceleration is temporarily outside the range of acceleration tolerance, aimed at compensation of force disturbance and gas leakage.

The applied algorithm of continuous valve control is slightly different than in previous considered case since it has to provide constant level of the actual pressure, not constant level of pressure determined at the beginning of the process. In order to determine the corresponding control law we again utilize the adiabatic equation of state (Eq. 12a), but now defined for the arbitrary time instant of the second stage of impact absorption process. This equation indicates that during the second stage the mass of gas m(t) has to be proportional to its actual volume V(t):

$$m_{opt}(t) = m_i \frac{V(t)}{V_i} = \frac{p_i}{RT_i} V(t) = \frac{p_i A}{RT_i}(u_{max} - u(t)) \tag{27}$$

while the mass flow rate of gas through the valve has to be proportional to actual piston velocity:

$$\dot{m}_{opt}(t) = -\frac{p_i A}{RT_i} v(t) \tag{28}$$

Thus, the required change of valve opening can be again determined directly from Eq. (13), in which actual velocity of the piston during each control period is approximated by linearly decreasing velocity:

$$A_v^i(t) = \frac{Av(t)}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_i}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_i}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\sqrt{RT_i}} \cong \frac{A[v_i + a_i(t - t_i)]}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_i}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_i}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\sqrt{RT_i}} \tag{29}$$

where $T_i = T_0(p_0/p_i)^{1/\kappa-1}$. According to above formula the continuous control law depends on kinematics of the piston, pressure and temperature of the gas at the previous control point. Thus, the complete close-loop control law for the second stage of the impact absorption process, which involves both the bang-bang and continuous control takes the form:

$$\begin{aligned}
&\text{if } |a(t_i)| \le a_x - \Delta a^{tol} \text{ then } A_v^i(t) = A_v^{min} \\
&\text{if } |a(t_i)| \ge a_x + \Delta a^{tol} \text{ then } A_v^i(t) = A_v^{max} \\
&\text{if } a_x - \Delta a^{tol} < |a(t_i)| < a_x + \Delta a^{tol} \text{ then } A_v^i(t) = \frac{A[v_i + a_i(t - t_i)]}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_i}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_i}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\sqrt{RT_i}}.
\end{aligned} \tag{30}$$

According to results of numerical simulations shown in Fig. 10 the Hybrid Path Tracking algorithm provides substantially higher robustness to disturbances than previously considered Simplified Path Tracking algorithm. In case of disturbance by additional elastic or viscous force the continuous part of the HPT algorithm maintains constant level of pneumatic force, but total value of generated force and object acceleration gradually change. Once they exceed the tolerance level the bang-bang part of the HPT algorithm is activated and the valve is instantly fully opened or closed in order to correct the actual value of acceleration.

Similarly, in case of sudden gas leakage the continuous part of the HPT control law is not able to maintain constant level of pressure so total pneumatic force and corresponding object deceleration decrease below the tolerance level. This effect is compensated by the bang-bang actions within the HPT algorithm, which rapidly closes the valve in order to increase the level of deceleration. Further, the control system tracks previously determined system path, which does not take into account the occurrence of gas leakage, the corresponding drop of absorber's resistance force and the corresponding decrease of impacting object deceleration. Such operation of the control system is incorrect since stopping the impacting object requires increased value of generated force and increased deceleration of the impacting object during the remaining part

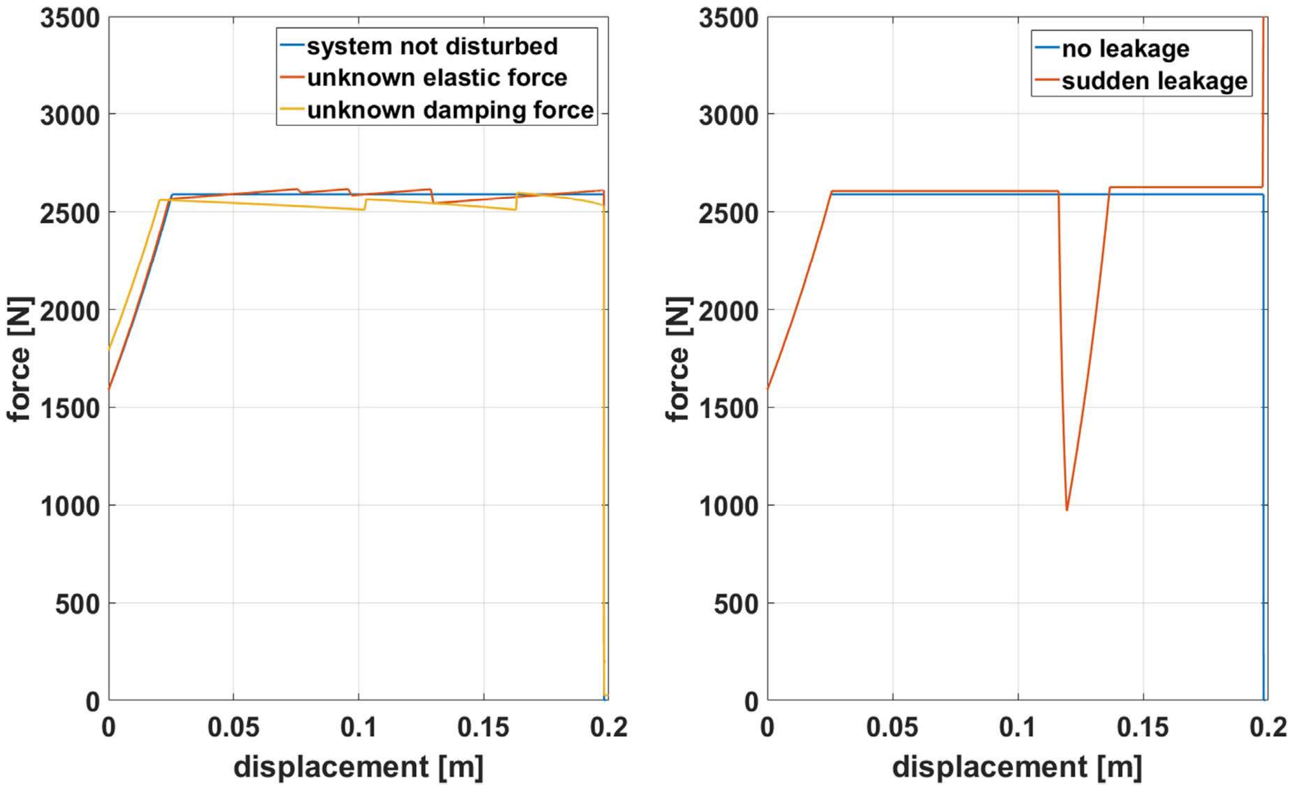

Fig. 10. Response of the system controlled with APF and HPT in case of unknown force disturbances and sudden gas leakage.

of the process. The consequence of tracking the original (not updated) system path is the lack of absorption of the entire impact energy by the pneumatic system and collision of the impacting object against cylinder bottom.

It can be concluded that AIA control system with Automatic Path Finding and Hybrid Path Tracking algorithm substitutes the functionality of impact identification and ensures robustness to force disturbance. Although the robustness to sudden gas leakage is increased in comparison to previous control system, it is still not satisfactory and further control system development is required.

4.5. Automatic Path Finding, Automatic Path Update and Hybrid Path Tracking

Eventually, the most advanced control system utilizing **Automatic Path Finding, Automatic Path Update** and **Hybrid Path Tracking** algorithms, is considered. The main difference in comparison to previous control systems is on-line adjustment of the reference system path during the process. Now, the reference acceleration will not be arbitrarily kept constant during the entire second part of the process but, if necessary, it will be modified depending on the actual kinematics of the impacting object. Such constructed control system has the ability of counteracting extremely harsh disturbances which cause substantial or long-lasting jumps from the assumed range of acceleration tolerance. This type of disturbances cannot be compensated simply by the return to the previously determined tolerance range since the amount of dissipated energy changes and previous optimal reference system path is no longer valid. Thus, the only possible solution is correction of the system path, which allows to preserve both the condition of impact energy dissipation and the condition of generated force optimality.

The proposed control system utilizes the combination of continuous and bang-bang control, similarly as previously analysed control system based exclusively on APF and HPT. However, the limits of tolerance range are defined by the kinematic condition utilizing actual values of piston velocity and displacement. Thus, the closed-loop control law executed after termination of the APF and the first iteration of the HPT takes the form:

$$\begin{aligned}
&\text{if } |a(t_i)| \le \frac{v(t_i)^2}{2(u_{max}-u(t_i))} - \Delta a^{tol} \text{ then } A_v^i(t) = A_v^{min} \\
&\text{if } |a(t_i)| \ge \frac{v(t_i)^2}{2(u_{max}-u(t_i))} + \Delta a^{tol} \text{ then } A_v^i(t) = A_v^{max} \\
&\text{if } \frac{v(t_i)^2}{2(u_{max}-u(t_i))} - \Delta a^{tol} < |a(t_i)| < \frac{v(t_i)^2}{2(u_{max}-u(t_i))} + \Delta a^{tol} \text{ then } A_v^i(t) = \frac{A[v_i + a_i(t-t_i)]}{\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_{ext}}{p_i}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{ext}}{p_i}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa-1}}\sqrt{RT_i}}
\end{aligned} \tag{31}$$

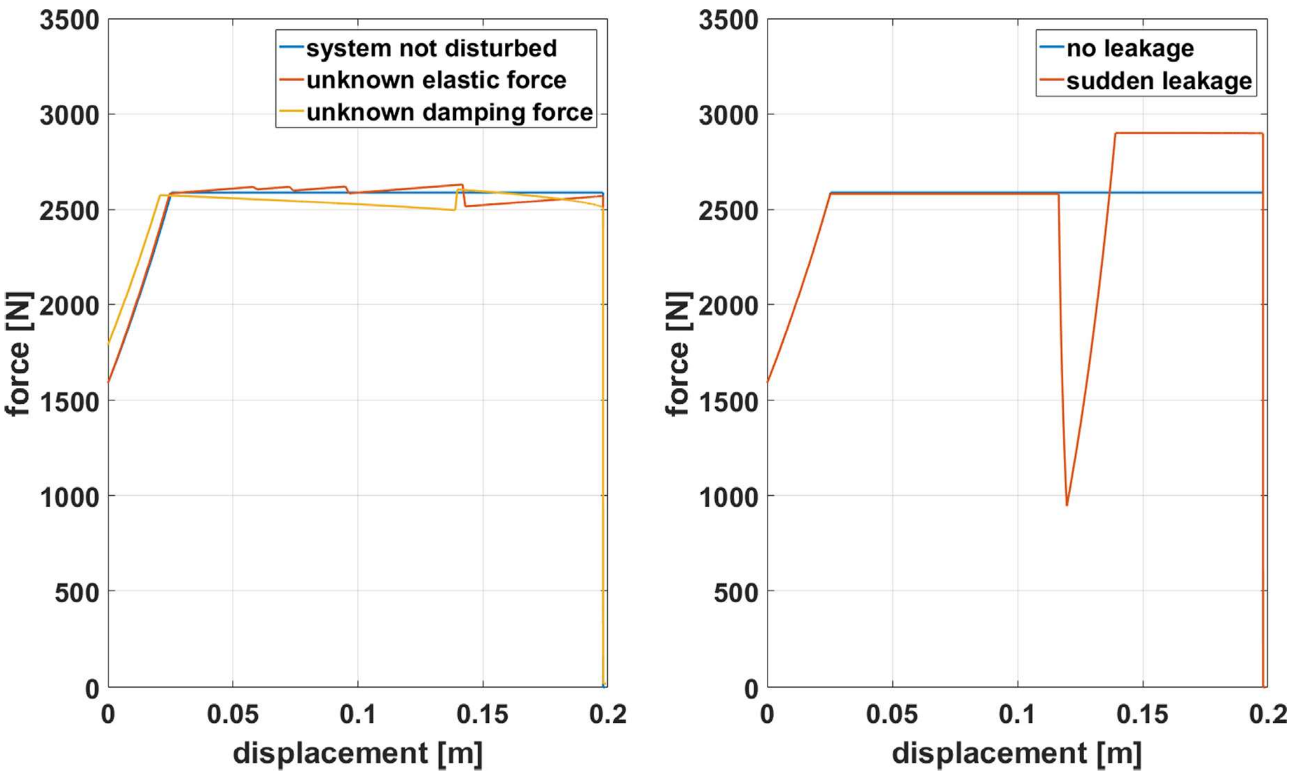

Fig. 11. Response of the control system with APF, APU and HPT in case of unknown force disturbances and sudden gas leakage.

The above control law describes coupled operation of the HPT and APU. Alternatively, the Eq. (30) can be used until the value of system deceleration remains in the tolerance range and when system optimality is lost the Eq. (31) is used to provide update of system path by recalculation of the actually optimal deceleration a_x.

The system with Automatic Path Finding, Automatic Path Update and Hybrid Path Tracking algorithms provides the highest robustness of all control systems considered so far. The results of simulations presented in Fig. 11a confirm that the system is robust to unknown force disturbances, similarly as system with APF and HPT but without APU.

The additional advantages of the system are revealed in case of extremely harsh disturbances causing substantial or long-lasting jumps from the assumed tolerance range when the update of the reference system path is inevitably required. In such case, the efficiency of the system based on combination of APF and HPT will significantly decrease, whereas system enhancement by APU will provide optimal system re-adaptation resulting in utilization of the entire stroke and the lowest possible level of deceleration. High robustness of the proposed control system can be observed during large unexpected gas leakage, which causes substantial decrease of the deceleration below the tolerance range (Fig. 11b). As a response to this situation, the system path is automatically updated with the use of kinematic optimality condition. The decrease of resistance force generated by pneumatic system and corresponding decrease of impacting object deceleration after the occurrence of leakage are compensated by the increase of resistance force and deceleration during the remaining part of the process. Since the control system tracks the updated system path, it provides dissipation of the entire impact energy and stopping the impacting object just before the cylinder bottom. The obtained maximal level of deceleration is larger than in no-disturbance case, however this is an inevitable consequence of the occurrence of adverse process of gas leakage.

Characteristic features of the introduced control system can be seen in Fig. 12, which presents how the valve area is adjusted in order to maintain the object deceleration within the tolerance range. The bang-bang actions are performed to compensate influence of force disturbances and sudden gas leakage. Each of these actions is associated with proper update of system path, which compensates the decrease of system efficiency. It should be pointed out that the system operation is based exclusively on measurements of the actual state of the system, without any external data about the excitation conditions, force disturbances and gas leakage.

The analysed control system based on Automatic Path Finding and Automatic Path Update with Hybrid Path Tracking fulfils all requirements for improved AIA control system formulated at the beginning of this section. The presented system can be considered as fully self-adaptive and extremely robust to misidentification of impact conditions and possible disturbances.

4.6. Practical implementation of the proposed control systems

There are two most important practical factors which may be critical in practical implementation of the proposed control systems for fluid-based absorbers:

1. the sampling frequency of measurements and control signal setting,
2. maximum opening of the valve applied in bang-bang control strategy.

Ad.1. The sampling frequency has the influence on detection of the optimal deceleration range by Automatic Path Finding (APF) algorithm, proper operation of the bang-bang control within Hybrid Path Tracking (HPT) and detection of the event of falling out from the deceleration tolerance range by Automatic Path Update (APU) algorithm. In the case of too small sam-

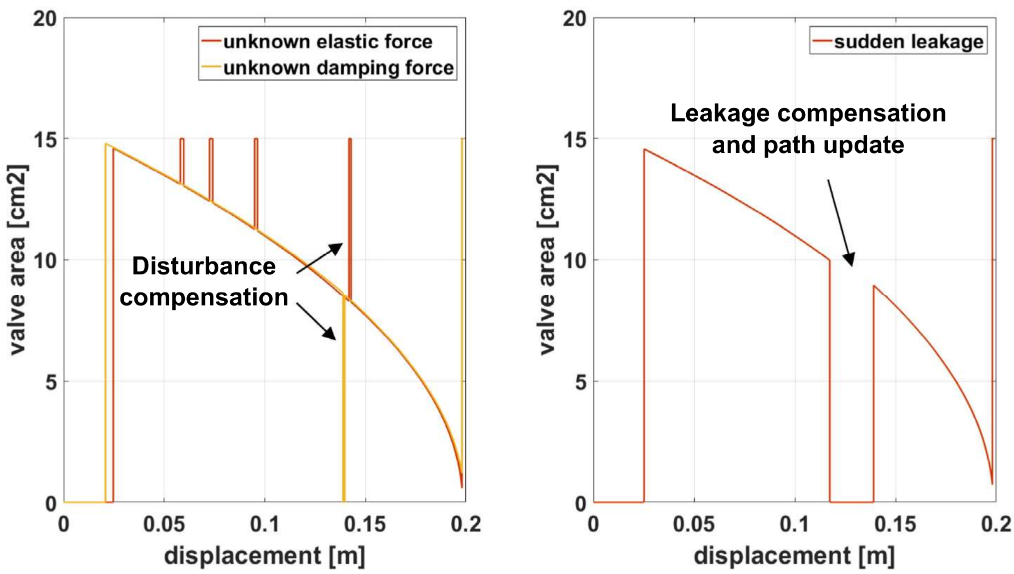

Fig. 12. Control (valve area) of the system based on the APF, APU and HPT in case of unknown force disturbances and sudden gas leakage.

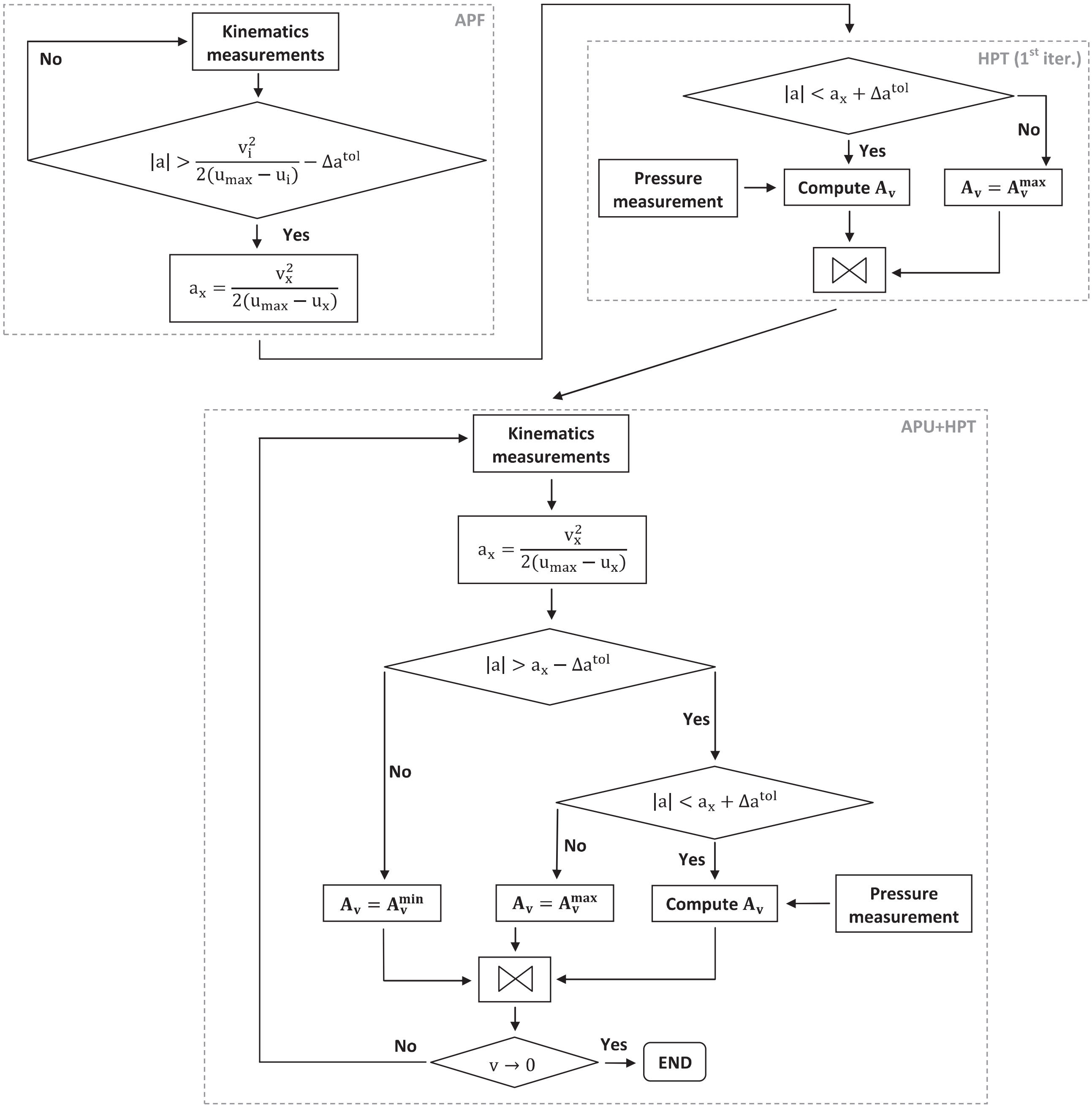

Fig. 13. Block diagram of the exemplary control procedure based on APF, APU and HPT.

Table 2
Comparison of analysed systems robustness to imprecise impact identification, unknown disturbances and change of system parameters (l – low, m – medium, h – high).

	Standard AIA systems			Improved AIA systems		
	Continuous open-loop control	On/off pressure feedback	On/off acceleration feedback	APF	APF + HPT	APF + APU + HPT
Impact identification	l	l	l	h	h	h
Disturbances	l	l/m	m	m	m/h	h
Fluid leakage	l	l/m	l/m	l	m	h

pling frequency the APF will not detect reaching the optimal deceleration range, but instead it will detect exceeding the upper tolerance level with certain delay. This will cause that the actual value of impacting object deceleration will be too high and optimality of the system will be lost. In the following part of the process too small sampling frequency will imply that the bang-bang control included in HPT will not be able to cause return of the impacting object deceleration to the optimal deceleration range, but instead it will generate its commutative jumps outside upper and lower tolerance levels. Moreover, the APU will detect a fall out from the optimal deceleration range with certain delay. Too late detection of exceeding the upper tolerance level will immediately generate too high value of impacting object deceleration, similarly as in the initial part of the process. In turn, too late detection of exceeding the lower deceleration tolerance level will cause too large drop of the actual deceleration value and the requirement of its excessive increase during the remaining part of the process.

Ad.2. The admissible maximum opening of the valve has the influence on the operation of the system when exceeding of the upper deceleration tolerance level is detected and HPT algorithm activates bang-bang control actions. Too large constant opening of the valve for the entire control step will not result in return of the actual impacting object deceleration to optimal deceleration range as required, but instead it will cause its drop below the lower tolerance level. This indicates that the applied control action is too intensive and the system is over-controlled. Such phenomenon is particularly distinct at the end of the impact absorption process when decrease of deceleration caused by the valve opening is not compensated by the increase of deceleration due to forward motion of the piston. Obviously, maximal allowable opening of the valve and required frequency of the system are interconnected with each other. For larger sampling frequency the amount of gas transferred within single control step is smaller so larger maximal opening of the valve can be applied.

The exemplary implementation of the control strategy involving Automatic Path Finding, Hybrid Path Tracking and Automatic Path Update can be illustrated by the block diagram of the step-by-step procedure, which can be directly executed by real-time control system (Fig. 13). The block-diagram has been divided into three main blocks. The first block indicates standalone operation of the APF algorithm, which determines the situation when lower deceleration tolerance level is exceeded and assigns the initial value of the optimal deceleration a_x. The second part of the diagram denotes first iteration of the HPT algorithm, which checks whether actual deceleration of the impacting object is below the upper tolerance level or not, and applies either optimal continuous change of valve opening or its maximal value, respectively. Eventually, the third block indicates joint operation of the APU and HPT algorithms. System operation starts with update of the optimal deceleration based on actual kinematics measurements. Further, the system sequentially checks the conditions if actual deceleration is above the lower tolerance level and below the upper tolerance level. If one of these conditions is violated either minimal or maximal opening of the valve is applied. In contrast, when both conditions are satisfied and actual deceleration is within the tolerance range, the optimal continuous change of valve opening is determined with the use of measured pressure value. The process is terminated when velocity of the impacting object approaches zero.

5. Conclusions

Presented study had revealed that standard control systems, developed for adaptive impact absorbing devices, are burdened with serious drawbacks including high sensitivity to precision of the impact identification and the lack of robustness to disturbances by additional forces and unexpected gas leakage. These disadvantages result from both mistakenly assumed strategy of single determination of the reference system path at the beginning of the process and improper choice of the path-tracking methods.

As a response, the authors had proposed several significant improvements of the control systems for adaptive shock-absorbers, which eliminate subsequent disadvantages of the standard approach. The versatility of the proposed control methods is provided by introduction of Automatic Path Planning method (Automatic Path Finding and Automatic Path Update) as well as application of dedicated Hybrid Path Tracking algorithm. The Automatic Path Planning allows for repeatable determination and update of the optimal reference path of the system at each control step by using full kinematic feedback. It automatically takes into account all phenomena and disturbances which had occurred in the system and provides their efficient compensation. In turn, the implemented Hybrid Path Tracking algorithm ensures improved capability of path realization by application of the continuous valve control based on definition of generated pneumatic force and isentropic flow model as well as bang-bang actions for compensation of possible disturbances.

Both improvements discussed above provide that the proposed control systems possess build-in capability of automatic adaptation to various impact conditions and disturbances and thus they can be classified as self-adaptive. In the ideal case of no disturbance they behave in the same manner as standard AIA control systems and optimally mitigate the impact loading, however without preliminary identification process. The advantages of the proposed control systems including their high robustness and re-adaptation capabilities are revealed during application in real mechanical devices, in which identification inaccuracies and disturbances are always present. The comparison of AIA control systems' robustness presented in Table 2 clearly proves the superiority the of proposed self-adaptive systems.

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Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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Hybrid Prediction Control for self-adaptive fluid-based shock-absorbers

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ABSTRACT

The paper covers detailed discussion on novel control system developed for adaptive fluid-based shock-absorbers serving for mitigation of unknown impact excitations. In order to provide complete independence of the control system from the loading conditions, the Hybrid Prediction Control (HPC) was elaborated. The proposed method is an extension of previously introduced kinematic feedback control which ensures optimal path finding, tracking and path update in case of high disturbance or sudden change of loading conditions. Implementation of the presented control system allows to obtain self-adaptive fluid-based absorbers providing robust impact mitigation. In contrast to previously developed methods of Adaptive Impact Absorption, the proposed control strategy does not require prior knowledge of impact excitation or its preliminary identification. The independence of applied control system from parameters of impact loading results in the capability of automatic path correction in the case of disturbance occurrence and re-adaptation to a number of subsequent impacts. The successful operation of the self-adaptive system is investigated with the use of numerical examples involving double-chamber pneumatic shock-absorber equipped with controllable valve. Efficiency of the HPC is proved by comparison with passive absorber as well as device equipped with adaptive and optimal control modules.

1. Introduction

Machines and engineering structures are often subjected to harmonic excitations as well as mechanical impacts. The impact absorption processes take very short time periods, typically at the level of several to tens of milliseconds. As a result it is still challenging to entirely replace the classical passive shock-absorbers by more efficient semi-active or active devices. Nevertheless, various theoretical methods for absorption of the impact loading were already proposed, proved by numerical simulations and tested experimentally. The significant research area involves the concept of Adaptive Impact Absorption (AIA) [1,2], which is in opposition to classical design methods based on passive energy absorbing structures. According to the AIA concept the optimal shock-absorbing system should provide adaptation to actual impact conditions and thus it should be equipped with system of sensors, the controller and active elements such as controllable valves. The objective of adaptation law is to dissipate the entire impact energy by using the lowest value of generated force and the lowest value of the corresponding deceleration of the impacting object.

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Abbreviations

AIA	Adaptive Impact Absorption
CMP	Control Mode Prediction
HPC	Hybrid Prediction Control
IDP	Inverse Dynamics Prediction
MIAC	Model Identification Adaptive Control
MPC	Model Predictive Control

List of symbols

A_1, A_2	cross-sectional area of the piston: 1 - on the side of compressed chamber, 2 - on the side of decompressed chamber
A_ν	effective area of the valve opening
c_p	constant pressure heat capacity
c_v	constant volume heat capacity
E_{imp}	energy of the impact
F_{abs}	reaction force of the absorber
F_D^{BOT}	contact force between piston and cylinder bottom
F_{dist}	disturbance force
F_D^{TOP}	contact force between piston and cylinder top
F_{ext}	external force exciting the system
F_{fr}	friction force
F_{max}	maximum value of absorber reaction force
F_{pneu}	pneumatic force
Q_m	mass flow rate
Q_v	volumetric flow rate
t_f	final time of the process
v_0	initial velocity of the impacting mass
η_{ae}	impact absorption efficiency
η_{se}	stroke efficiency
Δu_{tol}	tolerance for the final displacement of the piston
c	damping coefficient
d	stroke of the absorber
k	stiffness coefficient
m	mass of the gas
M	mass of the impacting object
p	pressure of the gas inside the absorber chamber
t	time
T	temperature of the gas
u	displacement of the piston
V	volume of the gas
α	thermal expansion coefficient
β	compressibility coefficient
κ	adiabatic exponent
ρ	density of the gas

Subscripts

0	relates to initial value of the quantity
1	relates to the quantities in compressed chamber
2	relates to quantities in decompressed chamber
i	number of the control step
ν	relates to quantities of the gas transferred through the valve
opt	relates to optimal value of the quantity

The research in the field of Adaptive Impact Absorption had concerned both the single shock-absorbers and more complex energy-absorbing skeletal structures. In both cases the attention was focused on practical applications of hydraulic as well as pneumatic shock-absorbers. In particular, the investigation of adaptive hydraulic dampers equipped with controllable mechanical valves was aimed at development of intelligent hydraulic bumper for mitigation of frontal collisions [3,4], elaboration and testing of semi-active landing gear [5,6], potential improvement of the adaptive landing gear [7], as well as detailed modelling and testing of the hydraulic viscous damper equipped with piezo-electric valve and PID controller [8]. In turn, the research concerning adaptive pneumatic dampers was oriented towards application of controllable pneumatic absorbers for mitigation of the dynamic response of buildings in seismic conditions [9,10], investigation of the air damping for heavy vehicles [11], application as systems for real-time mitigation of wind gusts [12] or components of landing gears for the small airplanes [13,14]. Application of various control methods to the suspensions based on magnetorheological fluids was discussed in [15]. The topical review includes examples of classical control systems, advanced modern controllers and hybrid control strategies, which are developed for commercial systems.

Another efficient absorber's designs were based on active control systems. The example is active vehicle suspension based on robust L_2 gain state-derivative feedback controller [16] and active control of a landing gear system equipped with an oleo-pneumatic shock absorber using robust linear quadratic regulator [17].

Focusing on the adaptation principles the authors distinguished two main categories of shock-absorbers, namely “adaptable systems” and “adaptive systems”. The adaptable systems utilize prior knowledge about the impact excitation. In result the optimal response of the absorber is calculated and corresponding control is applied. According to this procedure such systems are adaptive in terms of adaptation to the impact excitation. Nevertheless, system does not adapt to the changes of the process such as disturbances, changes of system parameters, next impact excitations. The example of adaptable system is the pneumatic absorber, developed previously by the authors, in which the optimal outflow of gas is provided by proper shaping of the orifice along the cylinder length [18]. The similar type of device is passive damper characterized by velocity and displacement dependent performance, described in [19].

The more sophisticated, adaptive systems utilize real-time control of force generated by the absorber during the impact absorption process and it is obtained through modification of the actual flow of the medium through the valve [12–14]. Typically, operation of such systems is based on two strong assumptions: i) knowledge of the complete model of the system and values of its parameters and ii) knowledge of external excitation, which is assumed to be present only at the beginning of the process. Finding optimal control strategy often involves two consecutive steps: i) determination of the optimal change of generated reaction force, which provides stopping of the impacting object with minimal level of deceleration, ii) determination of required valve opening by considering the classical control problem of regulation. The second stage can be executed either by open-loop control system with valve opening dependent on impact characteristics and time [20] or closed-loop control system with feedback to actual level of generated force [14,20,21].

Despite the fact that classical adaptive absorbers utilize online control of the valve opening in order to provide optimal deceleration of the impacting object, they do not guarantee entirely adaptive performance with proper change of reaction force in case of subsequent impacts or system disturbances. In order to provide system adaptation to such conditions, the control has to be updated according to actual state of the system or based on the online identification of the process parameters. Only then, we can consider the absorber as really adaptive impact absorbing system.

One of the methods aimed at providing optimal performance and robustness to possible system disturbances is the application of Model Predictive Control (MPC), known also as Receding Horizon Control (RHC), in which sequential optimization of the control signal is performed for finite time horizon [22,23]. The exemplary application of MPC for shock-absorbers is the air suspension system based on multi-mode switching, which provides efficient compensation of disturbances and was proposed in [24]. The MPC may lead to sub-optimal results in comparison to globally optimal solution. Nevertheless, the advantage of this approach is that it does not demand knowledge of loading conditions until the end of the process, but instead utilizes its prediction for the control horizon. Moreover, it reduces the computational burden related to determination of the control signal [25] and therefore it can be implemented in real-time control systems [26]. Let us note that MPC requires knowledge of all model parameters including impacting object mass, which can be treated as information about impact loading and should be considered as unknown in rigorous formulation of the self-adaptive impact mitigation problem. The possible solution is the application of Model Identification Adaptive Control (MIAC), in which disturbance forces are sequentially identified at each control step by using the equation of impacting object motion and conducted measurements of the actual system state. Then, the complete model of the system can be used to determine optimal system path and optimal control signal at the consecutive control step, similarly as in the case of MPC.

The introduced **Hybrid Prediction Control (HPC)** method effectively uses measurements of the actual system kinematics in order to replace the knowledge about external excitation and provide high robustness to various types of disturbances. However, it significantly differs from the methods described above since it does not require separate identification of the impacting object and numerical simulation of the system response in order to determine the optimal control strategy. In the proposed method the optimal system response is determined by using hybrid prediction technique. At each control step the required change of impacting object deceleration is predicted by using condition of global kinematic optimality for the remaining part of the process [27]. Further, the optimal change of valve area is calculated by using differential model of the system and the inverse dynamics approach. The above two-stage procedure is repeated at each control step. Since applied control strategy is entirely based on actual kinematics of the system it automatically takes into account all previously occurring phenomena including unknown impact excitation and random disturbances. Thus, the proposed system possesses the capability of automatic path correction in case of possible disturbances (e.g. occurrence of unpredicted elastic or viscous forces or possible gas leakage) and capability of system re-adaptation to subsequent impact excitations (e.g. secondary impact of additional mass). As a consequence, the proposed system can be distinguished from the other adaptive impact absorbing systems and can be classified as **self-adaptive system**.

The application of the self-adaptive shock-absorber is strongly justified in many practical engineering applications. Primarily, in majority of practical impact mitigation problems the exact mass or velocity of the impacting object is not precisely known. This concerns the situation of road/traffic accidents, e.g. the impact of car against adaptive road barrier when mass and velocity of the impacting vehicle is not known by the control system located inside the barrier or the case of frontal collision of two cars when mass and velocity of each vehicle is not known by the control system located inside the bumper of the second vehicle. Secondarily, the absorber may be subjected to the sequence of unknown impacts. For example, in the case of landing gear the primary requirement for it is to dissipate energy of the landing plane, but also to amortize the aircraft during taxing and on the runway. Finally, most of the fluid-based absorbers experience additional disturbance forces such as friction between piston and cylinder walls, viscous or elastic forces. This particularly concerns absorbers subjected to the impact loading, which sometimes is not perfectly axial or above the predicted maximal magnitude and may cause deformation or wear of particular system components such as sealing or compartment parts. Moreover, both single-chamber and double-chamber absorbers are prone to leakages, which influences their mechanical characteristics. All above mentioned arguments indicate that proposed self-adaptive system is characterized by better performance than standard adaptive systems and its application is strongly justified in engineering practice.

2. General model of double-chamber fluid-based shock-absorber

In the following sections we will analyse the class of controllable fluid-based absorbers, which are composed of two chambers filled with hydraulic or pneumatic fluid. The chambers are separated by a moving piston equipped with controllable valve. The absorber is aimed at mitigation of unilateral or bilateral impact or series of impacts acting on a rigid object of given mass (Fig. 1). Each impact excitation causes motion of the piston, change of fluid pressure in compressed and decompressed chamber and generation of the corresponding resistance force. During operation of the device the actions performed on the valve allow to control the actual rate of the fluid flow between the chambers. This leads to changes of the fluid pressure in both chambers and as a result the desired change of resistance force generated by the absorber can be obtained.

The starting point for further considerations will be universal mathematical model of the double-chamber fluid-based absorber assuming the most general thermodynamic description of the operational fluid, derived in Ref. [28]. Herein such model will be adopted for the case of impact absorption process. For the sake of simplicity we will treat the problem as one-dimensional and assume one mechanical degree of freedom, which indicates joint displacement of the impacting rigid object and the piston. Moreover, it will be assumed that the piston velocity resulting from impact excitation is relatively low (at the level of several m/s). Such assumption allows to neglect the wave propagation effects within the considered volumes of the fluid. Therefore, fluid pressure, temperature and density in each chamber will be considered as uniform during each time instant of the process. This standard assumption was also applied in previous classical papers dedicated to modelling of pneumatic vibration isolators [29,30] and shock-absorbers [31]. Consequently, thermodynamic modelling of the fluid will be performed by using Uniform Pressure Method (UPM) and it will be based on ordinary differential equations.

The proposed mathematical model of impact-subjected double-chamber absorber contains equation of motion of the impacting object (or amortized body under impact excitation) and the piston, equation of mass or volume balance for fluid filling each chamber and equation of thermodynamic energy balance for fluid filling each fluid chamber. The equation describing joint motion of the impacting object and the piston takes the form:

$$M\ddot{u} + A_1 p_1 - A_2 p_2 + F_{fr} + F_D^{BOT} - F_D^{TOP} + F_{dist}(t) = F_{ext}(t) \tag{1}$$

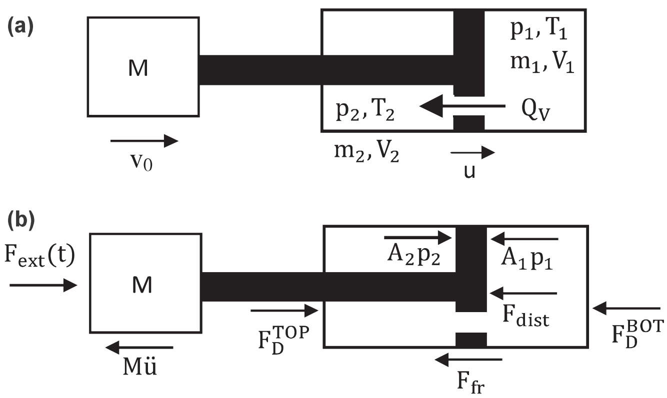

Fig. 1. Double-chamber fluid-based impact absorber under impact excitation: a) kinematic and thermodynamic quantities, b) forces acting on the impacting object and the piston.

where: M is the joint mass of the impacting object and the piston, u is actual joint displacement of impacting object and the piston, p_1 is pressure in the right (typically compressed) chamber, p_2 is pressure in left (typically decompressed) chamber, A_1 and A_2 are the cross-sectional areas of both chambers, F_{fr} is the friction force, F_D^{BOT} and F_D^{TOP} are contact forces between piston and cylinder bottoms, F_{dist} is a disturbance force caused by additional elastic or viscous effects, while F_{ext} is the external force. The equations of volume balance for fluid filling both chambers take the classical form:

$$\dot{V}_1 + \beta(p_1, T_1)V_1\dot{p}_1 - \alpha(p_1, T_1)V_1\dot{T}_1 = -Q_V \tag{2a}$$

$$\dot{V}_2 + \beta(p_2, T_2)V_2\dot{p}_2 - \alpha(p_2, T_2)V_2\dot{T}_2 = Q_V\frac{\rho_1}{\rho_2} \tag{2b}$$

where V is the total volume of the fluid enclosed in each chamber, $\beta(p, T)$ is the compressibility coefficient and $\alpha(p, T)$ is the thermal expansion coefficient, which both depend on pressure and temperature of the fluid in considered chamber. The terms at the r.h.s. of both equations indicate volumetric inflow rates into each chamber. Herein, these quantities are expressed in terms of volumetric outflow rate from the right (upstream) chamber Q_V and ratio of actual fluid densities ρ_1 to ρ_2. The balance of fluid volume can be also replaced by balance of fluid mass:

$$\dot{m}_1 = -Q_m, \quad \dot{m}_2 = Q_m \tag{3a,b}$$

where m is mass of the fluid in each chamber, while Q_m is mass outflow rate from the right chamber. Summation of both equations of mass balance yields an intuitive condition:

$$m_1 + m_2 = \text{const.} \tag{3c}$$

The equation of thermodynamic energy balance indicates that added enthalpy is transferred into change of internal energy and work done by the fluid. The equation of energy balance takes a slightly different form for the upstream than for the downstream chamber. In case of the right (upstream) chamber the enthalpy term is expressed in terms of parameters of the fluid inside the chamber:

$$\begin{aligned}&\dot{m}_1 c_p T_1 - Q_V p_1 + Q_V \alpha T_1 p_1 = \\ &\dot{m}_1\left[c_p T_1 - \alpha p_1 \rho_1^{-1} T_1 + p_1\rho_1^{-1}(\beta p_1 - \alpha T_1)\right] + m_1 c_p \dot{T}_1 - \alpha p_1 V_1 \dot{T}_1 + V_1\dot{p}_1(\beta p_1 - \alpha T_1) + p_1\dot{V}_1\end{aligned} \tag{4a}$$

where c_p is the constant pressure heat capacity. In turn, in case of the left (downstream) chamber the enthalpy term is expressed in terms of parameters of the gas submitted through the valve:

$$\begin{aligned}&\dot{m}_2 c_p T_\nu + Q_V\frac{\rho_1}{\rho_2}p_\nu - Q_V\frac{\rho_1}{\rho_2}\alpha T_\nu p_\nu = \\ &\dot{m}_2\left[c_p T_2 - \alpha p_2 \rho_2^{-1} T_2 + p_2\rho_1^{-1}(\beta p_2 - \alpha T_2)\right] + m_2 c_p \dot{T}_2 - \alpha p_2 V_2 \dot{T}_2 + V_2\dot{p}_2(\beta p_2 - \alpha T_2) + p_2\dot{V}_2\end{aligned} \tag{4b}$$

Under assumption of lossless valve flow and equality of both enthalpies the above equations can be summed up to give global thermodynamic energy balance of the absorber indicating the equality of work done over the fluid and change of its internal energy.

The system of differential equations (1)–(4) has to be complemented by four algebraic equations: the equation of state of the fluid, definition of chamber's volumes in terms of piston displacement and definition of the valve flow. The equation of fluid state defines relation between thermodynamic parameters of the fluid and it takes the form:

$$f(p, T, m, V) = 0 \text{ or } f(p, T, \rho) = 0 \tag{5}$$

The volume of both chambers can be expressed in terms of piston displacements as:

$$V_1 = V_1^0 - uA_1, \quad V_2 = V_2^0 + uA_2 \tag{6a,b}$$

The volumetric or mass flow rate of the fluid through the valve depends on pressures in upstream and downstream chambers, temperature of the fluid in upstream chamber and assumed change of valve opening during the impact process $A_\nu(t)$:

$$Q_{V\,/\,m} = f(p_1, p_2, T_1, A_\nu(t)) \tag{7}$$

In case of slightly compressible fluids, when the equations of volume balance are applied, the model governing the response of the absorber can be expressed by five differential equations involving the unknowns: u, p_1, p_2, T_1, T_2. In turn, in case of highly compressible fluids, when the equations of mass balance are applied, it is more convenient to express the model of the absorber in terms of four differential equations involving the unknowns: u, m_1, p_1, p_2 or u, m_1, T_1, T_2. The system of equations has to be complemented with the initial conditions of a proper order imposed on each of the above quantities. The model enables determination of the response of fluid-based double-chamber absorber under the impact of a rigid object of a given mass. Impact excitation can be modelled as initial velocity of the object or external impulse force.

The specification of the above model to the case of pneumatic damper with chambers filled with ideal gas is straightforward. For this purpose general form of the equation of state has to be replaced by classical ideal gas law, which enables determination of compressibility and thermal expansion coefficients in terms of temperature and pressure of gas:

$$V = \frac{mRT}{p}, \quad \beta = -\frac{1}{V}\frac{\partial V}{\partial p} = p^{-1}, \quad \alpha = \frac{1}{V}\frac{\partial V}{\partial T} = T^{-1} \tag{8a,b,c}$$

Consequently, the equations of energy balance for upstream and downstream chamber take simple form:

$$\dot{m}_1 c_p T_1 = \dot{m}_1 c_v T_1 + m_1 c_v \dot{T}_1 + p_1 \dot{V}_1 \tag{9a}$$

$$\dot{m}_2 c_p T_1 = \dot{m}_2 c_v T_2 + m_2 c_v \dot{T}_2 + p_2 \dot{V}_2 \tag{9b}$$

where c_p is a constant pressure heat capacity and c_v is a constant volume heat capacity. Let us note that the equation of the energy balance (9a) can be combined with the assumed equation of state (8a) and integrated analytically, what yields various forms of the adiabatic equation of state. Moreover, both equations of energy balance can be summed up and combined with equation of motion integrated over displacement. For modelling of the gas transfer between the absorber chambers it will be assumed that the flow of gas through the valve is described by classical isentropic flow model.

Finally, the considered model of the idealized (frictionless) double-chamber pneumatic adaptive absorber subjected to impact of the rigid object is composed of the following equations:

$$M\ddot{u} + A_1 p_1 - A_2 p_2 + F_D^{BOT} - F_D^{TOP} + F_{dist}(t) = F_{ext}(t) \tag{10}$$

$$\dot{m}_1 = -A_\nu(t)\sqrt{2}\sqrt{\frac{\kappa\left(\left(\frac{p_2}{p_1}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa+1}{\kappa}}\right)}{\kappa - 1}}\,\frac{p_1}{\sqrt{RT_1}} \tag{11}$$

$$m_1 + m_2 = m_0 \tag{12}$$

$$\frac{p_1 V_1^\kappa}{{m_1}^\kappa} = \text{const.} \tag{13}$$

$$\frac{1}{2}Mv_0^2 - \frac{1}{2}M\dot{u}^2 = m_1 c_v T_1 - m_1^0 c_v T_1^0 + m_2 c_v T_2 - m_2^0 c_v T_2^0 \tag{14}$$

$$p_1 V_1 = m_1 R T_1, \quad p_2 V_2 = m_2 R T_2, \tag{15}$$

$$V_1 = V_1^0 - uA_1, \quad V_2 = V_2^0 + uA_2 \tag{16}$$

$$\text{IC}: u(0) = 0,\ \dot{u}(0) = v_0, m_1(0) = m_1^0 \tag{17}$$

The mass M represents joint mass of the impacting object and the piston, while v_0 represents their joint initial velocity just after the impact. The above model of double-chamber pneumatic absorber remains in full agreement with the model presented in [32], which was derived from very general model of deformable inflatable structure. The passive response of the absorber for selected constant valve opening, including the influence of the initial pressure value and the influence of the value of constant valve area is presented in Fig. 2. According to graphs the pure passive operation is far from optimal response provided by smart devices controlled in semi-active way. This fact will be proved later using the numerical example prepared for comparison of proposed system with passive absorbers and systems utilizing different semi-active control methods. Obtained passive characteristics allow to analyse fundamental effects resulting from both improper valve opening and selection of initial pressure inside the chambers. Too small area of the valve causes rebounds of absorber piston and higher values of pneumatic force. In contrast, too large valve opening leads to lower values of pneumatic force but also results in undesired effect of hitting the absorber bottom. In reference to the Fig. 2b, it can be easily noticed that an increase of initial pressure inside the absorber chambers allows to change maximum value of absorber reaction force and impact absorption efficiency. The interesting effect is observed in the last phase of the passive impact absorption process. The pneumatic force is approximately equal to zero and the absorber piston slowly moves after rebound. Such situation occurs when the flow rate of gas transmitted through the valve is equal to the residual value of piston velocity.

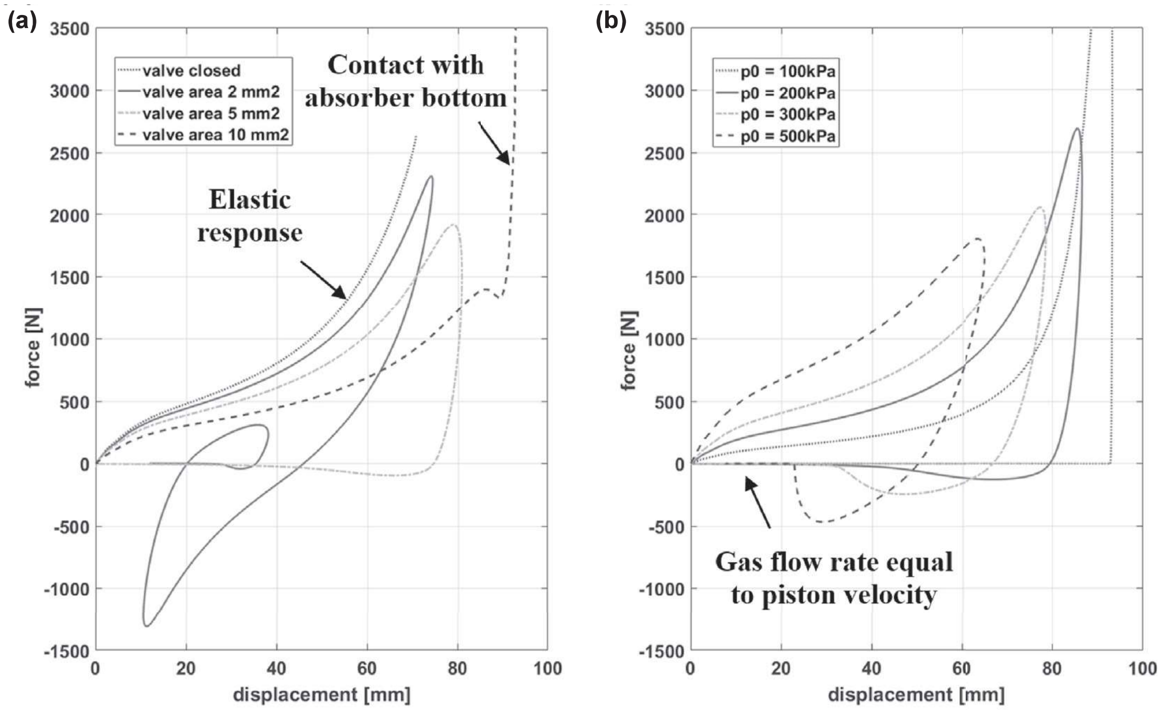

Fig. 2. Exemplary response of double-chamber pneumatic absorber under impact excitation: a) the influence of constant valve opening for initial pressure 300 kPa, b) the influence of initial pressure for constant valve opening equal 4 mm^2.

3. The problem of unknown impact mitigation

3.1. General problem formulation

For clear analysis of the self-adaptive impact absorbing system we will consider one degree-of-freedom mechanical system composed of rigid object of unknown mass M and, introduced in previous section, adaptive fluid-based absorber equipped with controllable valve. Assuming that the value of object mass and impact excitation are unknown, we had to elaborate a generic control system ensuring deceleration of the impacting object of initial velocity v_0 to zero (e.g. crash of cars) as well as amortization of protected object under sudden unpredictable force excitation.

Optimal operation of the system controlling the valve opening $A_\nu(t)$ should meet simultaneously two requirements set for adaptive impact absorbing systems:

- Dissipation of entire impact energy E_{imp} within available absorber stroke d;
- Minimization of maximum value of absorber reaction force $F_{abs}(t)$.

Optimal solution of the Adaptive Impact Absorption (AIA) problem is usually obtained by inverse dynamics approach [14] and can be calculated only if mass of amortized object (or decelerated impacting object) as well as excitation (initial velocity or external force) are known. Since unknown impact conditions are considered such solution cannot be obtained exactly. Therefore, we will consider control system which is independent from excitation conditions and provides response of the absorber comparable with the optimal solution of the AIA problem.

In Fig. 3 the system under impact excitation and formulation of unknown impact mitigation problem are presented. Aiming at possibly accurate resembling of the optimal system response we demand automatic finding and tracking of system path, which meets simultaneously the requirements of entire impact energy dissipation and minimization of reaction force. In order to obtain successful operation in case of additional disturbances and maximize system efficiency the system path should be updated to provide robust performance even if system parameters suddenly change. Update of system path should also ensure automatic re-adaption of the absorber, e.g., in case of series of impacts. In further analysis we will present the control system designed for extremely efficient mitigation of loading caused by unknown impact excitation. After appropriate design of the system and equipping it with sensors of high frequency of measurements, it is possible to automatically obtain response very close to optimal solution of the Adaptive Impact Absorption problem.

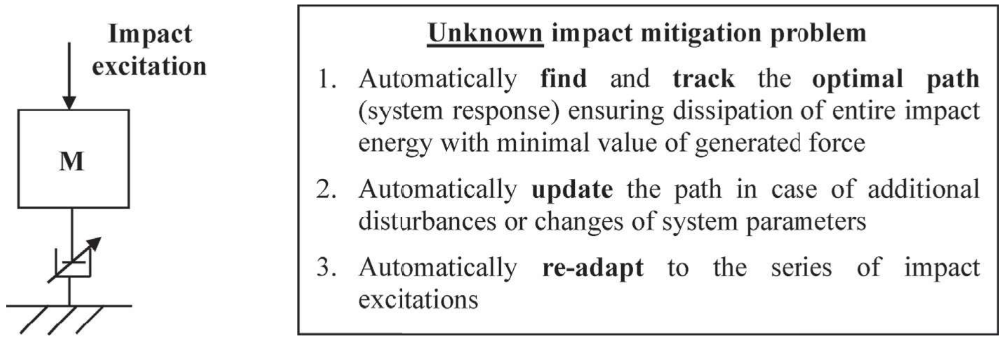

Fig. 3. Formulation of unknown impact mitigation problem.

3.2. *The corresponding control problem*

In order to ensure successful solution of the unknown impact mitigation problem stated above we have defined the path-tracking problem and applied valve control recalculated sequentially for each control step. For clearness and completeness of our discussion, we derive a solution of the path-tracking problem starting from classical formulation of the optimal impact mitigation problem which reads:

$$\text{Find } A_\nu(t) \geq 0 \text{ such that } \int_d \overrightarrow{F_{abs}} d\vec{s} = E_{imp} \text{ and } \max(F_{abs}(t) - F_{ext}(t)) \text{ is minimal} \tag{18}$$

where $F_{abs}(t)$ is the total force generated by the absorber being the sum of pneumatic force $F_{pneu}(t)$ and unknown disturbance force $F_{dist}(t)$ (internal friction, additional viscous or elastic force). The total impact energy E_{imp} results from the initial condition imposed on velocity of the impacting object, as well as the action of external force $F_{ext}(t)$, assumed as a set of high peaks of short duration for the sake of modelling of the subsequent impact excitations. Typically, the above impact mitigation control problem was considered for the simplified case, which did not include the unknown disturbance force $F_{dist}(t)$ and unknown external force $F_{ext}(t)$. In such case, general minmax problem can be reformulated into the force-based path-tracking problem:

$$\text{Find } A_\nu(t) \geq 0 \text{ such that } \int_d \overrightarrow{F_{abs}} d\vec{s} = E_{imp} \text{ and } \int_0^{t_f} \left(F_{abs}(A_\nu(t), t) - F_{abs}^{opt}(t)\right)^2 dt \text{ is minimal} \tag{19}$$

in which force $F_{abs}(A_\nu(t), t)$ is the total reaction force generated by the absorber, whereas the $F_{abs}^{opt}(t)$ is the optimal reaction force which should meet the requirements stated in Eq. (18). In the case of no constraints imposed on maximal valve opening the control problem is solved by using inverse dynamics method and feasible change of absorber reaction force is determined with the use of known parameters of the impact excitation, see [18] for details.

3.3. *The sequential path-tracking problem*

The equation of energy balance allows to express the optimal reaction force by mass of the impacting object, its actual velocity and displacement as well as actual value of the external force:

$$F_{abs}^{opt}(t) = \frac{M\dot{u}^2(t)}{2(d - u(t))} + F_{ext}(t) \tag{20}$$

Simultaneously, the absorber reaction force is the sum of pneumatic force $F_{pneu}(A_\nu(t))$ and unknown disturbance force $F_{dist}(t)$:

$$F_{abs}(A_\nu(t), t) = F_{pneu}(A_\nu(t)) + F_{dist}(t) \tag{21}$$

The velocity-displacement dependent term at r.h.s. of the Eq. (20) is the optimal deceleration force which ensures absorption and dissipation of the actual kinetic energy of the impacting object and minimization of its deceleration. The pneumatic force $F_{pneu}(A_\nu(t))$ at the r.h.s. of Eq. (21) depends on the time history of valve opening $A_\nu(t)$ and can be expressed in the following general form:

$$F_{pneu}(A_\nu(t)) = f_1\left(\int_0^t f_2(A_\nu(\bar{t}))d\bar{t}\right) \tag{22}$$

The solution of the **state-dependent force-based path-tracking problem**, which is obtained by inserting Eqs. (20) and (21) into Eq. (19), requires knowledge of the mass of the impacting object M as well as the knowledge about change of disturbance force and external force during the entire process. Therefore, the solution of the formulated state-dependent path-tracking problem cannot be directly found. In order to obtain the best possible solution of the above path-tracking problem we will implement an iterative procedure of the control recalculation. The discrete version of Eq. (20) takes the form:

$$F_{abs}^{opt}(t_i) = \frac{M\dot{u}(t_i)^2}{2(d-u(t_i))} + F_{ext}(t_i) \tag{23}$$

Therefore, the sequential force-based path-tracking is based on the repeated solving of the following control problem:

$$\text{Find } A_\nu(t) \geq 0 \text{ such that } \int_{t_i}^{t_f}\left(F_{pneu}(A_\nu(t)) - \frac{M\dot{u}(t_i)^2}{2(d-u(t_i))} - (F_{ext}(t_i) - F_{dist}(t_i))\right)^2 dt \text{ is minimal} \tag{24}$$

for the subsequent time periods (t_i, t_f). Obtaining the solution of the above control problem still requires the knowledge of the mass of the impacting object M, as well as the value of disturbance force and the value of external force at the actual time instant. Therefore, without the identification procedure of mass, actual disturbance and actual excitation, the formulated path-tracking problem cannot be solved. Since we are not able to predict loading conditions until the end of impact absorption process, it is justified to limit prediction horizon to the one control step. Moreover, according to the balance of forces present in the system, the difference between excitation force and disturbance force can be replaced by pneumatic force and inertia force at the actual time instant:

$$F_{ext}(t_i) - F_{dist}(t_i) = F_{pneu}(t_i) + M\ddot{u}(t_i) \tag{25}$$

We can use the above equation and assume that disturbance and external forces remain constant from the considered time instant until the end of the process. As a result we can reformulate the problem to the **sequential force-kinematic path-tracking**:

$$\text{Find } A_\nu(t) \geq 0 \text{ such that } \int_{t_i}^{t_{i+1}}\left(F_{pneu}(A_\nu(t)) - F_{pneu}(t_i) - M\left(\frac{\dot{u}(t_i)^2}{2(d-u(t_i))} + \ddot{u}(t_i)\right)\right) \tag{26}$$

It should be highlighted that the above problem includes specific form of the optimal absorber force, which ensures optimal deceleration of the excited object in case of unchanged impact conditions. Although the prediction time is very short and therefore the formulation is very local, it leads to precise solutions when single impact excitation is considered and no constraints on maximal opening of the valve are imposed. The above formulation does not include the actual values of the disturbance and excitation forces at the actual time instant. On the other hand, finding exact optimal control strategy still requires knowledge of the mass of the impacting object so it cannot be obtained.

Let us further focus on exact and approximate solution of the reformulated problem. We initially note that the globally optimal value of pneumatic force is determined by the formula:

$$F_{pneu}^{opt}(t_i) = F_{pneu}(t_i) + M\left(\frac{\dot{u}(t_i)^2}{2(d-u(t_i))} + \ddot{u}(t_i)\right) \tag{27}$$

Therefore the sign of the second term at the right hand side of Eq. (27) (further called kinematic condition) determines whether the actual value of pneumatic force is smaller or larger than the optimal one.

The exact solution of the single step of path-tracking problem in the considered case of unconstrained maximal value of the valve area is very intuitive. The optimal control strategy at a single control step depends on kinematic condition in the following way:

a. if $\frac{\dot{u}(t_i)^2}{2(d-u(t_i))} + \ddot{u}(t_i) > 0$ then: F_{pneu} should increase at max. rate until obtaining optimal value and further remain constant;

b. if $\frac{\dot{u}(t_i)^2}{2(d-u(t_i))} + \ddot{u}(t_i) = 0$ then F_{pneu} should remain constant $F_{pneu}(A_\nu(t)) = F_{pneu}(t_i)$;

c. if $\frac{\dot{u}(t_i)^2}{2(d-u(t_i))} + \ddot{u}(t_i) < 0$ then: F_{pneu} should decrease at max. rate until obtaining optimal value and further remain constant.

Let us note that in the above control strategy the knowledge of the impacting object mass is not required in order to determine the direction of change of the pneumatic force. It is though necessary if we want to determine the optimal value of the pneumatic force or alternatively, the required time of pneumatic force increase until reaching the optimal value. The formulation results in sequentially determined closed-loop control with feedback to actual state of the system. The important result is that a single control step requires exclusively simulation of the system response, but does not require conducting the optimization at the current control step.

4. Implementation of the self-adaptive impact absorbing system

4.1. Control principles for self-adaptive performance

In order to provide approximate, excitation-independent solution of the state-dependent path-tracking problem, we introduce additional constraints on applied operation of the valve. It will include three different control actions:

a. full closing of the valve leading to maximal feasible rate of pneumatic force increase (1st Control Mode),
b. maintaining constant level of pneumatic force during a single control step (2nd Control Mode),
c. full opening of the valve leading to maximal feasible rate of pneumatic force decrease (3rd Control Mode).

According to Eq. (27) the optimal value of piston deceleration corresponding to constant value of the reaction force can be expressed by the formula:

$$|a_{opt}(t_i)| = \frac{\dot{u}(t_i)^2}{2(d-u(t_i))} \tag{28}$$

The finite frequency of state measurements and control recalculation causes that the optimal value of deceleration cannot be obtained with ideal precision. As a result, it is reasonable to introduce the tolerance interval for the optimal system path to ensure stable operation of the absorber. In order to ensure dissipation of the entire impact energy we will assume that minimum acceptable value of piston deceleration is equal to optimal value given by Eq. (28). Moreover, maximum acceptable value of piston deceleration has to be limited in order to provide approximately optimal operation of the absorber. The choice of the upper tolerance is arbitrary and it can be correlated with surplus of actual piston deceleration or the value of its actual velocity. Here, however, we will apply other intuitive approach based on introduction of displacement tolerance Δu_{tol} which corresponds to the final displacement of the piston:

$$\left|a^{+}_{opt}(t_i)\right| = \frac{\dot{u}(t_i)^2}{2(d-u(t_i)-\Delta u_{tol})} \tag{29}$$

The above displacement tolerance can be considered as a safety factor for impact absorption process. The displacement tolerance Δu_{tol} can be fixed before the process (as e.g. 2% of the total stroke) or it can be chosen adaptively during the process (e.g. by using the surplus of deceleration in the first control step when its optimal level is exceeded). It can be proved that in the process with value of deceleration in the range $\left(|a_{opt}(t_i)|, \left|a^{+}_{opt}(t_i)\right|\right\rangle$ the value of the upper tolerance level $\left|a^{+}_{opt}\right|$ is gradually increasing.

4.2. Design of the control system — Hybrid Prediction Control

In contrast to the standard Model Predictive Control, which requires the complete numerical model of the system and the values of its parameters, we have derived the control method for which the full model of the system is not necessary. The proposed control law is based on two-stage prediction of the impact absorption process. According to this principle the kinematic condition is sufficient for determination of the system control mode (valve closed, valve fully open or the pneumatic force maintained constant). In turn, providing constant value of absorber reaction force is based on application of only thermodynamical part of the mathematical model of the system, introduction of the measured quantities, solution of the inverse dynamics problem and determination of the optimal change of valve area. Hereinafter the control technique will be called **Hybrid Prediction Control** (HPC). In order to visualize the control system architecture the block diagram is presented in Fig. 4. The self-adaptive operation of the shock-absorbing system is provided by the use of Control Mode Prediction (CMP) responsible for switching between valve closing, full opening or activation of the mode of maintaining constant level of pneumatic force, which will be provided by the second type of prediction, Inverse Dynamics Prediction (IDP).

Control Mode Prediction

The CMP is conducted with the use of integral model of the system resulting in the range of acceptable values of piston deceleration, which are defined by Eq. (28) and Eq. (29). It concerns the period lasting from actual time instant t_i to predicted final time t_f indicating dissipation of the entire impact energy. This type of prediction is performed during the entire considered process and it allows to choose the proper type of the control system operation (full closing/opening of the valve or maintaining constant level of pneumatic force). Deceleration of the piston below the value calculated from Eq. (28) results in system operation according to Mode 1, deceleration above the value calculated from Eq. (29) leads to Mode 3. In turn, the intermediate value indicates that system reached optimal constant level of reaction force and should maintain pneumatic force constant. In order to achieve such control strategy the second type of prediction is introduced.

Inverse Dynamics Prediction

The IDP is performed with the use of differential model of the system including valve area as the control variable. At the initial stage of the process when the valve remains closed and deceleration value gradually increases this type of prediction is not conducted since it would require the knowledge of impacting object mass. Once the deceleration reaches the tolerance range the differential model of the system is used to determine change of the valve area during a single control step using the inverse dynamics approach. By using the equations governing the system response (Eq. (11)–(16)) as well as the measurements of kinematic quantities and pressure values at the beginning of the control step, we are able to determine the optimal change of valve area in terms of time during a single control step. Simultaneously, the knowledge of the impacting object mass is not required since the equation of motion of the impacting object (Eq. (10)) is not directly used. Eventually, the formula defining optimal valve opening takes a general form:

$$A_v^{opt}(t) = f(u(t_i), v(t_i), a(t_i), p_1(t_i), p_2(t_i), t) \tag{30}$$

Application of the above change of valve area during a single control step allows for maintaining the constant level of pneumatic force. In the case when no additional disturbances are present the generated force can be maintained constant until the end of the process. In turn, when sudden changes of impact conditions appear, the deceleration value leaves the tolerance range and the valve has to be either fully opened or closed. In such cases the prediction is not conducted until the deceleration value reaches the tolerance range, only then the CMP activates IDP and again constant level of pneumatic force is maintained until the end of the process or next situation when the deceleration value leaves the tolerance range.

At the end of absorber stroke the piston velocity approaches zero and the value of the optimal deceleration $|a_{opt}|$ should also decrease to zero in order to avoid rebound of the piston. According to general control principles the CMP algorithm attempts to obtain the optimal value of deceleration and it fully opens the valve. This leads to equalization of pressures in both chambers and reduction of the total pneumatic force generated by the absorber. In considered case of equal areas on both sides of the piston the pneumatic force is reduced to zero, the system reaches its equilibrium position at the end of absorber's stroke and rebound of the piston is avoided.

In turn, in more realistic case when the areas on both sides of the piston are different the positive value of total pneumatic force is generated by equal values of gas pressure in both chambers. This force remains positive independently from the position of the piston inside the absorber. Therefore, the piston returns to its initial position at the beginning of absorber's stroke and equilibrium state is provided with the use of top delimiting spring. After cooling of the operational gas, pressure inside the absorber achieves its initial value and the absorber automatically becomes ready to the next impact excitation.

5. Study on robust self-adaptive performance of the system

The above control strategy will be illustrated using a simple example of system equipped with pneumatic double-chamber shock-absorber, in which the object of mass M is excited by initial velocity v_0, Figs. 1 and 3. For the sake of clarity, all parameters required for numerical simulations are provided and collected in Table 1. It should be pointed out that the value of

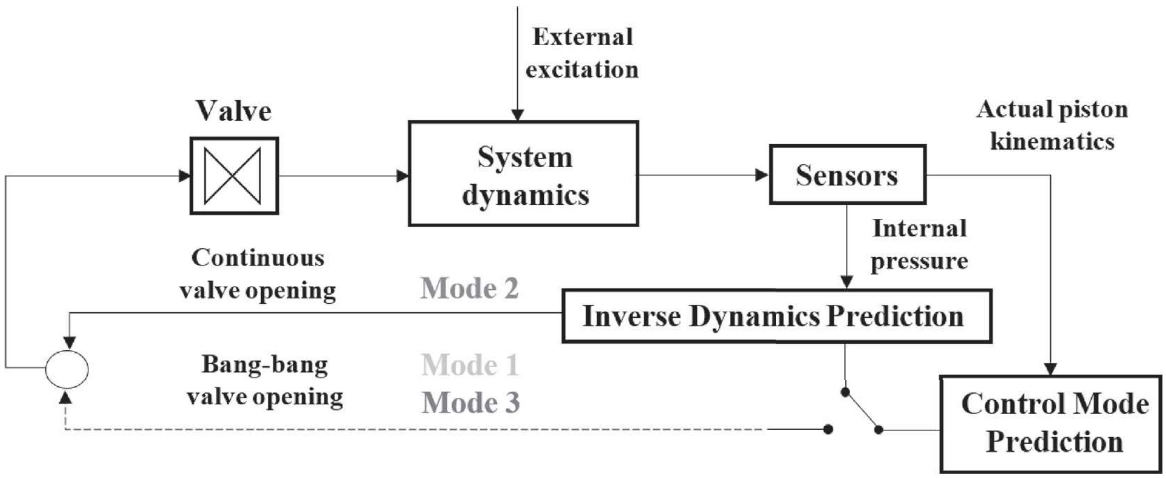

Fig. 4. Block diagram of proposed adaptive control system with Hybrid Prediction Control, the case when Inverse Dynamics Prediction is active.

impacting object's mass is unknown for control system and it is used only to simulate response of the entire system under impact excitation. The applied control is based exclusively on the knowledge of the absorber construction and measurements of piston kinematics and internal pressures in both chambers of the pneumatic shock-absorber.

The derived adaptation and control technique meets the first requirement formulated in Sec 3.1, i.e. it provides dissipation of the entire impact energy within available stroke with possibly low value of absorber's reaction force. Responses of the absorber equipped with HPC and corresponding valve opening in case of different impact velocities are shown in Fig. 5. The obtained operation of the absorber is composed of three stages: the first when valve is closed and the fastest possible increase of pneumatic force is observed, the second when reaction force is maintained at optimal constant level and the third when the valve is entirely opened to provide reduction of the pneumatic force resulting from pressure difference between absorber's chambers. In the remaining part of Sec. 5, we will prove that the second and third requirements are also fulfilled, i.e. the system automatically updates the optimal path in case of additional disturbances and it automatically re-adapts to the series of impact excitations. The assumed frequency of state measurements and control update will be equal to 3 kHz and displacement tolerance for kinematic optimality condition at the level $\Delta u_{tol} = 2$ mm.

5.1. System robustness in case of unknown additional disturbances

The presented numerical example is aimed at examination of system behavior in case of various unknown disturbances. The system is disrupted by unknown elastic or viscous forces and its responses are shown in Fig. 6. The force response is not purely pneumatic but it also includes either additional elastic force proportional to piston displacement by the stiffness coefficient $k = 10^3 \frac{N}{m}$ (Fig. 6a) or additional viscous force proportional to piston velocity by the damping coefficient $c = 20 \frac{Ns}{m}$ (Fig. 6b).

Similarly as presented above, at the beginning of impact absorption process the valve is closed and the pneumatic force increases at the fastest possible rate. After meeting the kinematic optimality condition the system starts maintaining the constant value of pneumatic force. In case of unknown elastic force acting on the absorber piston the value of deceleration treated as optimal is slightly lower than in ideal case with no additional disturbances. Analyzing the influence of system disruption by elastic force we can observe gradual increase of the reaction force. When upper tolerance of the optimality condition is exceeded the control system applies maximal opening of the valve in order to decrease the value of resulting force. After bang-bang action performed by the CMP the reaction force reaches the tolerance range and the pneumatic force is maintained at new constant level. Then, further increasing value of elastic force causes increase of the absorber reaction force. When upper tolerance for kinematic optimality is again exceeded the control system automatically repeats the process of path correction. Finally, the force response is similar as in the case without disturbances and the entire impact energy is

Table 1
Parameters of the system used in numerical simulations.

Suspended mass [kg]	Initial velocity of the mass [m/s]	Initial internal pressure in chambers [kPa]	Operational gas	Piston diameter [mm]
5	5	300	compressed air	40
Initial volume of top chamber [cm^3]	**Initial volume of bottom chamber [cm^3]**	**Piston initial position [mm]**	**Entire absorber stroke [mm]**	**Initial temperature of the gas [K]**
7.54	118.12	6	100	293.15

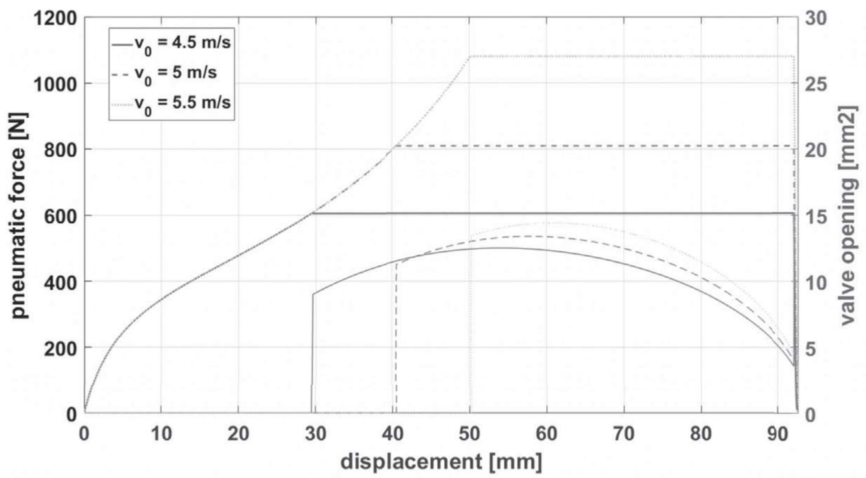

Fig. 5. Responses of the absorber equipped with HPC and corresponding valve opening in case of different impact velocities.

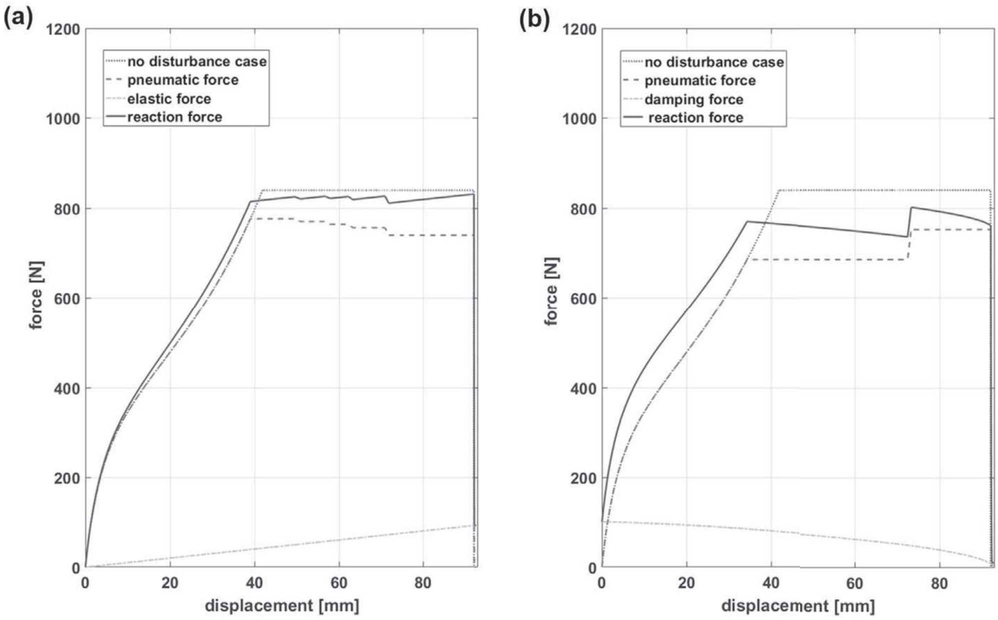

Fig. 6. Results of numerical simulation of the system response under unknown impact excitation in case of additional unknown disturbances: (a) additional elastic force, (b) additional viscous force.

absorbed by the system. However, not the entire energy is dissipated since certain part is accumulated in elastic element disrupting the absorber operation. If it operates in vertical position the elastic force can be partially balanced by the gravity. Otherwise, some rebound of the absorber piston may appear after presented impact absorption process.

Considering disruption of the system by unknown viscous force reveals the beneficial effect of this disturbance, i.e., the level of obtained reaction force is significantly lower than in no disturbance case. Such situation is connected with the fact that in analyzed numerical examples the pneumatic force has to increase from zero value to the level ensuring deceleration of the object to zero velocity within remaining part of the absorber stroke. Appearance of the damping component in total reaction force of the absorber provides an increase of the absorber efficiency because the value of reaction force required to obtain successful deceleration of the mass can be achieved earlier so consequently the value of pneumatic force can be lower. Together with increasing displacement of the piston the value of velocity-dependent damping force gradually decreases. When resultant reaction force is below the lower tolerance level determined from kinematic condition the absorber valve is closed by the CMP in order to obtain fast increase of generated reaction force. Then, when optimality condition is met the pneumatic force is maintained at the constant level until the lower tolerance level is reached again or impacting object is decelerated to zero, which terminates the system operation.

Another numerical example is aimed at demonstration of high robustness of the system in the case of simultaneous presence of different disturbance forces and discussion of the corresponding energy balance. Fig. 7a shows the force response of the system disrupted simultaneously by both elastic and viscous forces. Values of assumed stiffness and damping coefficients are $k = 1.5 \cdot 10^3 \frac{N}{m}$ and $c = 25 \frac{Ns}{m}$ respectively. The control system automatically finds the optimal system path ensuring dissipation of entire impact energy and minimization of the generated reaction force. Energy absorbed by the system within impact absorption process is composed of energy that was dissipated by pneumatic and viscous force as well as energy that was accumulated by the elastic component. The sum of energy dissipated and accumulated is equal to change of the kinetic energy of the impacting object. The complete energy balance presented in Fig. 7b proves the robust operation of the self-adaptive impact absorbing system and correctness of performed numerical analyses.

5.2. System re-adaptation to series of impacts

In order to prove re-adaptation capabilities provided by elaborated control system, the amortized object was subjected to the external force $F_{ext}(t)$ in the form of two impulses, the first of the value 20 kN and lasting first 0.001 s of the process, the second of the value 6 kN acting 0.0005 s according to following definition:

$$F_{ext}(t) = \begin{cases} 20\ \mathrm{kN} & \text{for} \quad 0 \le t \le 0.001\ \mathrm{s} \\ 6\ \mathrm{kN} & \text{for} \quad 0.0085\ \mathrm{s} \le t \le 0.009\ \mathrm{s} \\ 0\ \mathrm{kN} & \text{for} \quad \text{remaining time periods} \end{cases} \tag{31}$$

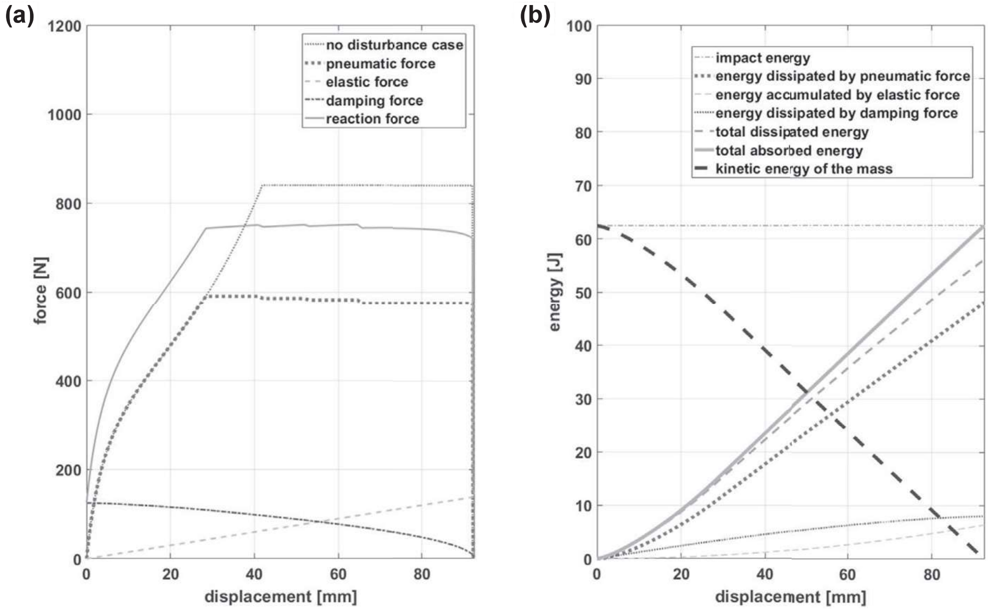

Fig. 7. Results of numerical simulation of system under unknown impact excitation in case of additional unknown disturbances: (a) force response in case of additional elastic and viscous force, (b) energy balance of absorbed energy.

In Fig. 8 the force response of the system under two impact excitations and valve opening calculated by the control system are presented. At the beginning of the process the valve is closed until the kinematic optimality condition is met. System starts maintaining constant value of pneumatic force but suddenly second force impulse acts on the mass and valve is automatically closed according to the CMP algorithm. Absorber reaction force increases until the kinematic optimality condition is met again. When the proper value of pneumatic force is reached, the IDP performs control of continuously changed valve opening until the end of absorber stroke. When assumed final displacement of the piston $(d - \Delta u_{tol})$ is reached the valve is fully opened. As mentioned earlier, this action ensures reduction of the pressure difference between chambers of the absorber in order to provide zero velocity in final piston position. Consequently the system will remain in new equilibrium position and the rebound of the absorber piston will not occur.

5.3. System response under bi-directional impulse excitations

Double-chamber shock-absorber can be used for reception of bi-directional excitations. In such case the proposed control method can be applied to provide optimal absorption of the sequence of impacts acting in opposite directions. When the piston moves in the opposite directions the control principles remain exactly the same and only changes of relative

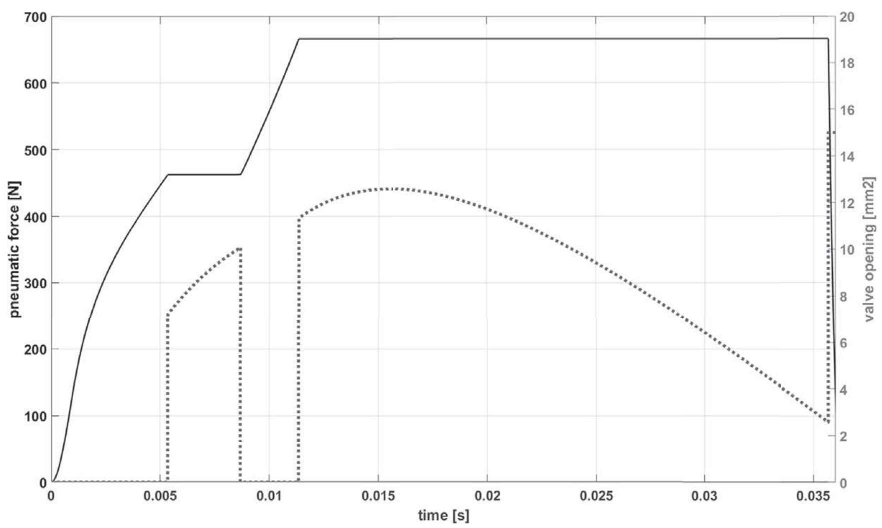

Fig. 8. Results of numerical simulation proving automatic re-adaptation of the system excited by sequence of two force impulses.

coordinates used by control system are demanded. Further, we will test the system which is excited by external force described by the following equation:

$$F_{ext}(t) = \begin{cases} 20\ \text{kN} & \text{for} \quad 0 \le t \le 0.001\ \text{s} \\ -16\ \text{kN} & \text{for} \quad 0.023\ \text{s} \le t \le 0.024\ \text{s} \\ 18\ \text{kN} & \text{for} \quad 0.050\ \text{s} \le t \le 0.051\ \text{s} \\ -16\ \text{kN} & \text{for} \quad 0.070\ \text{s} \le t \le 0.071\ \text{s} \\ 0\ \text{kN} & \text{for} \quad \text{remaining time periods} \end{cases} \tag{32}$$

Fig. 9a presents numerical simulation of the change of pneumatic force generated by the absorber in terms of time. The system permanently operates according to the control law formulated in Sec. 4. When the first force impulse acts on the mass the absorber piston starts moving and the value of pneumatic force increases until control system detects that system kinematics reaches the tolerance range. When the impacting object is subjected to unexpected force impulse of direction opposite to actual movement, the object is decelerated by both pneumatic force and external force. After complete stop of the impacting object, the external force causes its acceleration in the opposite direction. The control system detects change in direction of piston movement and continues checking kinematic optimality condition after reversal of relative coordinates. In case the kinematic optimality condition is met again the control system starts maintaining the constant value of pneumatic force in order to dissipate the impact energy within available stroke of the absorber. Further impact excitations result in the same sequence of valve control. After the last excitation the system decelerates the mass within available stroke ending its operation in the final position equal to Δu_{tol} or $(d - \Delta u_{tol})$, depending on the direction of the last excitation. In the presented example the displacement tolerance was increased to the fixed value $\Delta u_{tol} = 5$ mm in order to make final position of the piston more visible on the presented graph.

5.4. Self-adaptive system in case of on-off control

Selected constructions of fast valves for pneumatic shock-absorbers allow exclusively for bang-bang type of control. Such situation occurs for on-off valves, which can be either fully open or totally closed. In such case the simplified version of the proposed Hybrid Prediction Control has to be applied. In the adjusted version of the algorithm the Control Mode Prediction provides operation of the system in Mode 1 and 3. In turn, the Inverse Dynamics Prediction is not applicable since valve opening cannot be changed in a continuous manner. As a result the operation of the valve during time period (t_i, t_{i+1}) is performed according to the simple control law:

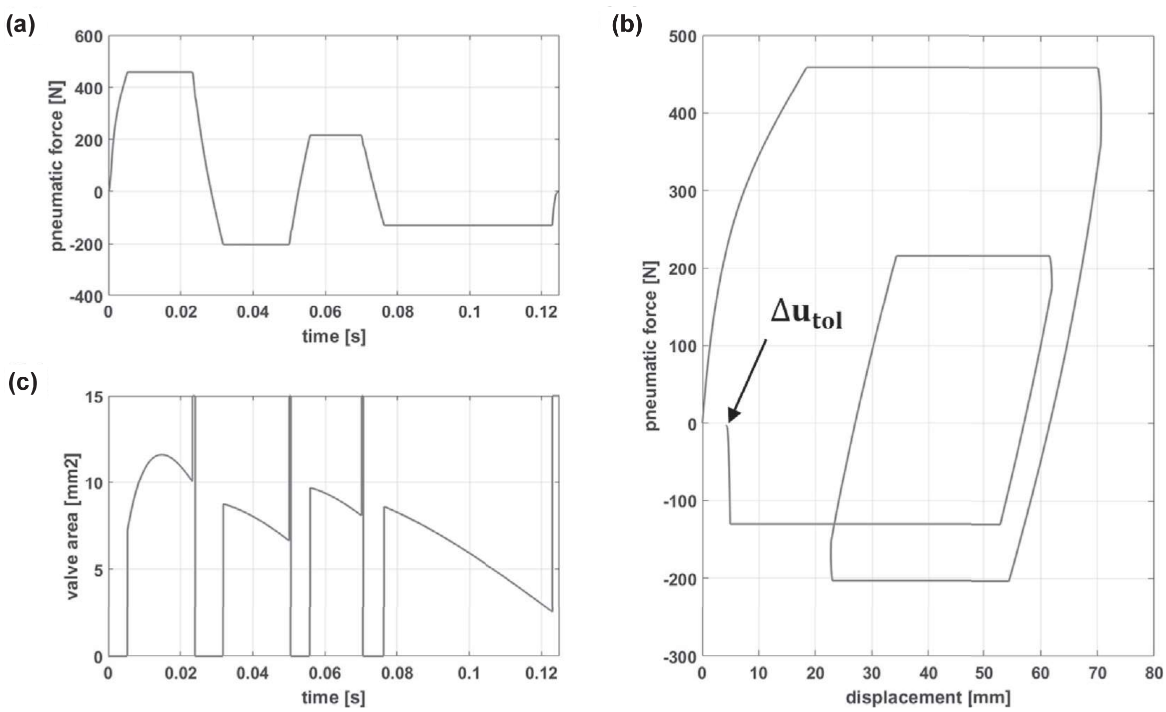

Fig. 9. Results of numerical simulation of the system under bi-directional impulse excitations (a) pneumatic force in time, (b) absorber response - force vs. displacement, (c) control in time.

$$A_v^i(t) = \begin{cases} 0 & \text{if} \quad |a(t_i)| < |a_{opt}| \\ A_v^{max} & \text{if} \quad |a(t_i)| > |a_{opt}^+| \\ A_v^{i-1}(t) & \text{if} \quad |a_{opt}| \le |a(t_i)| \le |a_{opt}^+| \end{cases} \tag{33}$$

Fig. 10 presents numerically obtained response of the system and applied control in terms of time, as well as response of the system in terms of displacement. Operation of the on-off system under bi-directional impact excitation is very similar to the operation of the system capable of continuous valve control based on IDP. However, inability of maintaining constant value of pneumatic force leads to oscillations of this force, which are especially noticeable at the end of absorber stroke. Such effects reveal the disadvantages of using on-off control in pneumatic systems and allow to formulate the conclusion that in case of on-off valve control the assumed absorber stroke should be shorter than in case of continuous valve control. Moreover, the time interval of control steps should be decreased. Nevertheless, the absorber controlled in on-off manner ensures successful operation of the self-adaptive system and provides performance comparable with previously discussed system.

6. Comparison of HPC with standard control methods

In order to prove the attractiveness of proposed control method, comparisons of HPC with standard adaptive and optimal control systems are presented in this section. The first comparison concerns the system operating under unknown disturbances, which were assumed the same as in Sec. 5.1. Fig. 11 presents response of the system disrupted by elastic or damping force in three cases:

- optimally selected value of constant valve opening $A_v = 7\ \mathrm{mm}^2$ — passive response;
- absorber equipped with Hybrid Prediction Control system;
- absorber with Model Identification Adaptive Controller (MIAC).

For quantitative comparison of absorbers several quality indices can be used, e.g., mean reaction force, impact absorption efficiency or stroke efficiency [33]. These measures of the absorber effectiveness are often recalculated to obtain specific values related to the absorber mass. In presented study all control methods are applied on the same device so the absorber mass does not influence values of these indices. In Table 2 values of maximum reaction force F_{max}, impact absorption efficiency η_{ae} and stroke efficiency η_{se} are collected. Following definitions of quality indices are assumed:

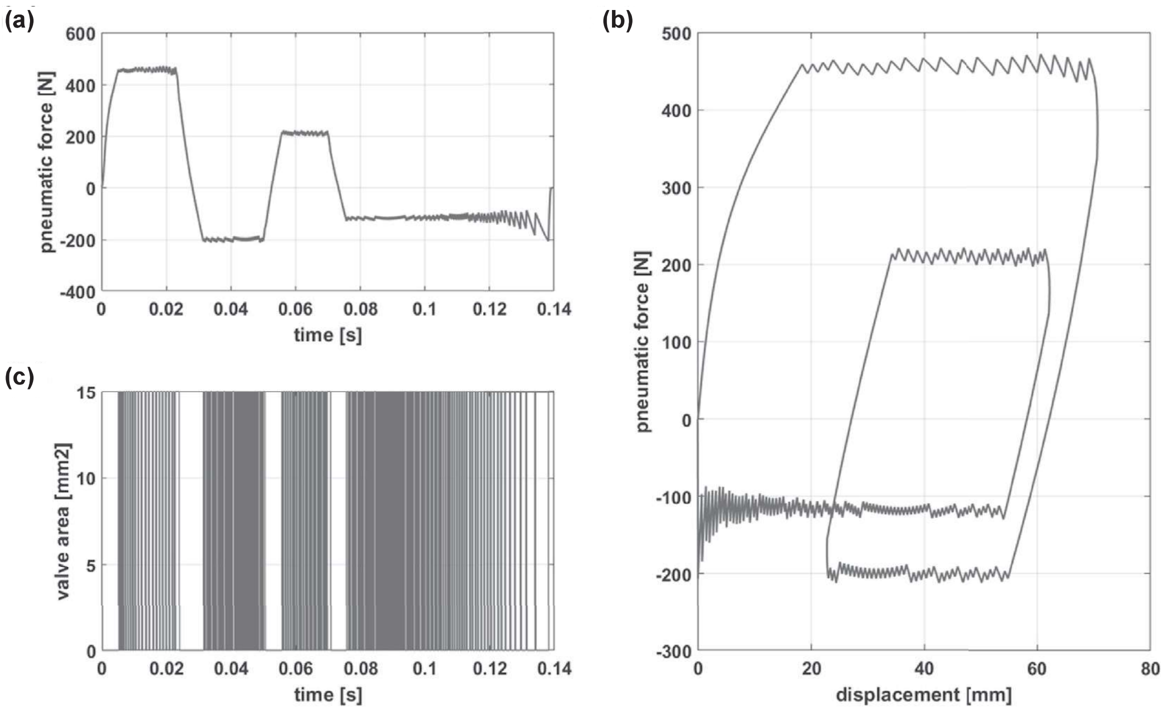

Fig. 10. Results of numerical simulation of system equipped with two-state valve under bi-directional impulse excitations (a) pneumatic force in time, (b) absorber response - force vs. displacement, (c) control in time.

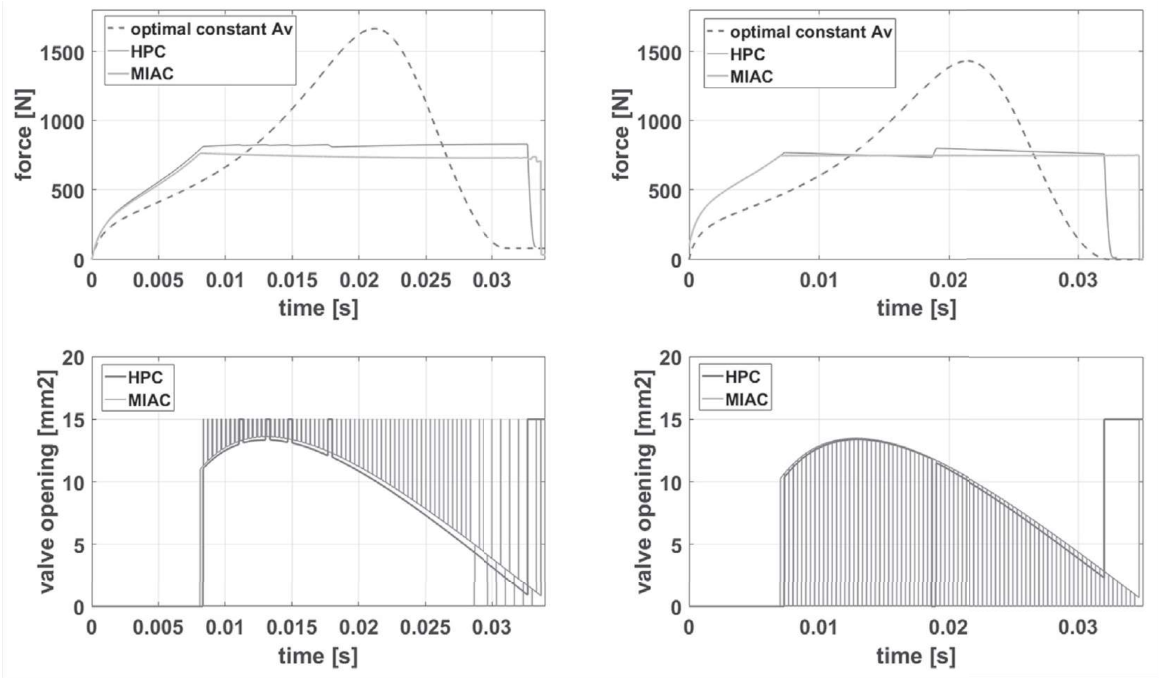

Fig. 11. Results of numerical simulations for comparison of the HPC performance with performance of passive absorber with constant valve opening and absorber equipped with MIAC system, left side: absorber disturbed by elastic force, right side: absorber disturbed by viscous force.

$$F_{max} = \max_{t \in [0, t_f]} F_{abs}(t) \tag{34}$$

$$\eta_{ae} = \frac{\int_0^{u_{max}} F_{abs}(\overline{u}) d\overline{u}}{F_{max} u_{max}} \tag{35}$$

$$\eta_{se} = \frac{u_{max}}{d} \tag{36}$$

where t_f is the final time of impact absorption process.

The response of passive absorber with optimally selected constant value of valve opening is moderately effective. Nevertheless, its performance is consistent with performance of standard passive absorbers described in the literature, e.g. [34]. In contrast, the reaction force of controlled absorber provides even twice lower value of maximum reaction force. Comparison of the HPC system with MIAC shows that the high performance can be obtained without identification of the impact loading. MIAC provides a bit more efficient response of the system but simultaneously requires conducting online numerical simulation of the absorber response in order to update optimal control signal. In turn, HPC system is slightly less effective but the control signal for each control step can be determined from relatively simple analytical formula. The difference between efficiency of MIAC and HPC results from assumed value of displacement tolerance Δu_{tol}. In presented MIAC system such tolerance does not exist and controller aims to achieve optimal level of piston deceleration and to utilize entire absorber stroke.

Table 2
Comparison of Hybrid Prediction Control with optimally designed passive system and absorber equipped with Model Identification Adaptive Controller.

		Optimally selected constant value of A_v	Hybrid Prediction Control	Model Identification Adaptive Controller
Elastic disturbance	Maximum value of reaction force [N]	1666	831.6	800.1
	Impact absorption efficiency [%]	45.4	81.2	83.1
	Stroke efficiency [%]	87.9	98.5	100
Damping disturbance	Maximum value of reaction force [N]	1434	801.8	751.6
	Impact absorption efficiency [%]	51.5	82.9	88.5
	Stroke efficiency [%]	90	100	100

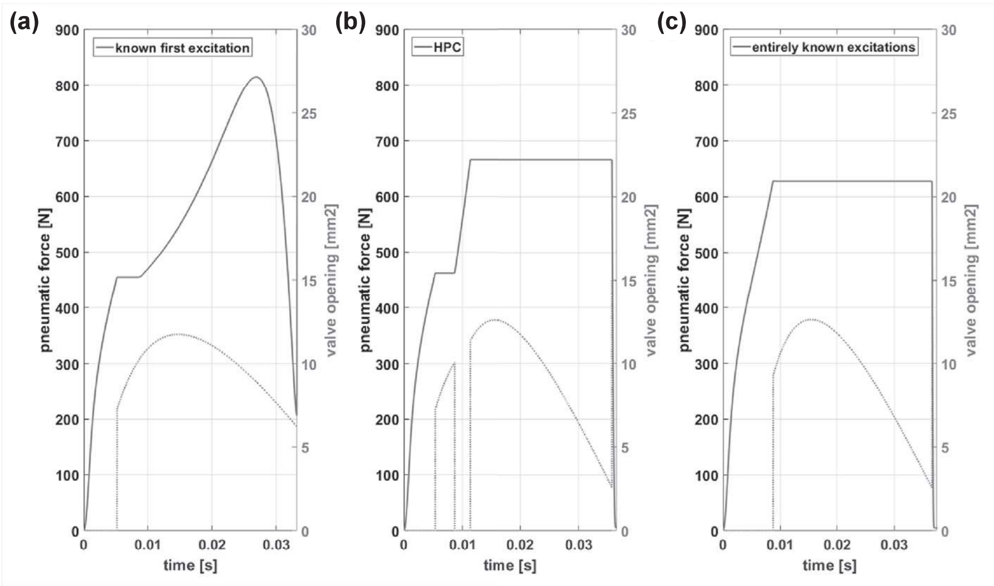

Fig. 12. Response of the absorber excited with two subsequent force impulses: (a) optimal control calculated without knowledge about second excitation, (b) solution provided by HPC system, (c) globally optimal response with full information about subsequent excitations.

Nevertheless, analysis of the control signal allows to notice that presented very efficient operation of the MIAC system is rather hypothetical because required corresponding speed of control signal changes is extremely high. Consideration of practical limitations concerning speed of the absorber valve opening and closing leads to the MIAC system of comparable performance as proposed HPC system. It should be also highlighted that cost of control calculations is much lower in case of HPC so it is more feasible for practical implementation in the absorber controlled in real time.

The second comparison of the HPC with other control methods is aimed at evaluation of the system performance in case of subsequent impact excitations. The system excitation is assumed the same as in Sec. 5.2. and the following control strategies are confronted:

- Optimal solution calculated for identified single impact excitation, without knowledge about second force impulse
- HPC control strategy — automatic re-adaptation just after second force impulse
- Globally optimal solution calculated in case of full knowledge about subsequent impacts

Numerically obtained responses of systems listed above in case of two subsequent impacts are shown in Fig. 12. The initial response of the conventional control system, which is not able to readapt to the second impact excitation, corresponds to the impact absorption efficiency $\eta_{ae} = 68\%$. Despite this fact, system operation cannot be considered as successful because the absorber does not dissipate the entire impact energy and piston hits the absorber bottom with residual velocity. Moreover, during the second stage of the process the maximum value of absorber reaction force is much higher than in case of the absorber with HPC or globally optimal response. Comparison of the HPC with the globally optimal solution shows that maximum reaction force and impact absorption efficiency can be at the similar level. In presented illustrative example differences are at the level of 6% - maximum reaction force $F_{max} = 666$ N and $\eta_{ae} = 83\%$ vs $F_{max} = 629$ N and $\eta_{ae} = 88\%$. Moreover, the globally optimal solution should be treated as hypothetical case because the exact knowledge about all subsequent excitations is required and thus realization of such control strategy is not feasible in practice. In case the next excitation can be identified during the process the Model Predictive Control (MPC) can be implemented. The performance of such system will be within the range appointed by the HPC and globally optimal response.

7. Summary and conclusions

The paper presents the subsequent steps of development and analysis of the novel control system based on Hybrid Prediction Control, which ensures self-adaptive performance of pneumatic absorber. The considerations start with formulation of general model of double-chamber fluid-based absorber equipped with controllable valve. Then, the problem of unknown impact mitigation is formulated as the problem of finding and tracking the optimal state-dependent path of the system, its automatic update in case of unpredicted disturbances and re-adaptation to series of impacts. The detailed analysis of exact solution of the posed control problem leads to the development of simplified hybrid control system based on the Control Mode Prediction, which is aimed at selecting the type of valve control (bang-bang or continuous), and Inverse Dynamics Prediction responsible for computing the continuous change of valve opening. The operation of the proposed control system is analyzed numerically using different types of disturbances, series of impacts and bi-directional impact excitations.

The proposed innovative self-adaptive system based on HPC provides efficient mitigation of unknown impact excitations in all diverse scenarios covered by presented numerical examples. The system ensures not only optimal response in case of unknown single impact excitations but it also provides successful mitigation of uni- and bi-directional impact series. Moreover, it is characterized by unprecedented in AIA systems robust operation under various disturbance conditions caused e.g. by additional elastic and viscous forces. Although the outstanding characteristics of the proposed control technique was shown on the illustrative example of double-chamber pneumatic shock-absorber, it is expected to be applicable in other types of fluid-based absorbers. Furthermore, the proposed general approach can be widely implemented in various practical applications thanks to independence from values of impacting object's mass, external excitations and disturbances.

Further research plans of the authors include experimental demonstration of the system operation and evaluation of real system robustness under disturbances as well as changes of system parameters. Another research plan is devoted to adjustment of the elaborated approach to the application of vibration mitigation system and development of dedicated control algorithms.

Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

Acknowledgment

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6. Author contributions

Adaptable pneumatic shock absorber,

published in the *Journal of Vibration and Control*, Vol. **25**, No. 3, pp.711-721, (2019).

Rami Faraj (corresponding author):

- developed the concept of adaptable pneumatic shock-absorber with overlapping slots and vents, and the prepared patent pending,
- converted the control problem into the optimal design problem,
- proposed the concept of system adaptation by the use of single reconfiguration technique,
- performed numerical simulations of the system operation,
- designed and manufactured system demonstrator,
- conducted experimental validation of the numerical model and verification of the concept of system operation,
- analysed and interpreted the results,
- wrote the manuscript and prepared figures.

Cezary Graczykowski:

- contributed to concept elaboration and preparation of patent pending,
- derived the model describing system dynamics,
- elaborated the method of calculating the optimal change of the valve area,
- verified numerical results by independently conducted numerical simulations,
- planned case studies for evaluation of the influence of manufacturing or model inaccuracies and designed the comparison with semi-active control strategy,
- contributed to interpretation of the results,
- contributed to the manuscript preparation.

Jan Holnicki-Szulc:

- conceived the concept of obtaining variable gas release by overlapping narrow slots and optimally shaped vents,
- contributed to preparation of the patent pending,
- helped to interpret the results,
- supervised the study.

Rami Faraj	Cezary Graczykowski	
Rami Faraj	Cezary Graczykowski	Jan Holnicki-Szulc

Can the inerter be a successful shock-absorber? The case of a ball-screw inerter with a variable thread lead,

published in the *Journal of the Franklin Institute*,

DOI: 10.1016/j.jfranklin.2019.04.012, 1-18 (2019).

Rami Faraj (corresponding author):

- elaborated the concept of ball-screw inerter with variable thread lead and prepared patent pending,
- derived the model of the system,
- contributed to numerical simulations,
- analysed and interpreted the results,
- conducted literature review and prepared the manuscript.

Łukasz Jankowski:

- contributed to numerical simulations,
- formulated and solved the optimization problem,
- helped to interpret the results,
- contributed to preparation of the manuscript.

Cezary Graczykowski:

- contributed to derivation and verification of the model, derived the energy balance,
- contributed to numerical simulations,
- helped to interpret the results,
- contributed to preparation of the manuscript.

Jan Holnicki-Szulc:

- contributed to elaboration of the concept,
- supervised the study.

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Jan Holnicki-Szulc

Adaptive inertial shock-absorber,

published in the *Smart Materials and Structures*, Vol. **25**, 035031-1-9 (2016).

Rami Faraj (corresponding author):

- elaborated the concept of adaptive inertial shock-absorber and prepared patent pending,
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- analysed and interpreted the results,
- wrote the manuscript and prepared figures.

Jan Holnicki-Szulc:

- conceived and supervised the study,
- helped to interpret the results.

Lech Knap:

- contributed to numerical simulations performed in MSC Adams,
- analysed the construction of proposed system,
- contributed to preparation of the manuscript.

Jarosław Seńko:

- developed the model and performed simulations in MSC Adams,
- tuned the model parameters for the case study,
- contributed to development of the system operation.

Rami Faraj .... Rami Faraj	.... Jan Holnicki-Szulc
.... Lech Knap	Seńko .... Jarosław Seńko

Development of control systems for fluid-based adaptive impact absorbers,

published in the *Mechanical Systems and Signal Processing*, Vol. 122, pp. 622-641, (2019).

Cezary Graczykowski (corresponding author):

- analysed the Adaptive Impact Absorption (AIA) methods developed so far and identified their insufficiencies,
- derived general model describing dynamics of single chamber fluid-based absorber,
- contributed to reformulation of the AIA problem,
- independently implemented numerical model of the system for verification of selected results,
- contributed to examination of the influence of imprecise impact identification and unpredicted force disturbances or sudden gas leakages on the response of identification-based AIA systems,
- contributed to elaboration of the novel kinematics-based control method,
- contributed to elaboration of the control algorithms based on Automatic Path Finding, Hybrid Path Tracking and Automatic Path Update,
- contributed to analyses of the proposed control method performance,
- contributed to the writing of the manuscript.

Rami Faraj:

- analysed the Adaptive Impact Absorption (AIA) methods developed so far and identified their insufficiencies,
- implemented numerical model of the pneumatic shock-absorber,
- contributed to reformulation of the AIA problem,
- implemented standard control systems based on feed forward control, two types of feedback control methods based on the pressure or acceleration measurements,
- contributed to examination of the influence of imprecise impact identification
- and unpredicted force disturbances or sudden gas leakages on the response of identification-based AIA systems,
- contributed to elaboration of the novel kinematics-based control method,
- contributed to elaboration of the control algorithms based on Automatic Path Finding, Hybrid Path Tracking and Automatic Path Update,
- contributed to analyses of the proposed control method performance,
- contributed to the writing of the manuscript and prepared figures.

Cezary Graczykowski .... Rami ... Faraj

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Hybrid Prediction Control for self-adaptive fluid-based shock-absorbers,

published in the *Journal of Sound and Vibration*, Vol. 449, pp. 427-446, (2019).

Rami Faraj (corresponding author):

- implemented model of the double-chamber pneumatic shock-absorber,
- contributed to formulation of the unknown impact mitigation problem and replacement of the Adaptive Impact Absorption (AIA) problem by the state-dependent path tracking,
- contributed to derivation and implemented the control law of the Hybrid Prediction Control (HPC) method based on the Control Mode Prediction and Inverse Dynamics Prediction,
- performed numerical simulations of the system under additional disturbances, series of impact excitations and bi-directional excitation,
- analysed system operation in case of control based on the 'on-off' valve,
- implemented adaptive and optimal control methods for comparison with the proposed HPC system,
- analysed and interpreted the results,
- contributed to writing of the manuscript and prepared figures.

Cezary Graczykowski:

- derived general model of double-chamber fluid-based absorber,
- contributed to formulation of the unknown impact mitigation problem and replacement of the Adaptive Impact Absorption (AIA) problem by the state-dependent path tracking,
- contributed to derivation of the HPC control law based on the Control Mode Prediction and Inverse Dynamics Prediction,
- identified two types of prediction performed within system operation,
- implemented numerical model of the system in order to verify selected results,
- helped to interpret the results,
- contributed to preparation of the manuscript.

Rami Faraj ....

Rami Faraj

Cezary Graczykowski ....

Cezary Graczykowski

Appendix 1

HIGH PERFORMANCE PNEUMATIC SHOCK-ABSORBERS FOR AERONAUTICAL APPLICATIONS

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Keywords: *Airdrop, Adaptive Impact Absorption, Landing Gear, Pneumatic Absorber, Semi-passive Control*

Abstract

The paper is aimed at development of high performance shock-absorbers for aeronautical applications. This contribution concerns pneumatic dampers because of their lightweight, technical simplicity and low manufacturing costs. The concept of semi-passive devices is introduced and single reconfiguration technique is discussed for both single- and double-chamber shock-absorber. Presented general approach to optimal design of the semi-passive devices can be applied for design of different types of fluid-based absorbers, e.g. hydraulic or oleo-pneumatic dampers. The absorbers can be used as a suspension of light airdrop system as well as a part of landing gear of small UAV.

1 Introduction

1.1 Motivation

Shock absorption phenomenon is present in many aeronautical systems. Problems of impact mitigation are discussed in papers concerning development of high performance landing gears [1-3], design of airdrop systems [4, 5] as well as techniques of structure self-protection [6, 7] that can be applied in space systems. Exemplary systems which are subjected to impact excitations are shown in Fig. 1.

Despite unquestionable progress in the field of smart sensors and actuators, which provide much better performance than systems used so far, a major part of absorbers used in practice are passive devices. This fact is caused by strict requirements for high system reliability and demand of fail-safe design. Nevertheless, efficiency of passive absorbers is limited and adaptation to impacts is impossible. The optimal response is obtained for particular operational conditions whereas, e.g., the airplane suspension should operate efficiently in typical landing conditions and simultaneously it has to meet requirements for maximum touchdown velocity specified in aviation regulations [8]. Excitations during these conditions are completely different [9]. This fact is the motivation to develop alternative solutions with adaptive capabilities.

(a)
(b)


Fig. 1. Systems subjected to impacts (a) airdrop system [10], (b) unmanned airplane during touchdown – photo taken during SAE Aero Design West 2016.

In this contribution the author discusses concepts of single reconfiguration technique aimed at adjustment of the system to different impact conditions and providing high performance which will be comparable with semi-active absorbers controlled in real-time [11]. The paper includes analyses of two pneumatic dampers: single-chamber absorber with gas release to the environment [12] and double-chamber absorber equipped with metering pin. Both devices ensure optimal impact response and adaptation to different loading conditions by means of single shape adjustment performed on selected system components.

1.2 Problem formulation

For the sake of clarity the design and analyses of the shock-absorbers are shown on example of 1 DOF system. Nevertheless, the results and conclusions from conducted research can be used for solving more complex impact absorption problems, which concerns systems with several DOF such as entire landing gear of the aircraft.

The object of mass M is equipped with pneumatic absorber and it is subjected to the impact defined by initial relative velocity v_0. The operational gas is compressed during movement of the piston and in the case of single-chamber shock-absorber it is released through the valve to the environment (Fig. 2a). In contrast, use of double-chamber device (Fig. 2b) allows to transfer the gas from compressed chamber (no. 1) to decompressed chamber (no. 2).

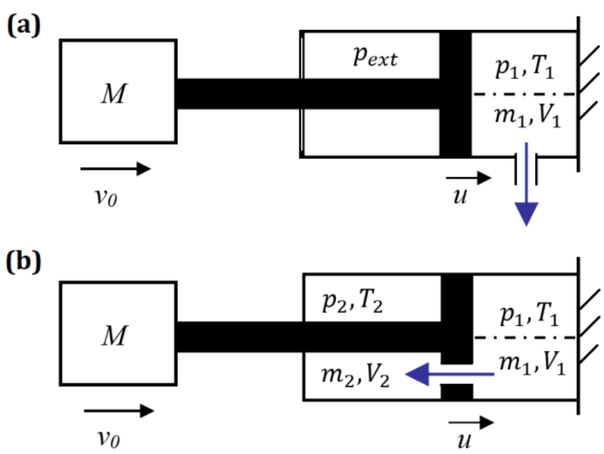

Fig. 2. Schemes of the system under impact excitation, object equipped with: (a) single-chamber absorber, (b) double-chamber absorber.

The time history of valve opening area $A_v(t)$ corresponds to the force response of the absorber being the function of internal pressure p_1 and external pressure p_{ext} (or internal pressures p_1 and p_2 in case of double-chamber device). Internal pressures depends on the mass of gas m, volume V and temperature T. Change of chambers volumes V_1 and V_2 is geometrically related to the piston displacement u and thermodynamically related to gas state variables mentioned above. Detailed description of the applied model of pneumatic single-chamber as well as double-chamber shock-absorber can be found in [13].

Entirely closed valve results in high increase of absorber reaction force due to pneumatic spring effect, whereas finite value of valve area A_v lead to slower gas compression because a particular amount of gas is released and as a result the system stiffness is decreased. For the actual state of the system, there exist a value of valve opening area for which pneumatic force starts decreasing. It means that the gas of pressure p_1 is no longer compressed although the volume V_1 decreases. It is also possible to find the valve opening which ensures constant value of absorber reaction force but to achieve this the valve area has to be variable in time [14].

One of widely used goal functions, that has to be minimized during optimization of the absorber operation, is the maximum value of the reaction force (1).

$$\max F_{react}(t) = \min \tag{1}$$

Simultaneously, the requirement (2) of entire impact energy dissipation within available absorber stroke d has to be met.

$$\int_0^d F_{react}(\bar{u})d\bar{u} = \frac{1}{2}Mv_0^2 \tag{2}$$

When we are able to appropriately control the gas release to provide constant value of absorber's reaction force, the optimal feasible solution of the formulated impact absorption problem will be two-phase operation of the shock-absorber:

- fastest possible increase of the reaction force – valve closed,
- maintaining constant reaction force of the value which ensures dissipation of entire impact energy within available stroke.

2 The concept of semi-passive pneumatic shock-absorbers

2.1 Proposed adaptation strategy

In order to ensure system operation consistent with optimal solution of the formulated min-max problem and provide as simple as possible adaptation mechanism, the concept of adaptable pneumatic shock-absorbers was elaborated.

The proposed adaptation strategy is composed of several short actions performed short time period before the impact:

- identification/prediction of excitation conditions,
- calculation of optimal impact mitigation scenario,
- reconfiguration of the system components.

After system reconfiguration the optimal response of the shock-absorber should be obtained in passive manner.

2.2 Single-chamber shock-absorber

2.2.1 Device construction and passive operation

In Fig. 3. the proposed semi-passive single-chamber pneumatic shock-absorber is shown. The optimal force response is obtained using two concentric cylinders, first with vents of proper shape and second cylinder with narrow slots. When absorber is subjected to impact excitation the relative movement of cylinders occurs. At the beginning of the process the gas is compressed because there is no overlapping area of slots and vents. After reaching the position u_x, which corresponds to the optimal value of absorber reaction force, slots and vents start overlapping and constant force value is maintained until the end of the stroke. The construction of proposed absorber demands introduction of the third phase of system operation, i.e., final exhaust of the gas. The reason for that is the fact that at the end of the impact absorption process some amount of gas remains in cylinders and internal overpressure has to be reduced to avoid rebound.

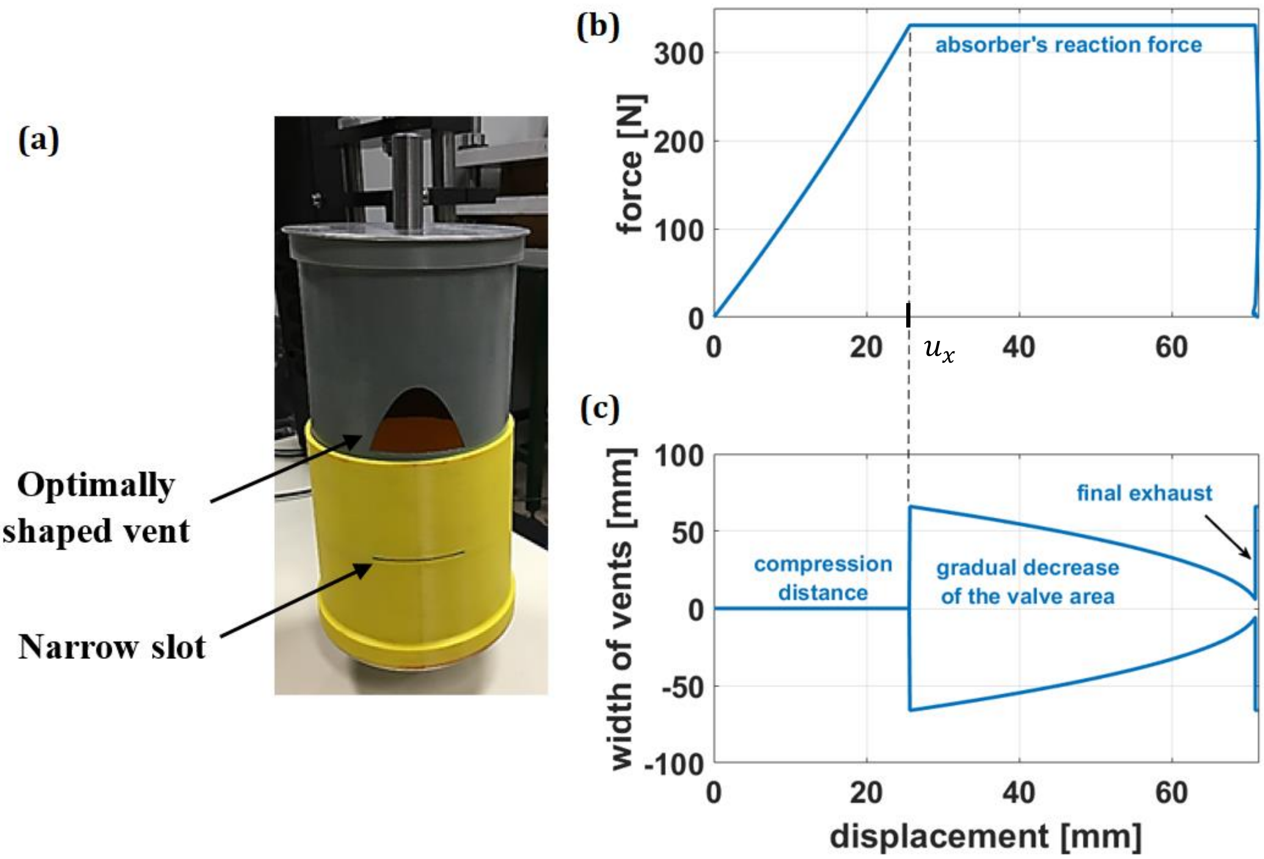

Fig. 3. Proposed single chamber pneumatic shock-absorber: (a) prototype device prepared using 3D printing technology, (b) optimal reaction force of the absorber in case of no initial overpressure, (c) shape of the vent ensuring optimal response of the absorber during overlapping of slots and vents.

The relative displacement u_x, further called 'compression distance', is determined using energy balance obtained by integration of the object's equation of motion with assumption of adiabatic gas compression during first phase of absorber operation. Inverse dynamics method applied for determination of optimal valve area $A_v(u)$ was presented and discussed in details in previous work [12]. Assuming that absorber has a particular number of slot-vent pairs n and slots are rectangles of height h, the optimal width of vents w as a function of relative displacement u can be calculated from the formula:

$$\frac{A_v(u)}{n} = \int_u^{u+h} w(\bar{u})d\bar{u} \quad (3)$$

The number of conducted simulations as well as experimental tests have shown that the simplified formula for vent shape can be applied:

$$\frac{A_v(u)}{n} = w(u)h \quad (4)$$

In order to ensure the reader that such simplification is reasonable, the influence of slot height h = 5 mm on the effective valve area and force response of the absorber is shown in Fig. 4.

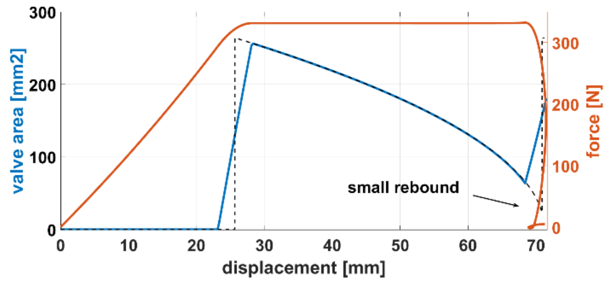

Fig. 4. Influence of not infinitesimal height of the absorber slots - system response in case of h = 5 mm.

The numerical simulation was conducted for system with parameters collected in Tab. 1.

Table 1. Parameters of simulated and manufactured pneumatic shock-absorber.

M **[kg]**	***v₀*** **[m/s]**	***p₀*** **[kPa]**	***T₀*** **[K]**
5	2.5	101.3	293.15
L **[mm]**	**∅*D*** **[mm]**	***d*** **[mm]**	**Slot-vent no.**
150	150	72.5	2

The height of the slot which is not infinitesimal or in other words the finite width of the vent near displacement equal to u_x leads to smoothening of the force response of the absorber. As a result small rebound is observed at the end of absorber stroke. Nevertheless, the final performance of the device is very close to theoretical optimal solution. The prototype manufactured for experimental validation of the concept has two slots of height h = 2 mm [12] so the feasible solution can be even closer to the optimal one.

2.2.2 Adaptation mechanisms

In order to ensure optimal response of the system in various excitation conditions, the shape of absorber's vents should be determined for all possible impact conditions. The influences of ±10% change of mass M or initial velocity v_0 are shown in Fig. 5a and Fig. 5b respectively.

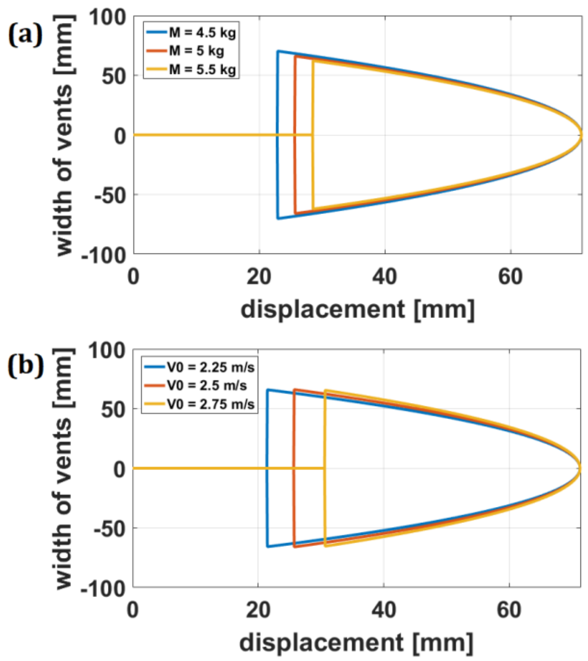

Fig. 5. (a) optimal shapes of the vent for different masses of the amortized object M, (b) optimal shapes of the vent for different initial velocities v_0.

It can be noticed that the most important parameter necessary for successful adaptation of the absorber is the compression distance u_x Indeed, the character of vents' shape is slightly different for various impact conditions but the influence of it is much smaller. In further discussion the advantage will be taken from this inference.

Now, let me introduce schemes of mechanisms which can be used for adjustment of the compression distance and vent shape. On the beginning, we have to choose the vent shape which will be cut in the one of absorber's cylinders. We can choose the widest vent and then lengthen it. As a result the adaptation will be realized by appropriate decrease of the vent. Alternatively, the global optimization problem can be formulated in order to find a compromise solution for all possible impact conditions. When the shape of vents is selected and cut precisely in the cylinder, we can move our attention to the adaptation mechanism. The compression distance u_x can be easily shortened or lengthened using moveable shutter as shown in Fig. 6a. For more optimal response of the absorber the additional shutters can be used to increase or decrease the width of vents (Fig. 6b). Side shutters can be mounted at the appropriately selected angle to ensure possibly best resembling of the optimal vent shape, which is calculated for predicted values of M and v_0. If the designed mechanism does not allow to obtain a exactly desired shape of vents, e.g. shutters are perpendicular as shown in Fig. 6b, the next optimization problem can be formulated to find a new, actually best value of u_x and to determine required opening of side shutters. According to the practice implementation of proposed approach the optimization processes should be done offline and lookup tables should be used.

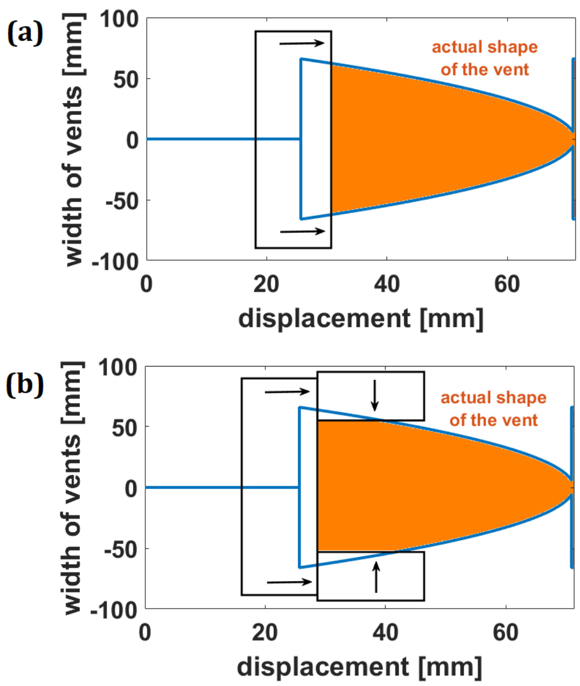

Fig. 6. (a) mechanism for compression distance adaptation, (b) mechanism for compression distance and vent shape adjustments.

2.2.3 Simplified adaptation strategy

In this section a brief discussion on the simplified adaptation technique is shown. According to the conclusion formulated during analyses of the change of optimal valve opening in case of different values of object mass and various initial velocities (Fig. 5), the proposed adaptation strategy will be based exclusively on the adjustment of compression distance u_x. In turn, the shape of absorber's vent is calculated for nominal impact conditions, as shown previously in Tab.1. The value of u_x is changed to the optimal value calculated for particular mass M and initial velocity v_0. In Fig. 7. the suboptimal responses of the absorber are shown. Although the character of the reaction force is quite similar to optimal case, some rebounds of the system occur. Depending on the application and operational requirements the designer of the system, which can be equipped with proposed absorber, should decide if such behaviour is acceptable or if additional re-shaping of the vents is necessary.

The performance of the absorber equipped with side shutters mounted at the proper angle will not be presented because the obtained system response has entirely optimal character and only level of reaction force is changed.

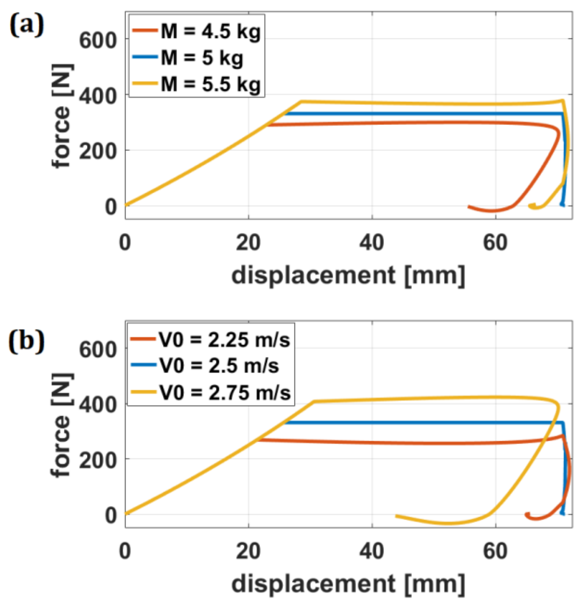

Fig. 7. Suboptimal response of the absorber in case of compression distance adaptation: (a) different masses of the amortized object, (b) different initial velocities of the object.

2.2.4 Experimental tests

For fast experimental validation of the concept of system operation the prototype device was designed and manufactured using 3D printing technology. The first goal was to ensure tightness of cylinders in case of no overlapping area of slot-vent pairs. Simultaneously, efficient relative movement of cylinders had to be provided. To achieve this the manufacturing conditions and tolerances for dimensions of cylinders have been selected carefully, and finally a lubricant was applied on the connection of cylinders. The gas release was obtained by using two slots and two corresponding vents. The values of shock-absorber parameters have been assumed the same as values used in numerical simulations. The only difference is the excitation conditions. For simulation purposes the impact was modelled by the mass with initial velocity, whereas the excitation during laboratory tests was kinematic. In order to provide correspondence between simulations and experiments the applied kinematic excitations had to resemble system kinematics in case of optimal impact absorption process. The prototype of the absorber was mounted in the laboratory test stand as shown in Fig. 8. The kinematic excitation was realized using fast hydraulic actuator. The reaction force of the absorber was measured directly using dedicated force sensor.

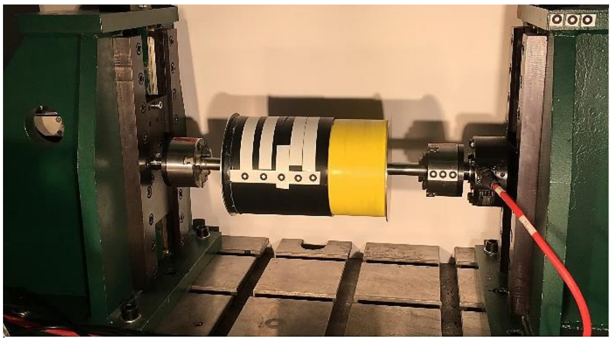

Fig. 8. The prototype of the proposed single-chamber pneumatic shock-absorber during experimental tests.

The set of different kinematic excitations (Fig. 9b) was applied to measure the absorber reaction force (Fig. 9a) in nominal conditions of optimal impact absorption (excitation 2) as well as the suboptimal response in case of different loading conditions. The force-displacement response of the system (Fig. 9c) corresponds well to the results obtained in numerical simulations.

After phase of fast gas compression the reaction force is maintained at almost constant level until the end of absorber stroke. In case of kinematic excitations 1 and 3 the lack of adaptation mechanism results in suboptimal response of the shock-absorber. Nevertheless, the obtained performance is much better than performance achieved by the use of typical absorbers with constant valve opening. The small oscillations of the absorber reaction force correspond probably to elastic deformations of the cylinders. The nonzero value of final reaction force is caused by the remaining overpressure inside the absorber. This results from the fact that the prototype device did not have wide opening for the final exhaust of the gas.

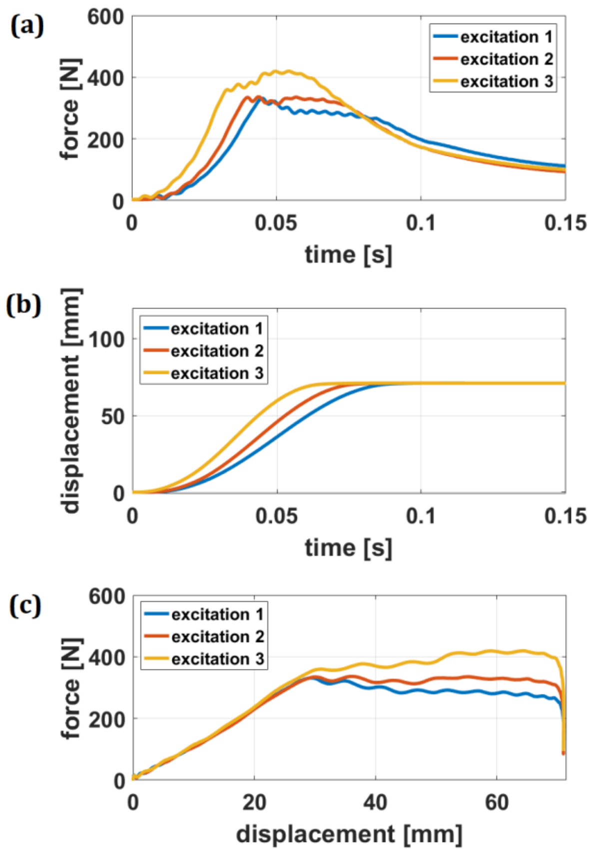

Fig. 9. (a) measured force response of the absorber, (b) applied kinematic excitations, (c) obtained force-displacement characteristics of the absorber.

The conducted experimental tests have proved feasibility of the concept of proposed absorber operation.

2.3 Double-chamber shock-absorber

2.3.1 Device design and adaptation

The idea of obtaining variable gas release using overlapping vents and slots can be extended for the case of double chamber shock-absorber. In this case the more convenient technical solution is a use of metering pins of variable shape and holes placed in the absorber piston. Metering pins play the analogous role as vents in the single-chamber shock-absorber, while holes in the piston correspond to the slots in the single-chamber device.

In Fig. 10a the optimal response of the double-chamber pneumatic shock-absorber is shown. In order to obtain such good response the operation of the device has to be divided into two phases. The first phase corresponds to the fastest possible increase of reaction force due to lack of gas transfer between chambers (Fig. 10b) resulting in compression of the gas located in chamber 1 and decompression of the gas in chamber 2. When the optimal value of compression distance u_x is reached and further maintaining of constant value of reaction force will provide dissipation of entire impact energy within available stroke, the gas transfer between chambers has to be enabled. The area of the valve between chambers in proposed device will be defined as a projection area appearing between metering pins and walls of corresponding holes. The height of the piston influences similarly the performance of the absorber as height of the slot in single-chamber absorber but there is a slight difference. Namely, the first phase of absorber operation lasts until displacement being a sum of u_x and piston height h_p is reached. Only further movement corresponds to gas transfer between the chambers. In case of significant height of the piston, the value h_p can be taken into account during determination of the compression distance u_x. Also the shift of cross section corresponding to the valve area should be considered. Moreover, the process of flow through the canal should be simulated using more detailed models.

In further discussion, the height of the piston h_p will be assumed to be very small. As a result the diameter of metering pin d_{mp}, ensuring appropriate values of valve area $A_v(u)$ created due to movement of piston with holes relative to metering pins, can be calculated using simple formula:

$$d_{mp}^2 = d_h^2 - \frac{4A_v(u)}{n\pi} \tag{5}$$

where d_h is the diameter of holes in the piston.

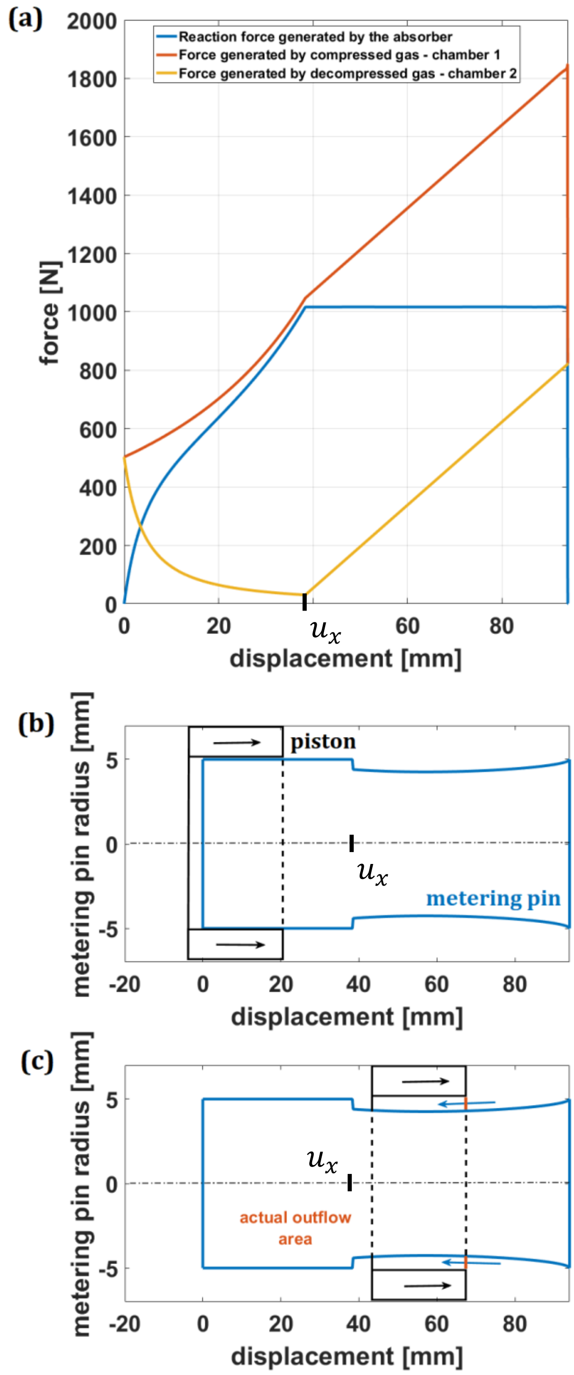

Fig. 10. (a) optimal force response of pneumatic double-chamber shock-absorber, (b) piston movement relative to metering pin during first phase of operation – no gas release, (c) piston movement relative to optimally shaped metering pin during transfer of the gas between chambers.

The procedure of determining the optimal valve area $A_v(u)$ is based on the solution of inverse dynamics problem and it is not presented in this paper which is aimed at development of single reconfiguration method for adaptation to different impact conditions. More information about applied method can be found in [12-14].

Numerical results concerning double-chamber shock-absorber have been obtained for the object of the mass M equal to 10 kg and initial velocity v_0 of 4 m/s. Parameters of the shock-absorber are collected in Tab. 2.

Table 2. Parameters of analyzed system equipped with double-chamber shock-absorber.

M [kg]	v_0 [m/s]	p_0 [kPa]	T_0 [K]
10	4	400	293.15
L [mm]	$\varnothing D$ [mm]	u_0 [mm]	$\varnothing d_h$ [mm]
100	40	6	10

In order to propose a relevant mechanism providing single reconfiguration of the system for adaptation to predicted impact conditions, the shape of metering for different masses of the amortized object and different initial velocities was calculated and presented in Fig. 11.

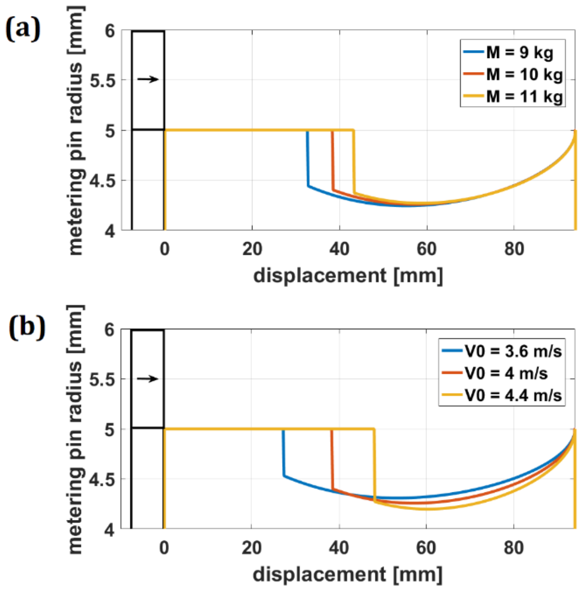

Fig. 11. (a) optimal shapes of metering pin for different masses of amortized object, (b) optimal shapes of metering pin for different initial velocities.

The shape change of metering pin depends more on the value of initial velocity of decelerated object than on the value of object's mass. It is intuitive effect because energy that has to be absorbed and dissipated is a quadratic function of velocity and linear function of mass. The interesting fact is that for masses varied by $\pm 10\%$ about half of the metering pin shape is almost the same and the visible difference is caused by increase or decrease of compression distance. Change of initial velocity of the object results in completely different shape of metering pin.

2.3.2 Metering pin re-shaping for adaptation to different impact excitations

In Fig. 12a the scheme of proposed adaptation mechanism is shown. In order to easily change the shape of metering pin, it should be divided into two main parts:

- a ring which can move along metering pin axis and in result it ensures adjustment of the compression distance,
- a core of metering pin which should be designed as a morphing structure which is able to change its external shape.

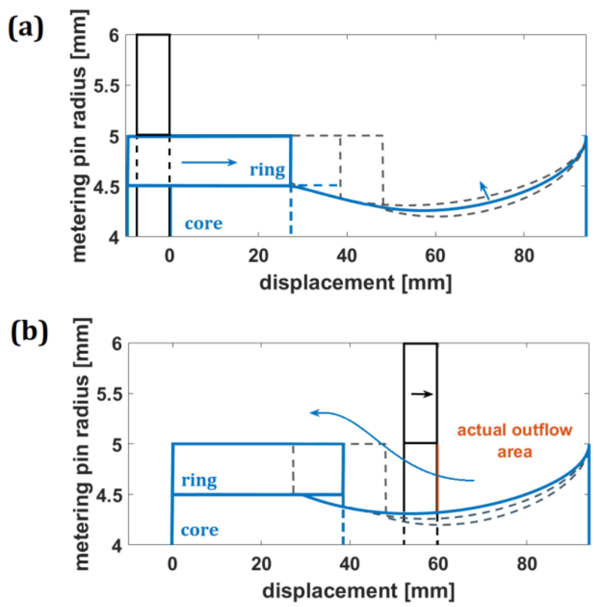

Fig. 12. (a) scheme of the mechanism for adaptation of metering pin shape, (b) gas flow between chambers during second phase of system operation.

Appropriate position of the ring ensures change of compression distance, which plays a significant role because it relates to the level of reaction force that should be maintained possibly constant during second phase of the impact absorption process. If the gas flow between chambers start too early, probably a part of impact energy will not be dissipated and the piston will hit the absorber bottom. In contrast, if gas flow start too late, the objective of absorber optimization (minimization of the reaction force) will not be even approximately fulfilled.

In Fig. 12b the operation of adapted system is shown. In presented example the compression distance was lengthened due to displacement of the ring and the shape change of metering pin core.

2.3.3 Simplified adaptation strategy

In this section, similarly as for single-chamber shock-absorber the analyses of simplified adaptation technique of metering pin is provided. In Fig. 13. the force responses of the absorber excited by the impacts defined by 10% higher values of the object mass *M* and initial velocity v_0 are shown. In case of higher mass value the adaptation of compression distance by the move of ring and no change of the core shape leads to the response close to optimal. In contrast, the increase of initial velocity causes the necessity of shape adjustment of metering pin core.

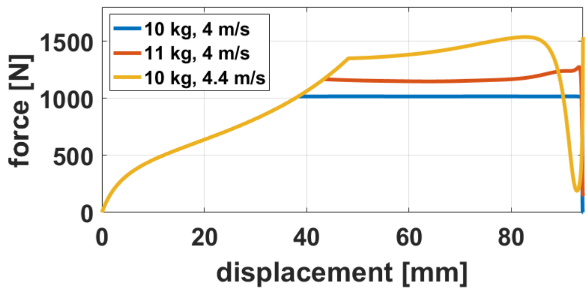

Fig. 13. Force response of double-chamber shock-absorber in case of 10% higher values of the mass or initial velocity – only compression distance adapted.

3 Conclusions

The presented research was aimed at development of high performance pneumatic shock-absorbers which can be used in aeronautical applications. Adaptive capabilities of proposed absorbers have been revealed and simplified adaptation mechanisms were discussed. The obtained response of proposed absorbers is significantly better than response of typical passive dampers with constant valve opening. Moreover, the performance achieved during system adaptation to predicted impact conditions is comparable with performance of smart semi-active devices controlled in real-time. The significant advantage of the proposed solutions is simplicity of their construction and possibility of fail-safe design. The presented general approach to the design of semi-passive pneumatic devices can be applied for elaboration of other types of fluid-based absorbers.

In particular, the following content was presented in the paper:

- elaboration and analyses of adaptation techniques for semi-passive absorbers – single-chamber as well as double-chamber device,
- proposal of simplified adaptation mechanisms and investigations of their influence on the system response,
- experimental study concerning operation of the single-chamber pneumatic shock-absorber.

Further research will concern development and practical implementation of dedicated mechanisms serving for adaptation of the absorber by means of single system reconfiguration. In addition, the author will make an effort to meet the requirement of fail-safe design.

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Copyright Statement

Appendix 2

Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi, wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia zrzucanych ładunków

Przedmiotem wynalazku jest absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi, wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia zrzucanych ładunków. Proponowane urządzenie należy do grupy rozwiązań, które umożliwiają łagodzenie skutków udarów mechanicznych poprzez sprężanie powietrza (absorpcja energii uderzenia) i następnie jego upust (dyssypacja zakumulowanej energii).

Znany jest z amerykańskiego opisu patentowego US5884734 siłownik pneumatyczny zawierający dwie komory, których połączenie może być otwarte lub zamknięte, przy czym połączenie to jest zależne od pozycji elementu uszczelniającego znajdującego się pomiędzy komorami.

Wynalazek US6705642B1 stanowi rozwiązanie umożliwiające sterowanie otwarciem wypływu gazu z poduszki powietrznej poprzez usunięcie połączeń mechanicznych w tkaninie poduszki. Element upustowy ma formę połączenia opartego na zasadzie działania zamka błyskawicznego sterowanego poprzez usunięcie cięgna. Rozwiązanie to nie umożliwia zamknięcia uprzednio otwartego przepływu.

Patent EP00423981A opisuje niesterowalne zawory upustowe do poduszek gazowych, których otwarcie inicjowane jest ciśnieniem gazu poprzez uniesienie elementu konstrukcyjnego zaworu, który ma postać cienkościennej, podatnej membrany. Przekroczenie progowego ciśnienia wewnętrznego powoduje stopniowe unoszenie części membrany powodując zwiększenie wypływu gazu. Rozwiązanie to nie umożliwia sterowania procesem wypływu medium.

Opis patentowy EP00592879A1 opisuje rozwiązania sterowalnych zaworów upustowych, w których regulacja przepływu gazu realizowana jest przez wzajemne przemieszczenie elementów posiadających ukształtowane kanały przelotowe. W stanie zamkniętym elementy przysłony zaworu zasłaniają światło kanału blokując przypływ gazu. Wymuszone przemieszczenie przysłony powoduje otwarcie przepływu gazu w kanałach przelotowych. Opisywana grupa zaworów charakteryzuje się relatywnie dużą masą i niską zwartością konstrukcji oraz potencjalnie dużymi czasami aktywacji.

Z polskiego patentu PL 214845 znany jest adaptacyjny sposób dyssypacji energii uderzenia oraz absorber pneumatyczny. W powyższym sposobie dyssypacji energii na podstawie zidentyfikowanej masy i prędkości uderzającego obiektu wyznacza się optymalny przebieg opóźnienia uderzającego obiektu i odpowiadającej mu siły wywieranej na tłok amortyzatora. Podczas procesu dyssypacji energii uderzenia, po osiągnięciu wymaganej wartości siły utrzymuje się tę siłę na stałym poziomie poprzez sterowany zaworem elektrycznym przepływ powietrza następujący aż do chwili, gdy tłok osiągnie położenie krańcowe. Utrzymanie w przybliżeniu stałej wartości siły działającej na tłok realizowane jest przez naprzemienne otwieranie i zamykanie zaworu elektrycznego lub regulowanie wielkości otworu przelotowego zaworu. Przedmiotem wynalazku jest również absorber pneumatyczny wyposażony w czujniki ciśnienia znajdujące się w każdej z komór, czujniki przyspieszenia i siły kontaktowej zamocowane do tłoczyska oraz sterowalny zawór kontrolujący przepływ powietrza.

Opracowane zostały także rozwiązania dedykowane pewnym aplikacjom jak np. absorber pneumatyczny z patentu US 4030715 przeznaczony do zastosowania w samochodach i innych pojazdach. Z polskiego patentu PL 187957 znana jest natomiast konstrukcja amortyzatora pneumatycznego przeznaczonego w szczególności do roweru.

Istnieją koncepcje i prototypy adaptacyjnych amortyzatorów pneumatycznych, które niestety wymagają zastosowania szybkich zaworów o ciągłym zasilaniu prądem elektrycznym i ciągłym trybie pracy podczas procesu absorpcji i dyssypacji uderzenia. Amortyzatory gazowe ze sterowalnym upuszczaniem powietrza do atmosfery oraz sterowalnym przepływem pomiędzy komorami (absorbery dwukomorowe) były analizowane w kontekście efektywnego tłumienia drgań. Ponadto, proponowane było zastosowanie pneumatycznego tłumika do łagodzenia odpowiedzi dynamicznej budynków poddanych obciążeniu sejsmicznemu. W literaturze naukowej stosunkowo szeroko przedstawiany jest temat zastosowania zaworów piezoelektrycznych do sterowania przepływem gazu, gdzie jednym z ich zastosowań było tłumienie drgań w podwoziu małego samolotu.

Zadaniem wynalazku jest akumulacja energii uderzenia poprzez sprężanie powietrza i dyssypację zgromadzonej energii poprzez upust powietrza. Zarówno sprężanie jak i upust powietrza odbywają się w ściśle zaprojektowany sposób, co umożliwia efektywne zastosowanie absorbera pneumatycznego o adaptowalnej charakterystyce odpowiedzi do zabezpieczenia konstrukcji podlegających udarowi mechanicznemu. Wynalazek składa się z dwóch jednostronnie otwartych cylindrów wstępnie nasuniętych na siebie. Zewnętrzny cylinder jest otwarty od góry i posiada co najmniej jeden poziomy, korzystnie podłużny otwór. Wewnętrzny, otwarty od dołu cylinder posiada taką samą liczbę otworów co cylinder zewnętrzny, a pionowe osie symetrii otworów w tych cylindrach pokrywają się. Przed uderzeniem cylindry są nasunięte na siebie w taki sposób, że otwory cylindrów nie pokrywają się, co uniemożliwia upust powietrza. Względne przemieszczanie cylindrów pod wpływem działającego obciążenia powoduje sprężanie powietrza do wyznaczonej, optymalnej wartości ciśnienia. Odpowiednia wartość ciśnienia powietrza wewnątrz cylindrów uzyskiwana jest dla ściśle określonego przemieszczenia względnego cylindrów. Dalsze przemieszczanie cylindra górnego względem cylindra dolnego prowadzi do nasuwania się otworów cylindrów i umożliwia upust powietrza. Optymalny kształt otworów zapewnia wykorzystanie całego skoku absorbera przy utrzymaniu stałej wartości generowanej siły reakcji. Kształt otworów cylindra wewnętrznego wyznaczany jest poprzez rozwiązanie zadania odwrotnego dynamiki absorbera dla przewidywanych warunków uderzenia. W przypadku, gdy absorber jest używany dla różnych warunków udaru, wystarczy zmienić jedynie wewnętrzny cylinder na cylinder o kształcie otworów dedykowanym dla innych warunków obciążenia. W efekcie uzyskiwane jest pasywne, ale adaptowalne urządzenie o quasi-optymalnej charakterystyce równoważnej zastosowaniu sterowalnych aktywnie zaworów upustowych. Wzajemne usytuowanie i kształt otworów zapewnia utrzymanie stałej siły reakcji absorbera dla określonej charakterystyki udaru, na którą składają się m.in. masa obiektu, prędkość początkowa kontaktu z podłożem oraz jego rodzaj.

Przedmiot wynalazku w przykładowym wykonaniu został przedstawiony na rysunkach. Fig.1 przedstawia rzut izometryczny absorbera pneumatycznego, Fig. 2 stanowi rzut płaski absorbera uwidaczniający usytuowanie komponentów absorbera względem siebie.

W przykładzie wykonania absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi według wynalazku składa się z cylindra **1** dolnego, otwartego od góry oraz cylindra**2** górnego, otwartego od dołu. Cylinder **2** jest umieszczony przesuwnie w cylindrze **1**, przy pomocy rowków **3** znajdujących się w cylindrze **2** i wpustów**4** znajdujących się w cylindrze **1**. Cylinder **1** posiada co najmniej jeden poziomy, korzystnie podłużny otwór **5**, a cylinder **2** co najmniej jeden otwór **6**. W początkowej konfiguracji urządzenia otwory **5** i **6** nie

nachodzą na siebie, a odległość między dolną krawędzią otworu **6** a górną krawędzią otworu **5** wynosi **d**.
Odległość **d** równa jest względnemu przemieszczeniu cylindrów **1** i **2** powodującemu sprężenie powietrza znajdującego się w cylindrach do optymalnej wartości, przy której stała siła reakcji absorbera generowana w dalszej fazie pracy urządzenia, tzn. podczas pokrywania się otworów **5** i **6**,zapewniawykorzystanie całego skoku absorbera równego sumie długości otworu **6** i odległości **d** między krawędziami otworów **5** i **6**.

W przykładzie wykonania w cylindrze **1** dolnym zamontowane są dźwignie **7** przy pomocy trzpieni **8**. Dźwignie **7** połączone są z cięgnami lub linkami **9**, na których końcach znajdują się kołki **11**, umieszczone w uchach **12** na górze cylindra **2** górnego. Cięgna lub linki **9** są podparte prowadnicami **10** zamocowanymi u szczytu cylindra **1**.

Kształt otworu **6** oraz wymiary otworu **5** są tak dobrane dla przewidywanych warunków uderzenia, aby zapewnić utrzymanie stałej siły reakcji absorbera podczas upustu powietrza spowodowanego nachodzeniem na siebie otworów podczas absorpcji i dyssypacji energii udaru. Praca urządzenia rozpoczyna się podczas kontaktu dźwigni **7** z gruntem. Kontakt ten powoduje obrót dźwigni wokół osi trzpieni **8**, co w konsekwencji powoduje pociąganie napiętych cięgien lub linek **9** i wysunięcie się kołków **11** z uch **12** górnego cylindra **2**. Pozwala to na względne przemieszczenia cylindrów. Ruch względny cylindrów powoduje sprężanie powietrza znajdującego się wewnątrz nich, aż do momentu, gdy otwory **5** i **6** zaczynają na siebie nachodzić. Następuje wtedy upust powietrza, aż do całkowitego wyhamowania ruchu względnego cylindrów. Dobrany w procesie optymalizacji kształt otworów **6** i wymiary otworów **5** zapewniają uzyskanie stałej siły reakcji absorbera i wykorzystanie całego skoku absorbera równego sumie długości otworu **6** i odległości **d** między krawędziami otworów **5** i **6**.

Cechą charakterystyczną rozwiązania jest optymalne sprężenie i upust powietrza zapewniające absorpcję i dyssypację energii uderzenia przy zachowaniu stałej siły reakcji absorbera. Optymalność dyssypacji energii uzyskana jest poprzez odpowiedni dobór względnego usytuowania i kształtu par otworów. Kształt otworów wewnętrznego cylindra i ich usytuowanie względem otworów zewnętrznego cylindra zależy od charakterystyki uderzenia i wyznaczane są poprzez rozwiązanie odwrotnego zadania dynamiki absorbera.

Zastrzeżenia patentowe

1. Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi, wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia ładunków znamienny tym, że posiada otwarty od góry cylinder **1** dolny z co najmniej jednym poziomym, korzystnie podłużnym otworem **5**,w którym to cylindrze**1**umieszczony jest przesuwnie, otwarty od dołu cylinder **2** górny z co najmniej jednym otworem **6**, znajdującym się w odległości **d** od otworu **5**.

2. Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi, wypełniony powietrzem atmosferycznym zwłaszcza do łagodzenia przyziemienia ładunków zgodnie z zastrz. 1 znamienny tym, że cylinder **1** dolny posiada zamocowane na trzpieniach **8** dźwignie **7** z cięgnami lub linkami **9** przechodzącymi prowadnicami **10**, a na końcu tych cięgien lub linek są kołki **11** umieszczone początkowo w uchach **12** przymocowanych do szczytu cylindra **2** górnego.

3. Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia ładunków zgodnie zzastrz. 1 znamienny tym, że osie pionowe otworów **5** i **6** pokrywają się.

4. Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia ładunków zgodnie z zastrz. 1znamienny tym, że odległość początkowa **d** pomiędzy krawędzią dolną otworu **6** cylindra **2** i krawędzią górną otworu **5** cylindra **1** równa jest względnemu przemieszczeniu cylindrów **1** i **2** powodującemu sprężenie powietrza znajdującego się w cylindrach do optymalnej wartości, przy której stała siła reakcji absorbera podczas pokrywaniasię otworów **5** i **6** zapewnia wykorzystanie całego skoku absorbera równego sumie długości otworu **6** i odległości **d** między krawędziami otworów **5** i **6**.

5. Absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia ładunków zgodnie z zastrz. 1znamienny tym, że kształt otworów **6** i wymiary otworów **5** zapewniają uzyskanie stałej wartości siły reakcji absorbera i wykorzystanie całego skoku absorbera równego sumie długości otworu **6** i odległości **d** między krawędziami otworów **5** i **6**.

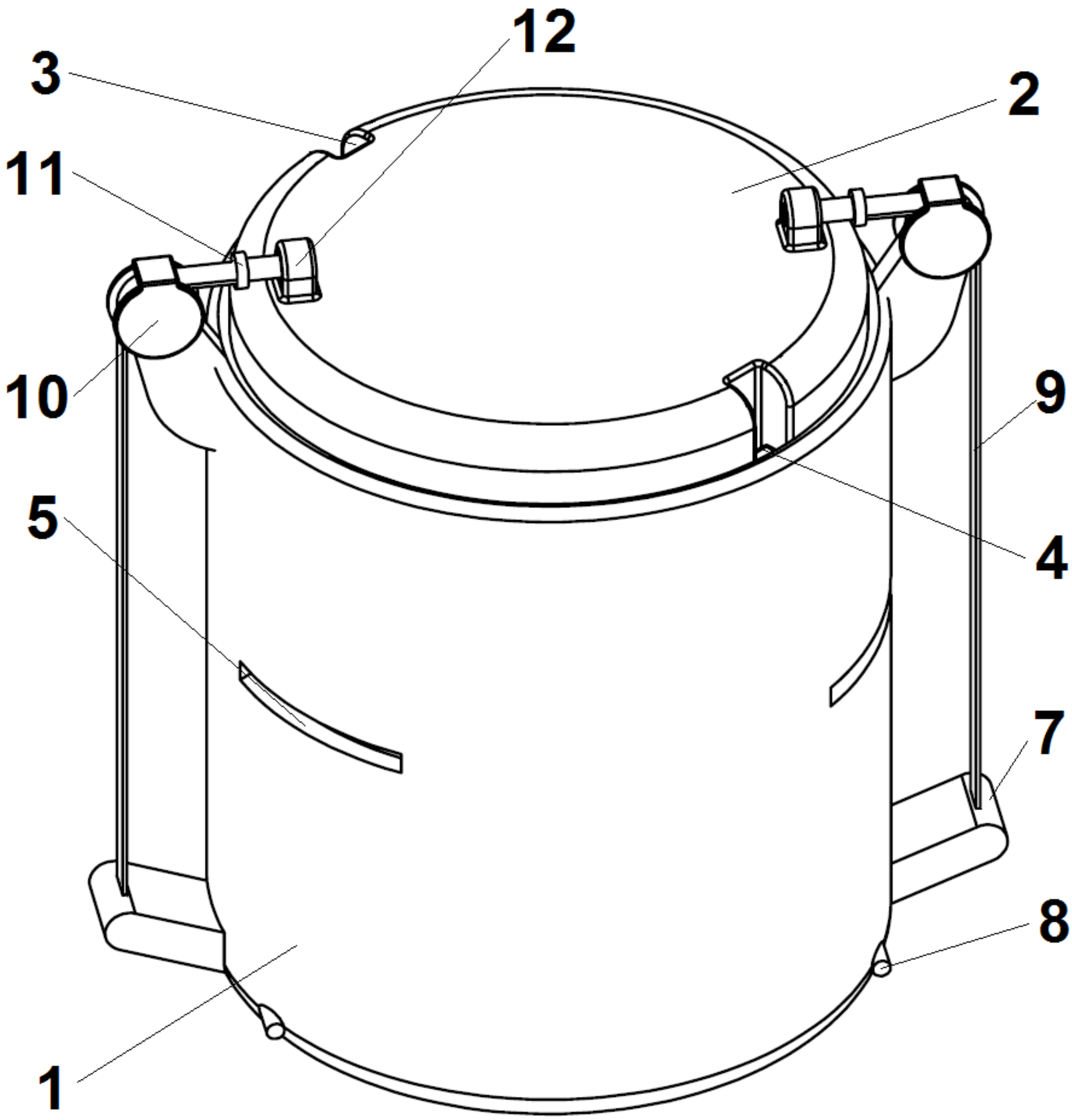

Fig..1 Rzut izometryczny absorbera pneumatycznego

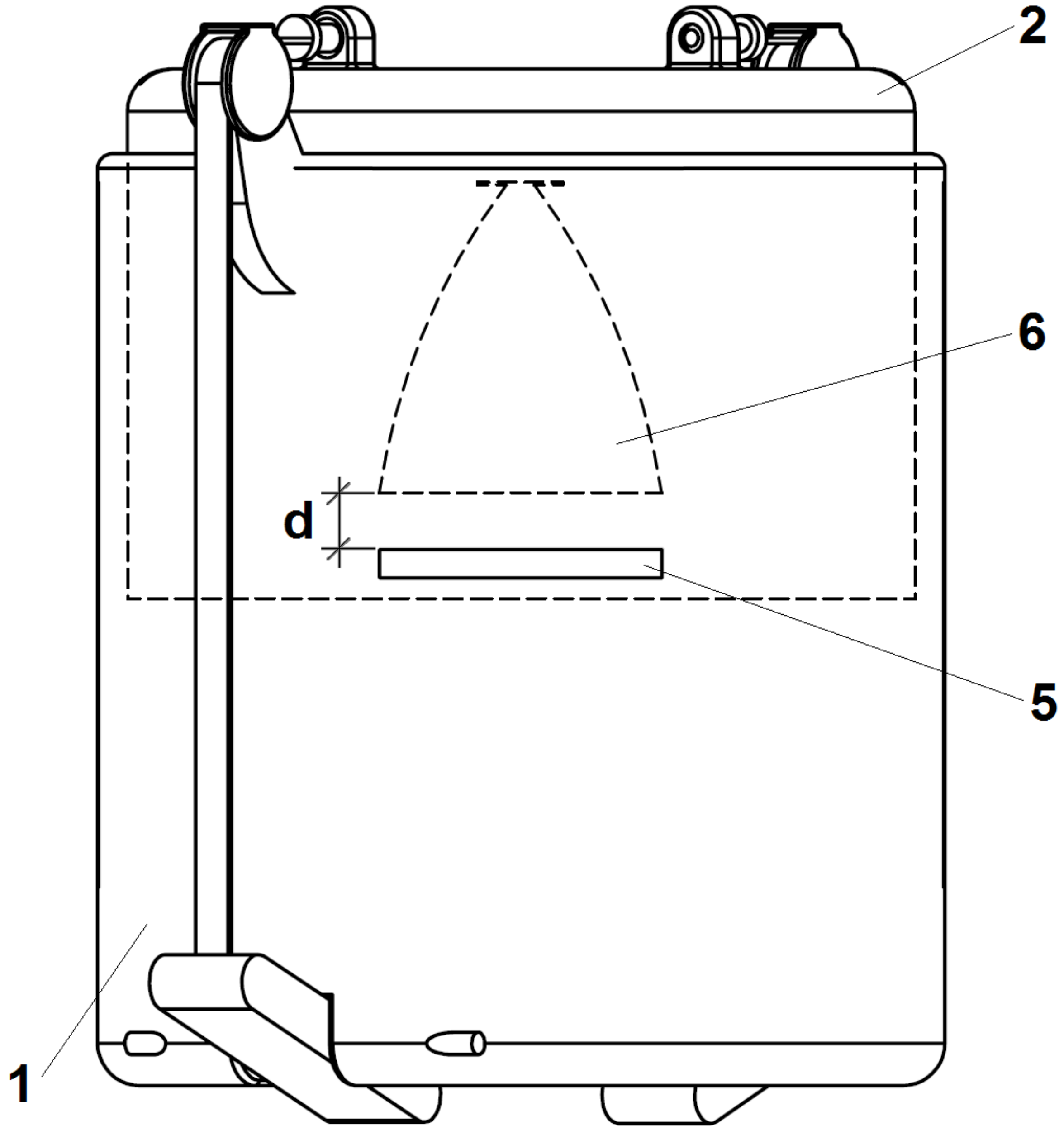

Fig.2 Rzut płaski absorbera-pneumatycznego

Skrót

Przedmiotem wynalazku jest absorber pneumatyczny o adaptowalnej charakterystyce odpowiedzi, wypełniony powietrzem atmosferycznym, zwłaszcza do łagodzenia przyziemienia ładunków, posiadający otwarty od góry cylinder (**1**) dolny z co najmniej jednym poziomym, korzystnie podłużnym otworem (**5**),w którym to cylindrze (**1**) umieszczony jest przesuwnie, otwarty od dołu cylinder górny (**2**) z co najmniej jednym otworem (**6**), znajdującym się w odległości **d** od otworu (**5**). Cylinder (**1**) dolny posiada zamocowane na trzpieniach (**8**) dźwignie (**7**) z cięgnami lub linkami (**9**) przechodzącymi prowadnicami (**10**), a na końcu tych cięgien lub linek znajdują się kołki (**11**) umieszczone początkowo w uchach (**12**) przymocowanych do szczytu cylindra (**2**) górnego.

(5 zastrzeżeń patentowych)

Appendix 3

Absorber śrubowy o zmiennym skoku gwintu tocznego

Przedmiotem wynalazku jest absorber śrubowy o zmiennym skoku gwintu tocznego, którego głównym zastosowaniem jest absorpcja energii uderzeń i tłumienie drgań.

Siła reakcji proponowanego urządzenia wynika z efektów bezwładnościowych i geometrii gwintu kulowego, a więc można zaklasyfikować je do grupy tzw. „inerterów". Cechą wyróżniającą proponowany układ jest to, że geometria śruby o gwincie tocznym, niezależnie od prędkości początkowej wymuszenia, zapewnia uzyskanie stałej siły reakcji absorbera. Znane jak dotąd układy tego typu zapewniają uzyskanie siły reakcji proporcjonalnej do różnicy przyspieszeń na ich końcach. W efekcie nie są one w stanie zapewnić stałej wartości siły reakcji podczas hamowania obiektu. Nadają się za to bardzo dobrze do akumulacji energii mechanicznej i stosowane są m.in. w dynamicznych eliminatorach drgań, gdzie energia drgającego układu przepływa przez dodatkowy układ inercyjno-sprężysto-tłumiący, co skutkuje efektywną redukcją amplitudy drgań wymuszonych. Zupełnie inną, znaną grupą urządzeń są tłumiki śrubowe, które zapewniają efektywną dyssypację energii poprzez wydłużenie efektywnej drogi, na której dochodzi do rozpraszania energii poprzez tarcie, efekty lepkościowe lub inne mechanizmy dyssypacji energii.

W artykule M.Z.Q. Chen'a, C. Papageorgiou, F. Scheibe, F.C. Wang'a i M. Smith'a pt. „The missing mechanical circuit element" opublikowanego w 2009 r., w IEEE Circuits and Systems Magazine, tom 9 nr 1, str. 10-26, można znaleźć dwa rozwiązania mechaniczne zapewniające uzyskanie układu bezwładnościowego o charakterystyce inertera, czyli zapewniające proporcjonalność siły reakcji do różnicy przyspieszeń dwóch terminali urządzenia. Zaprezentowane inertery oparte są odpowiednio na układzie zębatki z kołem zębatym i kołem zamachowym oraz układzie śruby kulowej z nakrętką i kołem zamachowym. W literaturze naukowej można natknąć się również na inne realizacje techniczne tego typu układów np. inertery cieczowe, w których rolę wirującej masy pełni ciecz przepychana przez przewód, najczęściej o kształcie spiralnym.

Jak wcześniej wskazano, inertery najczęściej wykorzystywane są w zagadnieniach tłumienia drgań. Urządzenie typu inerter zaprojektowane do łagodzenia odpowiedzi udarowej zostało przedstawione w opisie patentowym PL 227058. Patent przedstawia rozwiązanie techniczne, w którym ruch posuwisty śruby wykorzystywany jest do akumulacji energii w pierścieniach o przeciwnych gwintach. Energia wymuszenia akumulowana jest początkowo w zewnętrznym pierścieniu, a następnie następuje rozprzęgnięcie śruby z wewnętrznym pierścieniem, który jest napędzany w kierunku przeciwnym do zewnętrznego pierścienia, sprzęganego jednocześnie ze środkowym pierścieniem. Pomiędzy wewnętrznym i środkowym pierścieniem znajduje się ciecz magneto-reologiczna. W efekcie energia zakumulowana w zewnętrznym pierścieniu jest rozpraszana poprzez oddziaływanie pierścieni obracających się w przeciwnych kierunkach. Dyssypacja energii jest kontrolowana przy użyciu czujnika przemieszczenia oraz elektromagnesów wpływających na lepkość cieczy magneto-reologicznej.

W opisie EP 1510721 przedstawiono amortyzator, oparty na mechanizmie śrubowym z nakrętką kulkową, w którym ruch teleskopowy jest przekształcany na ruch obrotowy, używany do napędu silnika, generującego siłę magnetomotoryczną powodującą wytłumienie drgań. Ze zgłoszeń JP 2013024256 i US 5449054 znane są tłumiki rotacyjne oparte na lepkich płynach, w których obrotowe rotory zapewniają efektywną dyssypację energii. Znane są także amortyzatory śrubowe przedstawione w opisach PL 229926 i PL 230102, gdzie siły tłumiące powstają w wyniki mieszania materiałów sypkich, a uzyskiwane charakterystyki zależą zarówno od zastosowanych materiałów roboczych jak i geometrii gwintu oraz elementów mieszkających.

W porównaniu z proponowanym wynalazkiem, wszystkie z wyżej przedstawionych rozwiązań technicznych charakteryzują się innym sposobem generowania siły reakcji absorbera, tj. poprzez efekty bezwładnościowe lub efekty czysto dyssypacyjne takie jak lepkość czy tarcie. W prezentowanym wynalazku generowana siła reakcji wynika z efektów bezwładnościowych, przy czym istotną rolę przyjęła geometria gwintu tocznego. Dzięki zmienności gwintu pojawiająca się siła reakcji zapewnia efektywne hamowanie obiektu oddziaływującego na śrubę absorbera.

Istota wynalazku polega na tym, że absorber śrubowy o zmiennym skoku gwintu tocznego złożony jest ze śruby o malejącym skoku gwintu oraz łożyskowanej nakrętki kulkowej. Zmiana skoku gwintu tocznego powoduje powstanie dodatkowej składowej siły zależnej od prędkości ruchu postępowego śruby. W rezultacie siła reakcji absorbera F, składa się z dwóch głównych członów zgodnie z ogólnym wzorem:

$$F = b(x-y)(\ddot{x}-\ddot{y}) + f(x-y)(\dot{x}^2-\dot{y}^2) \qquad (1)$$

Pierwszy składnik siły reakcji absorbera odpowiada sile reakcji typowego inertera, tzn. sile proporcjonalnej do różnicy przyspieszenia liniowego $\ddot{x}$ śruby i przyspieszenia liniowego $\ddot{y}$ obudowy, z tym, że współczynnik b nazywany „inertancją" jest funkcją różnicy przemieszczeń liniowych x śruby i y obudowy. Drugi człon siły reakcji F jest wynikiem zmienności skoku gwintu śruby kulowej i jest proporcjonalny do różnicy kwadratów prędkości liniowych $\dot{x}$ śruby i $\dot{y}$ obudowy oraz funkcji f zależnej od różnicy przemieszczeń liniowych x śruby i y obudowy.

Rozwiązanie według wynalazku charakteryzuje się kompaktową konstrukcją zapewniającą uzyskanie w pasywny sposób optymalnej charakterystyki hamowania obiektu amortyzowanego lub uderzającego. Jego zakres aplikacji jest bardzo szeroki i może obejmować zarówno łagodzenie skutków uderzeń i drgań w różnego typu urządzeniach i pojazdach, jak i poprawę odpowiedzi konstrukcji budowlanych narażonych na wymuszenia udarowe i harmoniczne.

Przedmiot wynalazku w przykładowym wykonaniu został opisany poniżej i pokazany na rysunku, na którym Fig. 1 przedstawia widok aksonometryczny absorbera śrubowego o zmiennym skoku gwintu tocznego, zaś Fig. 2. stanowi wizualizację geometrii przykładowej bruzdy gwintu tocznego, wyznaczonej w wyniku symulacji numerycznej.

Śruba **1** kulowa posiada co najmniej dwie spiralne bruzdy **2** gwintu **3** tocznego i połączona jest z nakrętką **4** poprzez kulki **5**. Nakrętka **4** łożyskowana jest w cylindrycznej obudowie **6** przy użyciu łożysk **7** i **8**. Korzystnie, na początku długości śruby bruzdy **2** są równoległe do osi śruby **1** (patrz Fig. 2), co odpowiada nieskończenie dużej wartości skoku gwintu **3** i pozwala na wyeliminowanie dodatkowego elementu kontaktowego, typowo używanego w celu uniknięcia zderzenia niesprężystego w początkowej fazie absorpcji uderzenia. Wraz z długością śruby **1**, skok gwintu **3** maleje do określonej, małej wartości, a więc linia spiralna bruzd **2** ulega stopniowemu zagęszczaniu. Ruch posuwisty śruby **1** pod wpływem działającego wymuszenia powoduje stopniowe rozpędzanie nakrętki **4** w związku z malejącym skokiem gwintu zgodnie z kształtem bruzd **2**. Wyhamowanie obiektu amortyzowanego i śruby **1** następujące w momencie osiągnięcia skrajnego położenia śruby **1** względem obudowy **6**. Korzystnie, geometria bruzd **2** wyznaczona jest poprzez rozwiązanie zagadnienia odwrotnego dynamiki i pozwala na uzyskanie stałej wartości siły reakcji absorbera. W przypadkach nietypowych obciążeń dopuszcza się niemonotoniczną zmianę skoku gwintu **3**, tzn. linia spiralna bruzd **2** może na pewnych odcinkach śruby podlegać rozgęszczać się.

Przedstawiony przykład wykonania jest szczególnie uzasadniony w przypadku działania absorbera w orientacji poziomej. W przypadku użycia absorbera do łagodzenia uderzeń w orientacji pionowej, należy rozważyć użycie mechanizmu blokującego ruch względny śruby względem obudowy przed wystąpieniem wymuszenia dynamicznego. Przykładowo, można użyć element ulegający zniszczeniu w sposób przełamania po wystąpieniu wymuszenia. Rozwiązanie techniczne mechanizmu blokującego ruch początkowy nie jest przedmiotem opisanego wynalazku.

Przykład wykonania absorbera podany jest jedynie w charakterze nieograniczających wskazań dotyczących wynalazku i nie może w żaden sposób ograniczać zakresu ochrony, który jest określony poprzez zastrzeżenia patentowe.

Zastrzeżenia patentowe

1. Absorber śrubowy o zmiennym skoku gwintu tocznego, **znamienny tym**, że posiada śrubę **1** kulową o co najmniej dwóch bruzdach **2** gwintu **3** tocznego o zmiennym skoku oraz nakrętkę **4** kulową połączoną ze śrubą **1** poprzez kulki **5**, przy czym nakrętka **4** jest łożyskowana w obudowie **6** przy użyciu łożysk **7** i **8**.

2. Absorber śrubowy o zmiennym skoku gwintu tocznego według zastrz. 1, **znamienny tym**, że skok gwintu tocznego **3** maleje wraz z długością śruby **1** kulowej.

3. Absorber śrubowy o zmiennym skoku gwintu tocznego według zastrz. 1, **znamienny tym**, że bruzda **2** jest początkowo równoległa do osi śruby.

4. Absorber śrubowy o zmiennym skoku gwintu tocznego według zastrz. 1, **znamienny tym**, że skok gwintu **3** tocznego maleje w taki sposób, że siła reakcji absorbera pozostaje stała aż do zatrzymania hamowanego obiektu.

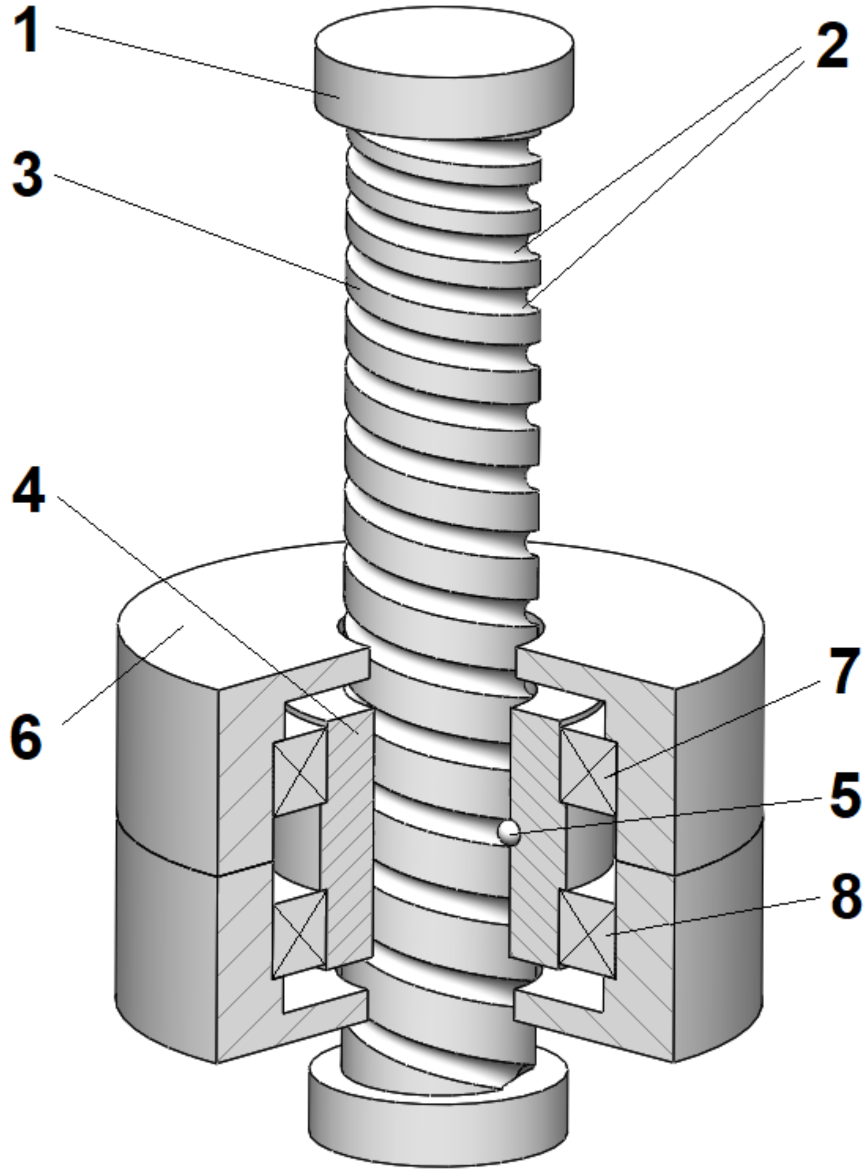

Fig. 1.

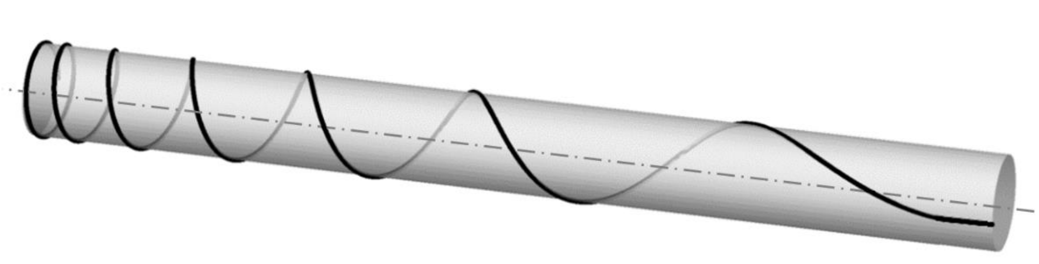

Fig. 2.

Skrót opisu

Wynalazek służy pasywnemu łagodzeniu wymuszeń dynamicznych i zapewnia on uzyskanie optymalnego hamowania obiektu niezależnie od jego prędkości początkowej. Zgodnie z wynalazkiem absorber śrubowy o zmiennym skoku gwintu tocznego posiada śrubę **1** kulową o co najmniej dwóch bruzdach **2** gwintu **3** tocznego o malejącym skoku i nakrętkę **4** kulową łożyskowaną w obudowie **6**.

Appendix 4

RZECZPOSPOLITA
POLSKA

Urząd Patentowy
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(54) **Tłumik bezwładnościowy drgań**

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PL 227058 B1

Opis wynalazku

Przedmiotem wynalazku jest tłumik bezwładnościowy do tłumienia drgań wywołanych udarem.

Istnieje wiele rodzajów tłumików mechanicznych, będących pasywnym zabezpieczeniem urządzeń narażonych na uderzenia i podlegających drganiom lub obciążeniom harmonicznym. Charakterystyki ich często dedykowane są określonym zastosowaniom, a zakres ich stosowalności jest ograniczony.

Znany jest z opublikowanego opisu EP 1510721 amortyzator, oparty na mechanizmie śrubowym z nakrętką kulkową, przekształcającym ruch teleskopowy na ruch obrotowy, używany do napędu silnika, w którym generowana jest siła magnetomotoryczna wykorzystywana do wytłumienia drgań.

Przedstawione w opisie CN 203702983 urządzenie służące tłumieniu drgań w układzie przeniesienia napędu samochodu to podwójne koło zamachowe z cieczą magneto-reologiczną o zmiennym tłumieniu dopasowywanym do warunków nisko- i wysoko – częstotliwościowych drgań poprzez sterowanie zmiennym polem magnetycznym wytwarzanym przy użyciu cewek.

Opublikowany opis CN102052423 przedstawia tłumik drgań skrętnych, włączany pomiędzy urządzenia napędzane i napędzające, który wykorzystuje możliwość zmiany własności wiskotycznych cieczy magnetoreologicznej, przy użyciu zmian pola magnetycznego, wytwarzanego przy użyciu cewek i sterowanego poprzez układ elektroniczny, na podstawie wskazań czujnika kątowego, co pozwala na kontrolowanie przekazywanych obrotów i momentu napędzającego.

Przedmiotem wynalazku jest tłumik bezwładnościowy do tłumienia udaru, którego istota polega na tym, że posiada trzpień, przejmujący energię uderzenia, umieszczony przesuwnie w ramie i wyposażony w tuleję o śrubowej powierzchni zewnętrznej, z gwintem niesamohamownym. Na tuleję jest nakręcony pierwszy pierścień o walcowej powierzchni zewnętrznej, umieszczony obrotowo w drugim pierścieniu, posiadającym śrubową powierzchnię zewnętrzną o linii śrubowej przeciwnej do linii śrubowej tulei. Z drugim pierścieniem współpracuje poprzez połączenie gwintowe trzeci pierścień o śrubowej powierzchni wewnętrznej, wyposażony dodatkowo w rozsuwalnie osadzone ciężarki, które zwiększają moment bezwładności wirującej masy. Korzystnie ciężarki zawieszone są na ramionach osadzonych na wspomnianym trzecim pierścieniu przegubowo. Z ramionami są połączone, za pośrednictwem drugiego przegubu, prowadniki, których swobodny koniec jest osadzony przesuwnie w rowku trzeciego pierścienia.

Pierwszy pierścień i drugi pierścień posiadają w pobliżu obu powierzchni czołowych kołowe, symetrycznie usytuowane wybrania tworzące przy czołach wspomnianych pierścieni kanaliki wypełnione cieczą magneto-reologiczną. Na zewnątrz kanalików znajdują się elektromagnesy. Pierwszy i drugi pierścień mają także od dolnego czoła wspólne łożysko ślizgowe. Pomiędzy powierzchniami łączącymi tuleję z pierwszym pierścieniem jest pierwszy piezo-bloker, a pomiędzy powierzchniami łączącymi drugi pierścień z trzecim pierścieniem jest drugi piezo-bloker. Pomiędzy trzecim pierścieniem i ramą jest drugie łożysko ślizgowe z blokadą ruchu poprzecznego. Piezo-blokery oraz elektromagnesy są połączone ze sterownikiem, z którym połączony jest także czujnik przemieszczenia.

Prezentowane rozwiązanie tłumika bezwładnościowego jest rozwiązaniem szeroko skalowalnym, o dużym zakresie stosowalności oraz możliwości adaptacyjnej zmiany charakterystyk tłumienia drgań i obciążeń harmonicznych.

Tłumik według wynalazku został przedstawiony w przykładzie wykonania na rysunku, na którym fig. 1 przedstawia rzut aksonometryczny tłumika, zaś fig. 2 – przekrój wzdłużny bezwładnika, a fig. 3 – – szczegół połączenia skrajnych części pierścieni w powiększonej skali.

Tłumik składa się z trzpienia **1**, umieszczonego przesuwnie w ramie **2** i połączonego stabilnie z tuleją **3**, o powierzchni zewnętrznej w postaci nie-samohamownej linii śrubowej o gwincie „prawoskrętnym”, oraz z zespołu bezwładnika **4**.

Bezwładnik **4** złożony jest z pierścieni **5**, **6** i **7**, gdzie pierwszy pierścień **5** jest nakręcony na tuleję **3**. Na pierwszy pierścień **5** nałożony jest obrotowo drugi pierścień **6**. Pierwszy pierścień **5** i drugi pierścień **6** posiadają w pobliżu obu powierzchni czołowych kołowe, symetrycznie usytuowane, wybrania tworzące przy czołach wspomnianych pierścieni kanaliki **12** wypełnione cieczą magneto-reologiczną **13**. Drugi pierścień **6** i trzeci pierścień **7** są sprzęgnięte ze sobą powierzchniami **8**, o linii śrubowej ze zwojami nie-samohamownymi, o gwincie „lewoskrętnym”. Pomiędzy tuleją **3** a pierwszym pierścieniem **5** jest pierwszy piezo-bloker **9**, a pomiędzy drugim pierścieniem **6** i trzecim pierścieniem

7 jest drugi piezo-bloker **10**. Pierwszy pierścień **5** i drugi pierścień **6** mają także od dolnego czoła, wspólne łożysko ślizgowe **15**.

Z trzecim pierścieniem 7 połączone są, za pośrednictwem ramion **16** i prowadników **17**, dwa ciężarki **18'**, przy czym końce ramion **16** zamocowane są, odpowiednio, do ciężarków **18'** i do pierwszych przegubów **19** umieszczonych na trzecim pierścieniu **7**. Prowadniki **17** połączone są z ramionami **16** poprzez drugie przeguby **21**, zaś ich drugi koniec umieszczony jest przesuwnie w rowkach **20**.

Ponadto trzeci pierścień **7** połączony jest z poprzeczką **22** ramy **2** poprzez łożysko ślizgowe **23** z blokadą ruchu poprzecznego, znajdujące się w pierścieniowym wybraniu **24** w trzecim pierścieniu **7**.

Na trzpieniu **1** znajduje się czujnik przemieszczenia **25**, połączony poprzez sterownik **26** z piezo-blokerami **9** i **10** oraz elektromagnesami **14**.

Tłumik bezwładnościowy posiada trzy strefy dyssypacji energii:

- Połączenie śrubowe pierścieni **6** i **7** przy nacisku i ruchu drugiego pierścienia **6** powoduje powstanie sił tarcia w kierunku przeciwnym do kierunku ruchu pierścienia **7**;
- Połączenie śrubowe tulei **3** z bezwładnikiem **4** przy nacisku powoduje, analogicznie jak wyżej, rozpraszanie energii poprzez tarcie powierzchni podczas ruchu tulei **3** w kierunku przeciwnym do kierunku ruchu pierścienia **5** bezwładnika **4**;
- Połączenie pierwszego pierścienia **5** z drugim pierścieniem **6** za pośrednictwem cieczy magneto-reologicznej **13**, o sterowanych przez elektromagnesy **14** właściwościach lepkościowych stwarza przy ruchu względnym pierścieni **5** i **6** opory tarcia pomiędzy tymi powierzchniami.

Pod wpływem udaru rozpoczyna się osiowe przesuwanie trzpienia **1**, powodujące ruch obrotowy masy zwanej bezwładnikiem, poprzez połączenie gwintowe między tuleją **3** związaną z trzpieniem **1** a bezwładnikiem. W sytuacji przekroczenia określonego poziomu przemieszczenia się trzpienia **1**, monitorowanego przy pomocy czujnika **25**, sterownik **26** powoduje włączenie pola magnetycznego, wytwarzanego przez elektromagnesy **14**, usztywniającego ciecz magneto-reologiczną **13** i odblokowanie drugiego piezo-blokera **10** w przypadku ściskania trzpienia **1** oraz pierwszego piezo-blokera **9** w przypadku jego rozciągania. Następnie, w chwili spadku prędkości narastania przemieszczenia poniżej określonego progu, sterownik **26** dokonuje wyłączenia pola magnetycznego, upłynniając ciecz magneto-reologiczną **13**, oraz przełączenia piezo-blokerów **9** i **10**. Kolejną czynnością wykonywaną przez sterownik **26** jest ponowne, stopniowe usztywnienie cieczy magneto-reologicznej **13**.

Sposób adaptacyjnej redukcji drgań, przy wykorzystaniu przedstawionego tłumika bezwładnościowego, polega na, zsynchronizowanym z pomiarami czujnika przemieszczenia **25**, umieszczonego na trzpieniu **1**, przełączaniu piezo-blokerów **9** i **10** i sterowaniu polem magnetycznym, wytwarzanym przez elektromagnesy **14**, pobudzającym ciecz magnetoreologiczną **13** tak, aby uzyskać maksymalny efekt dyssypacji energii skumulowanej w pierścieniach **5**, **6** i **7**.

Zastrzeżenia patentowe

1. Tłumik drgań wywołanych udarem zawierający ciecz magneto-reologiczną, **znamienny tym**, że posiada trzpień (**1**), przejmujący uderzenie, umieszczony przesuwnie w ramie (**2**) i wyposażony w tuleję (**3**) o śrubowej powierzchni zewnętrznej, na którą jest nakręcony pierwszy pierścień (**5**) o walcowej powierzchni zewnętrznej, umieszczony w drugim pierścieniu (**6**), mającym śrubową powierzchnię zewnętrzną o linii śrubowej przeciwnej do linii śrubowej tulei (**3**), z którym współpracuje trzeci pierścień (**7**) o śrubowej powierzchni wewnętrznej, wyposażony w rozsuwalnie osadzone obciążniki (**18**), przy czym pierścienie (**5**) i (**6**) posiadają w pobliżu powierzchni czołowych kołowe, symetrycznie usytuowane, wybrania tworzące kanaliki (**12**) wypełnione cieczą magneto-reologiczną (**13**), oraz mają od jednego czoła, pomiędzy wspomnianymi pierścieniami (**5**) i (**6**), łożysko (**15**) ślizgowe, zaś na powierzchniach łączących tuleję (**3**) z pierwszym pierścieniem (**5**) oraz drugi pierścień (**6**) z trzecim pierścieniem (**7**) znajdują się piezo-blokery (**9**) i (**10**), a pomiędzy trzecim pierścieniem (**7**) a ramą (**2**) jest drugie łożysko (**23**) ślizgowe, ponadto pomiędzy pierwszym pierścieniem (**5**) a drugim pierścieniem (**6**) są umieszczone elektromagnesy (**14**).
2. Tłumik drgań według zastrz. 1, **znamienny tym**, że obciążnik (**18**) tworzą dwa ciężarki (**18'**), zawieszone na ramionach (**16**) osadzonych w przegubach, a ze wspomnianymi ramionami

(**16**) są połączone, za pośrednictwem drugiego przegubu (**21**), prowadniki (**17**), których swobodny koniec jest osadzony przesuwnie w trzecim pierścieniu (**7**).

3. Tłumik drgań według zastrz. 1, **znamienny tym**, że elektromagnesy (**14**), piezo-blokery (**9**) i (**10**) są połączone ze sterownikiem (**26**), który połączony jest także z czujnikiem przemieszczenia (**25**).

Rysunki

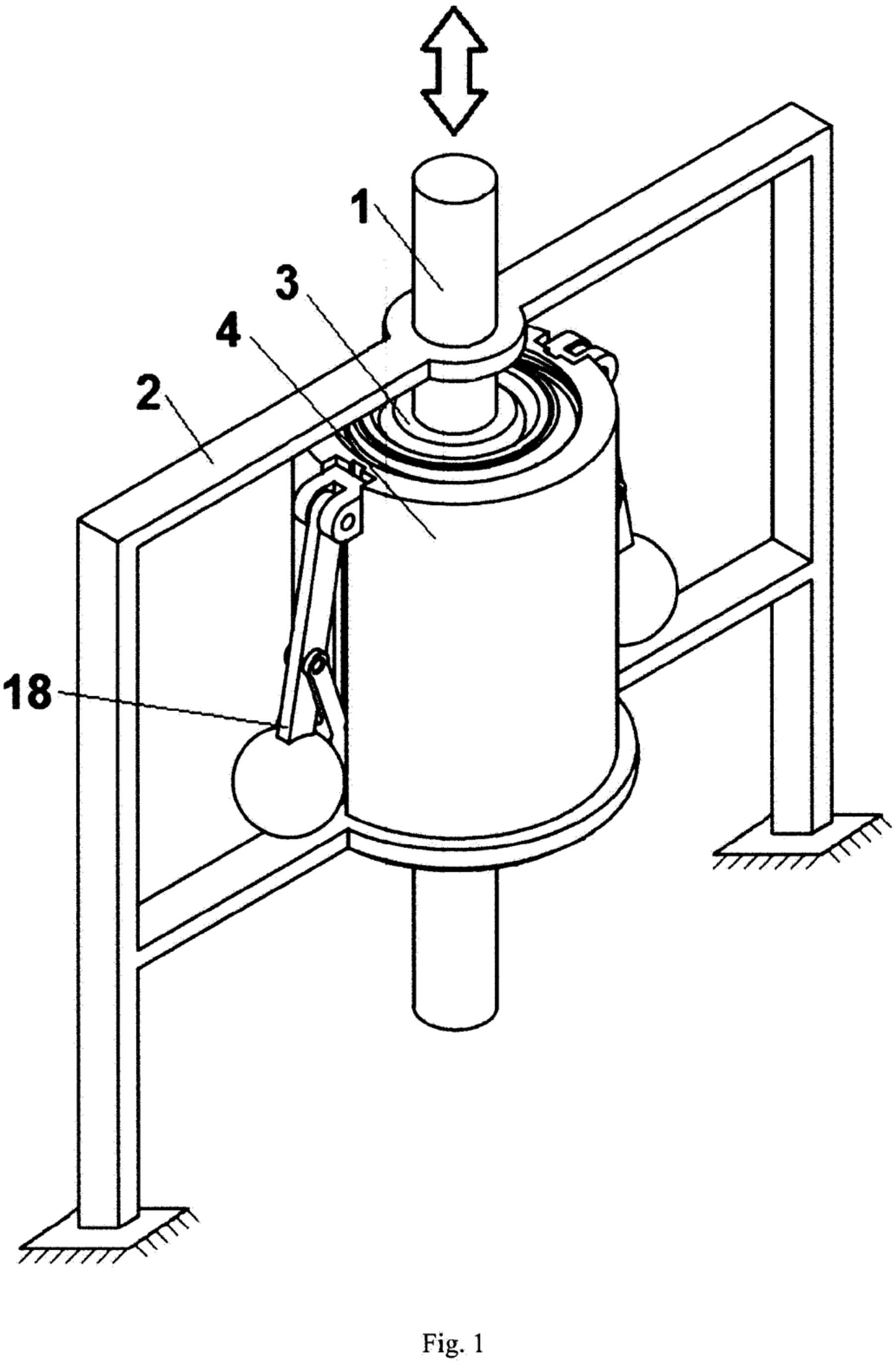

Fig. 1

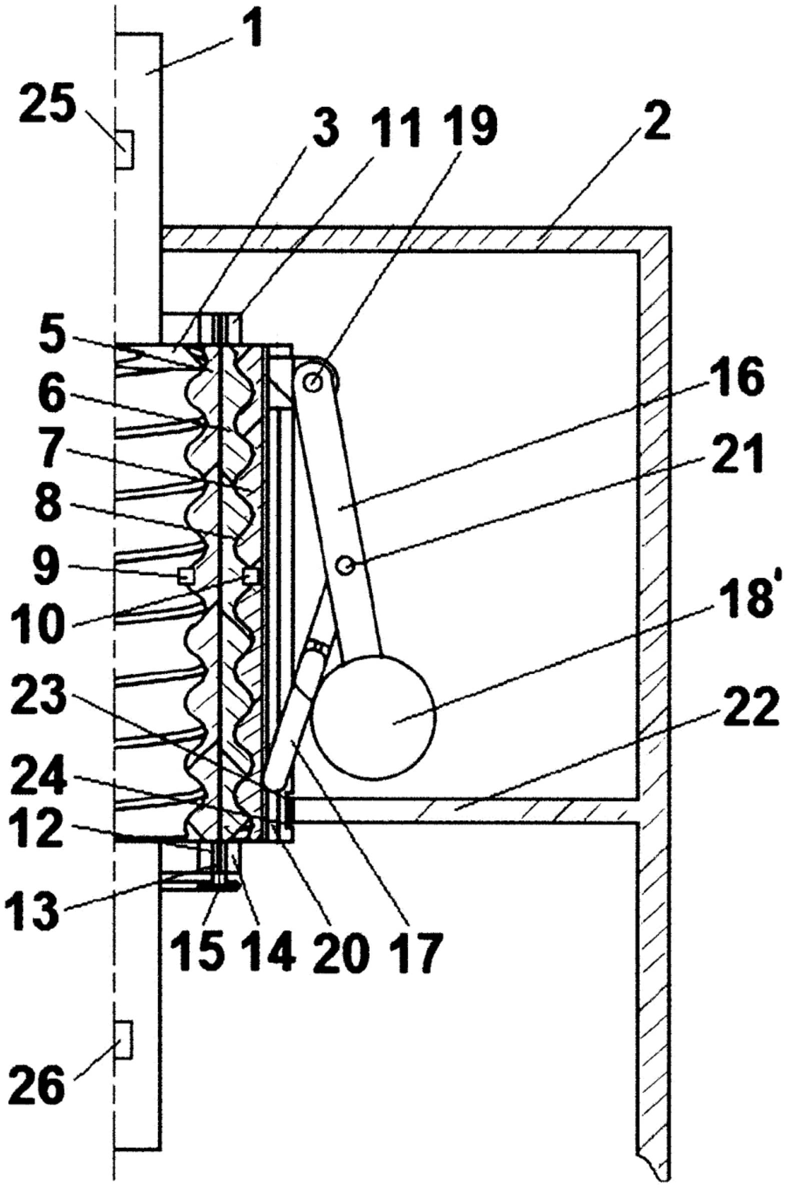

Fig. 2

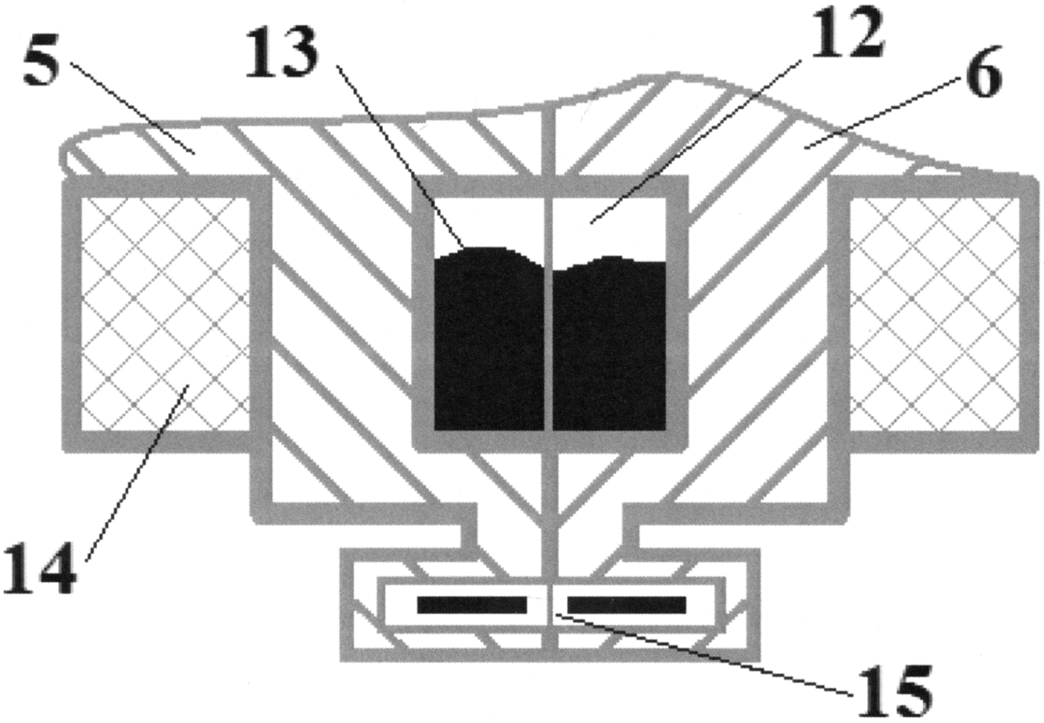

Fig. 3

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Appendix 5

Self-adaptive fluid-based absorbers for impact mitigation and vibration damping

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Abstract

The paper presents development of innovative, self-adaptive, fluid-based absorbers and investigation of their application for mitigation of impacts and forced vibrations. The considered absorbers are composed of two chambers filled with fluid and separated by a piston equipped with a controllable valve. The valve enables control of the fluid flow between the chambers and adjustment of the actual value of force generated by the absorber. The aim of the research is to develop the strategy of valve control providing self-adaptive operation of the absorber ensuring dissipation of submitted energy by using minimal value of generated force. The paper includes description of self-adaptive impact absorber, presentation of the control system and numerical simulation of its effectiveness in the case of impact excitation and harmonic loading. It is concluded that self-adaptive system provides optimal mitigation of impact excitation, but its response in the case of harmonic loading is not always optimal and requires further improvement.

1. Introduction

Mitigation of impact loads and damping of vibrations remain challenging engineering problems. In case of impact excitation the main challenge is short period of the process (several to tens of milliseconds). In turn, in the case of harmonic excitation the main challenge is continuous supply of the external energy to the system and the requirement of its repetitive operation. Nowadays, the recent fast progress in technologies of sensors and actuators enables replacing classical passive systems by more efficient semi-active or active devices. It can be observed that the research efforts are dedicated separately to mitigation of impact excitation and damping of vibrations. The most important lines of research in these disciplines are briefly discussed below.

The important research area in the field of semi-active impact mitigation is the so-called Adaptive Impact Absorption (AIA) [1, 2]. The concept assumes that optimal shock absorbing system is composed of embedded system of sensors, the hardware controller and adaptive elements (so called structural fuses), which control the process of energy dissipation in semi-active way. The system constructed according to the principle of AIA utilizes data measured by sensors to identify the actual impact loading and performs adaptation providing optimal mitigation of impact. Typically, the objective of adaptation is to dissipate the entire impact energy with the lowest value of deceleration of the impacting object or the lowest overload of the impacted structure. The most popular AIA problems concern design of adaptive fluid based absorbers. The research in this field was focused on hydraulic [3, 4, 5, 6] or pneumatic [7, 8, 9] dampers equipped with fast controllable valves, as well as dampers based on magneto-rheological [10, 11, 12, 13] or electro-rheological fluids [14]. Nevertheless, other innovative types of absorbers e.g. based on inertial effects were also analyzed [15]. The motivation for development of these systems was a number of practical applications such as mitigation of frontal collisions of cars, high performance landing gears or seismic isolation of structures.

The research concerning semi-active vibration damping is nowadays very extensive and relates to various types of problems and applications. In general, the problems of vibration damping can be divided into two separate groups: damping of free vibrations and mitigation of forced vibrations. In the former case the objective is to dissipate the energy introduced to the structure and to decrease of its oscillatory motion possibly fast. In turn, in the latter case the objective is to dissipate submitted energy and to obtain desired response of the structure. In both groups of problems various types of dissipaters and control techniques are applied. Among the most innovative solutions we can distinguish application of tuned mass damper with inerter of changeable inertance [16], controllable structural nodes with an on/off ability to transmit moments [17] and pneumatically controlled granular structure [18], to mention just a few. The practical applications of the vibration techniques encompass damping of flexible space structures [19, 20], vehicle-track/span systems [21, 22], design of suspension and seismic isolation systems [23, 24], damping of torsional vibrations in electro-mechanical systems [25, 26].

The review of the above mentioned systems for semi-active impact mitigation and vibration damping reveals that most of these systems require the knowledge about applied excitation and they do not operate optimally when variations of system parameters or harsh disturbances occur. Moreover, there are no universal control strategies which can be applied both the problem of impact mitigation and vibration damping. This paper attempts to fill this gap in the subject literature by proposing application of self-adaptive system to both mitigation of impact and damping of vibrations. The concept of such system was introduced by authors in [27] and developed in [28]. Herein its application will be extended to the problem of mitigation of forced vibrations, which was not analyzed before.

2. Basic concept of self-adaptive absorber

The proposed absorber serving for impact mitigation and vibration damping is composed of rigid compartment and two fluid chambers, Fig. 1. The chambers are separated by the piston equipped with valve, which enables control the actual mass flow rate of the fluid in order to adjust the actual value of reaction force generated by the absorber. In general, the chambers can be filled with either hydraulic or pneumatic fluid, however further discussion will be illustrated on the example of pneumatic absorber. The mathematical model of the absorber is described in previous paper of the authors [28]. The absorber will be subjected to two types of dynamic excitation:

- impact loading modeled by mass of the impacting object and its initial velocity (Fig. 1a),
- harmonic excitation modeled by time-dependent harmonic force (Fig. 1b).

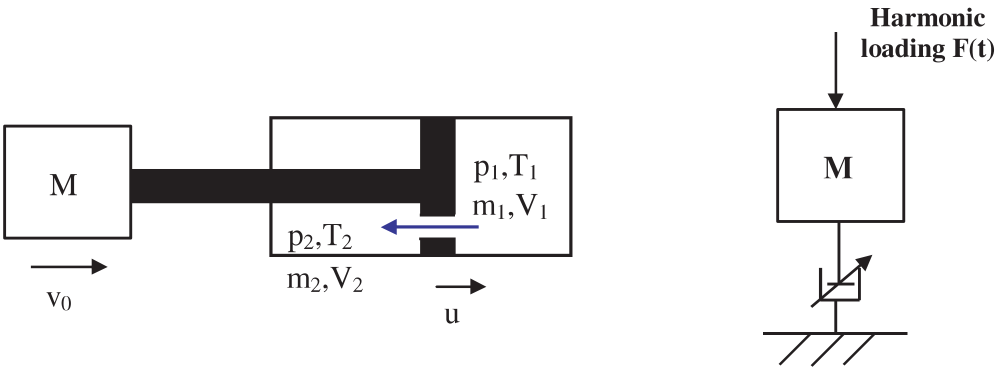


Figure 1: The considered problems: a) adaptive pneumatic absorber subjected to impact excitation, b) the same absorber (shown schematically) under harmonic loading

In the first case the objective of the adaptive absorber is to dissipate initial kinetic energy of the impacting object and decrease its velocity to zero with minimal level of deceleration. In the second case the considered object does not have initial kinetic energy, however the energy is continuously submitted to the system by the action of the external harmonic force. As a result, at the beginning of the process the considered object is accelerated by external force and its velocity increases. Therefore, the objective of the adaptive absorber is to counteract the external force and to stop the object with minimal level of generated reaction force and minimal level of corresponding deceleration. In addition, the proposed self-adaptive absorber has to fulfill three following requirements:

- automatic, online adjustment to actual dynamic loading with neither prior knowledge of the external load nor application of load identification system,
- robust operation in case of disturbances such as occurrence of additional damping or friction forces generated by the absorber,
- robust operation in case of unexpected change of system parameters such as sudden leakage of the fluid from one of the chambers.

In order to fulfill the above stated strict requirements three innovative elements of the control strategy are introduced. At first, the Automatic Path Finding (APF) algorithm is applied to determine optimal system path during the initial stage of operation by using complete information about actual system kinematics and the condition of kinematic optimality for the remaining part of the process. Secondly, the dedicated Hybrid Path Tracking (HPT) algorithm, which utilizes both continuous and bang-bang control, is used for robust tracking of determined system path. The former control law is based on mathematical model of the absorber and it is applied during typical operational conditions, whereas the latter one is used during short periods of system disturbance. Thirdly, the Automatic Path Update (APU) algorithm based on full kinematic feedback is triggered after the occurrence of harsh disturbances or changes of system parameters in order to properly modify previously determined system path. Each of these processes is performed in fully automatic way during operation of the self-adaptive system.

In order to make operation of the system easily understandable we will precisely describe each algorithm included in the proposed control strategy together with corresponding fundamental mathematical formulae.

Automatic Path Finding (APF) – the algorithm performed during first stage of impact absorption (pure energy accumulation – valve closed) until actual value of piston deceleration allows to decrease velocity of the impacting object to zero within remaining stroke of the absorber. The APF is terminated at the moment t_x when the following kinematic condition is satisfied:

$$|a(t_x)| \geq |a_{opt}| \tag{1}$$

The optimal piston deceleration $|a_{opt}|$ ensures dissipation of entire impact energy within absorber stoke *d* and can be described by the formula:

$$|a_{opt}| = \frac{v(t_x)^2}{2(d-u(t_x))} \tag{2}$$

The finite frequency of state measurements and data processing causes that deceleration of the piston at time instant t_x usually differs from the value $|a_{opt}|$. In such situation we cannot obtain exact realization of the optimal system path and thus we introduce the tolerance interval defining acceptable range of deceleration of the impacting object.

Hybrid Path Tracking (HPT) – the algorithm performed after termination of APF, aimed at possibly best following of the optimal system path ensuring dissipation of entire impact energy within available absorber stroke. The control law applied by HPT is determined for time period $(t_i, t_{i+1}\rangle$ and takes the form:

$$A_v^i(t) = \begin{cases} A_v^{min} & if \ \ |a(t_i)| < |a_{opt}| \\ A_v^{max} & if \ \ |a(t_i)| > |a_{opt}^+| \\ A_v^{opt}(t) & if \ \ |a_{opt}| \leq |a(t_i)| \leq |a_{opt}^+| \end{cases} \tag{3}$$

The bang-bang control is performed when absorber operates outside the tolerance interval and it is aimed at the fastest possible return of the system to the optimal path. The upper limit of deceleration tolerance interval is designated by $|a_{opt}^+|$ and it corresponds to assumed maximum shortening of the admissible stroke of the absorber Δu_{tol}:

$$|a_{opt}^+| = \frac{v(t_x)^2}{2(d-\Delta u_{tol}-u(t_x))} \tag{4a}$$

Alternatively, the value $|a_{opt}^{+}|$ can be defined with the use of surplus of acceleration Δa_{tol} or the surplus of velocity Δv_{tol}:

$$|a_{opt}^{+}| = \left(\frac{v(t_x)^2}{2(d-u(t_x))} + \Delta a_{tol}\right), \quad |a_{opt}^{+}| = \left(\frac{(v(t_x)+\Delta v_{tol})^2}{2(d-u(t_x))}\right) \tag{4b,c}$$

The continuous change of the valve opening providing generation of constant reaction force can be calculated by using the model of thermodynamic and flow processes taking place in the absorber, see [27]:

$$A_v^{opt}(t) = f(u_x, v_x, a_x, p_{1x}, p_{2x}, t) \tag{5}$$

Automatic Path Update (APU) – the algorithm ensuring successful operation of the impact absorbing system in case of harsh disturbances, changes of system parameters or need of system re-adaption (e.g. series of unknown impact excitations). The APU algorithm sequentially updates the optimal value of system deceleration $|a_{opt}|$ determined with the use of APF algorithm. It operates after each control period and calculates the actual optimal value of deceleration at time instant t_i by using the formula:

$$|a_{opt}| = \frac{v(t_i)^2}{2(d-u(t_i))} \tag{6}$$

Let us note that proper operation of the self-adaptive system requires also sequential update of the upper limit of deceleration tolerance interval $|a_{opt}^{+}|$.

Let us note that in the proposed control method the tolerance interval for deceleration of the impacting object $\langle a_{opt}, a_{opt}^{+}\rangle$ is asymmetric, which directly results from the assumption that admissible displacement of the piston is in the range $\langle d - \Delta u^{tol}, d\rangle$. Such assumption ensures successful operation of the absorber – the piston will never hit absorber's bottom if the tolerance value is appropriately selected for the available frequency of measurements and data processing. In order to reduce effort and possible mistakes appearing during system tuning we can apply the automatic adaptation of the tolerance levels at time instant t_x which terminates operation of the APF algorithm; it will be discussed in further section.

As mentioned above, the definition of upper limit of deceleration tolerance interval $|a_{opt}^{+}|$ can be based on maximum shortening of the absorber stroke Δu_{tol} (displacement tolerance), velocity surplus $\Delta \mathrm{v}_{\mathrm{tol}}$ or acceleration surplus Δa_{tol}. Selection of different definitions leads to differences in system behavior. In particular, assumption of acceptable level of absorber stroke shortening (Eq. 4a) results in increasing in time upper limit of deceleration tolerance interval. In such case, when no disturbances are present, the control system will not change the level of generated reaction force. In contrast, application of definition based on acceleration surplus (Eq. 4a) and velocity surplus (Eq. 4b) may result in often automatic corrections of the preliminary determined level of generated force. Such behavior is beneficial when extremely fast measurement and data processing system can be implemented. Nevertheless, in case of relatively low frequency of system operation, it is better to define the deceleration tolerance interval based on maximum shortening of the absorber's stroke. As a result, temporary system destabilization will be avoided and system efficiency maintained.

3. Control system architecture

According to introduced definitions of algorithms discussed in previous section they can be considered as processes appearing during system operation. Graphical representation of the processes conducted during unknown impact mitigation is shown in Fig. 2. The APF is performed until deceleration of the piston and impacting object is insufficient and maintaining its constant value would result in hitting the absorber bottom (the case of impact energy not entirely absorbed). When kinematic condition (Eq. 2) is met, "Scenario 2" is valid and the APF process terminates. In further operation the HPT has to be performed to ensure successful operation of the system. Realization of the actually optimal control path with acceptable tolerance indicates that absorber still operates according to "Scenario 2" so that continuous control of valve opening is applied. If unexpected disturbances occur and cause the reaction force jumps out the tolerance interval, the control system has to react immediately. Otherwise, too low value of reaction force ("Scenario 1") will result in the piston hitting the absorber bottom or too high value of reaction force

(“Scenario 3”) will cause that deceleration will not be optimally minimized and small rebound before the end of the absorber stroke will occur. In order to reduce influence of disturbances and to maintain high efficiency of the absorber the HPT will close the valve for “Scenario 1” or fully open it for “Scenario 3”. The fact that system deceleration remains outside the optimal path for certain period of time causes the automatic path recalculation is required and it is performed by the APU algorithm.

Similarly, if next impact excitation occurs before the end of the first impact absorption process, the system will determine new actually optimal path by applying APU and will track this path with the use of HPT. It should be highlighted that determination of actually optimal path of the system is based on the kinematic condition describing optimality of the remaining part of the impact absorption process. This indicates that if there are no changes of impact conditions, the system will follow (with acceptable tolerance) the globally optimal solution. In turn, when the process includes unpredicted disturbances or several unknown impact excitations, the system will operate in robust, suboptimal way.

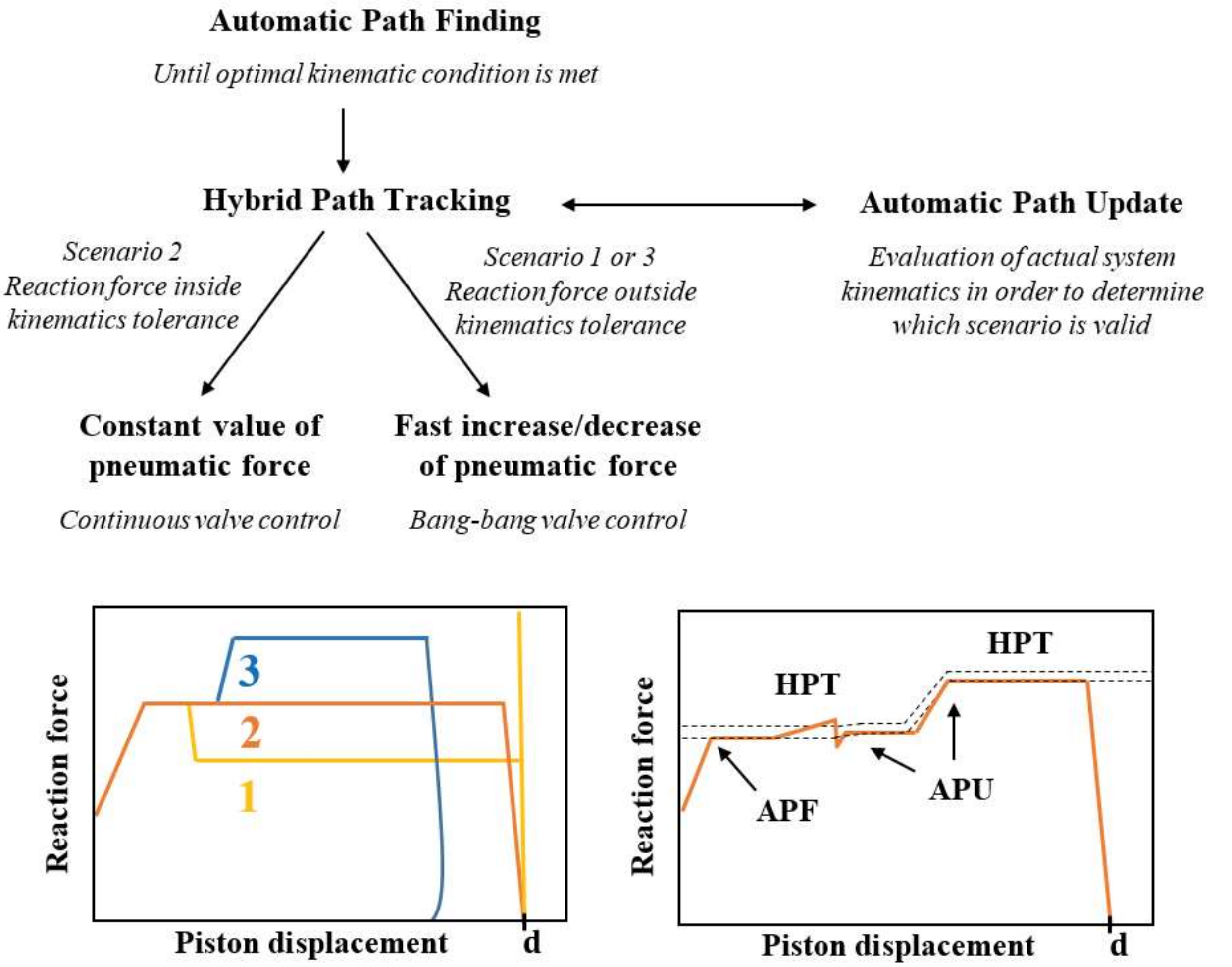


Figure 2: Scheme of self-adaptive system operation

Following the general idea of discussed self-adaptive impact absorbing system, we will introduce a block diagram of the control system ensuring desired system operation (Fig. 3). The considered system is subjected to dynamic excitation. The system response is measured by kinematic sensors (e.g. accelerometers, linear encoders) and pressure sensors (internal pressure in both absorber chambers). Path control system based on actual piston kinematics determines which scenario of system operation is valid at the moment. In case of too low or too high value of piston deceleration the bang-bang control of the valve is applied. Otherwise, the piston deceleration is optimal with acceptable tolerance and the reaction force has to be maintained constant. Therefore, measured values of internal pressure in absorber chambers and measured kinematic quantities are used to determine the optimal continuous valve control by applying mathematical model of the absorber.

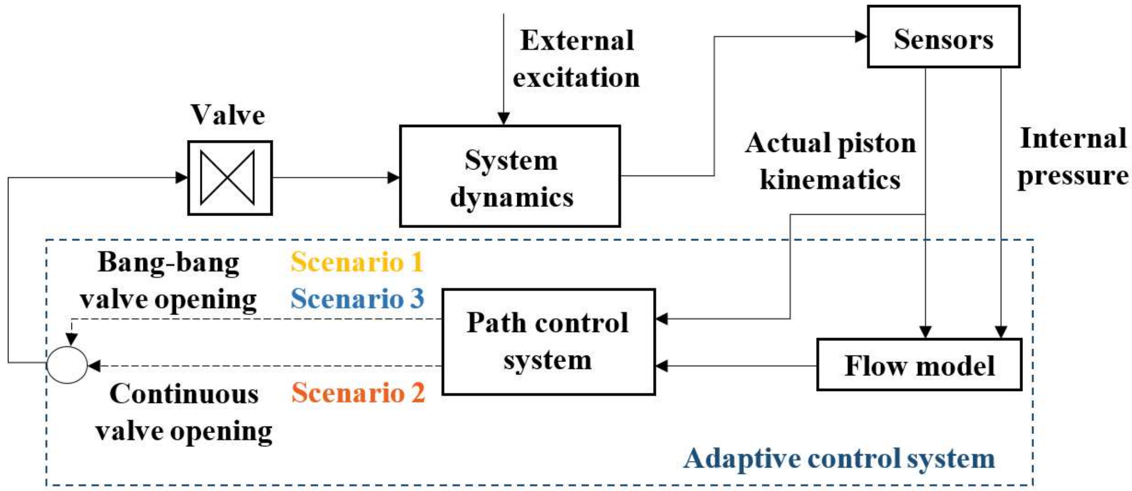


Figure 3: Block diagram of proposed self-adaptive control system

It should be indicated that the proposed control system does not have typical tuning parameters as in e.g. PID or LQR controllers. Nevertheless, two system parameters have a significant influence on the system operation and they have to be considered when control system is designed: i) frequency of kinematics and internal pressure measurements and ii) time of data processing. In further discussion we will use the term Frequency of Control Recalculation (FCR) resulting from speed of measurements gathering and data processing. The finite value of FCR results in small inaccuracy of the system operation and tracking of actually optimal system response with certain delay. In practice, the maximum achievable value of FCR is imposed in advance because of technological limitations. Because of that the tolerance imposed on optimal value of piston deceleration should be adjusted to the frequency of system operation and can be treated as tuning parameter of the control system.

4. System operation under impact excitation

The present section is aimed at investigation on the response of self-adaptive system under impact loading with a special attention to influence of the displacement tolerance determining the interval of acceptable piston decelerations, which can be treated as nearly optimal values. The influence of under- and overestimation of the upper limit of displacement tolerance is shown in Fig. 4

According to the graph in Fig. 4a, which presents response of the system operating with high value of FCR, overestimated value of the tolerance can result in lack of dissipation of the entire impact energy. This effect results from the fact that high frequency system finds the optimal value of absorber reaction force with high precision. In this situation assumption of high tolerance for maximum piston displacement is in contradiction to achieved accuracy of finding the optimal value of pneumatic force. When assumed final displacement of the piston $d - \Delta u_{tol}$ is exceeded, the valve is automatically opened resulting in light impact of piston against absorber bottom. This indicates that for system with high values of FCR the displacement tolerance should be selected at appropriately small level or some adjustment of proposed control method should be made.

If the tolerance is decreased to the proper value, the entire impact energy will be dissipated within assumed available stroke, simultaneously ensuring successful minimization of absorber reaction force. On the other hand, if displacement tolerance is decreased excessively, some overshoots of reaction force may appear (Fig. 4b), despite this the entire absorber stroke is finally utilized with better precision. In contrast, Fig. 4c presents behavior of the system operating with low values of FCR. It can be noticed that increase of displacement tolerance reduces force overshoot but also results in shortening of the utilized stroke. It can be concluded that value of displacement tolerance for optimal deceleration interval should be selected carefully. Depending on the operational requirements, the system without overshoots in the reaction force or system with higher stroke utilization can be a preferred option.

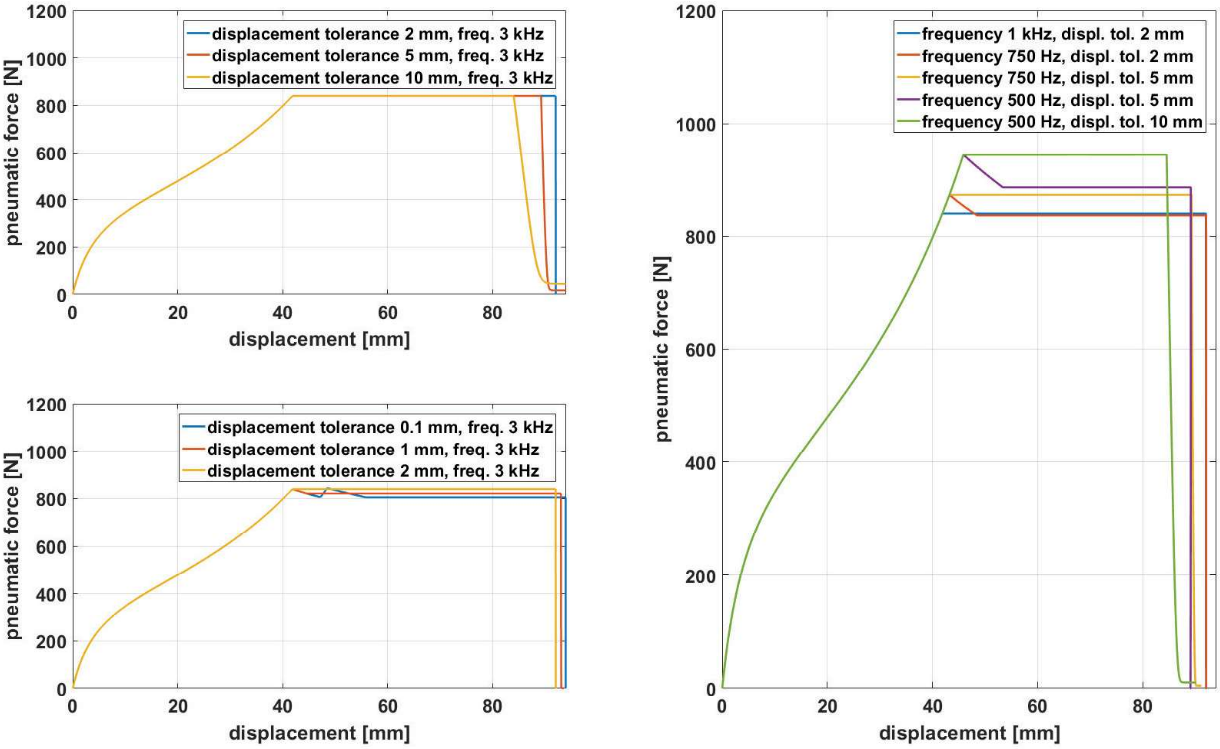


Figure 4: Influence of control recalculation frequency and optimality condition tolerance on the system response under single impact excitation

Automatic adjustment of displacement tolerance

Presented above analyses of the influence of displacement tolerance revealed difficulties in appropriate selection of tolerance for given value of FCR. In order to avoid this problem a simple automatic adjustment of the tolerance value can be introduced. At the beginning of impact absorption process we assume a certain value of the piston displacement tolerance Δu_{tol}, as it had been done before. When the reaction force reaches the value within the range corresponding to the optimal deceleration interval $\langle |a_{opt}|, |a_{opt}^{+}| \rangle$ and thus HPT is activated, the displacement tolerance is recalculated to ensure that entire impact energy will be dissipated within absorber stroke according to the equation:

$$\Delta u_{tol}^{adapted} = d - u(t_i) - \frac{v(t_i)^2}{2|a(t_i)|} \tag{7}$$

Thanks to tolerance adjustment made when the HPT is activated, the impact energy will be entirely dissipated for both low and high values of FCR independently on the initially assumed value of displacement tolerance. In Fig. 5 the response of systems operating with different values of FCR and initial displacement tolerance $\Delta u_{tol} = 5\ mm$ are shown. In case of low level of FCR the system may need to correct the path before initially assumed value of displacement tolerance and corresponding deceleration interval are met. Nevertheless, when the optimal deceleration interval is reached the displacement tolerance is recalculated to the value of actual deceleration $|a(t_i)|$ being in the range $\langle |a_{opt}|, |a_{opt}^{+}| \rangle$. As a result, maintaining the pneumatic force on the corresponding constant level results in stopping the impacting object on the distance $(d - \Delta u_{tol}^{adapted})$ without any rebound and without hitting the end of absorber stroke. Proposed adjustment of the control principle provides also successful operation of the system working with high value of FCR and low value of displacement tolerance. The proposed control system with automatic adjustment of displacement tolerance will always ensure dissipation of entire impact energy within absorbers stroke.

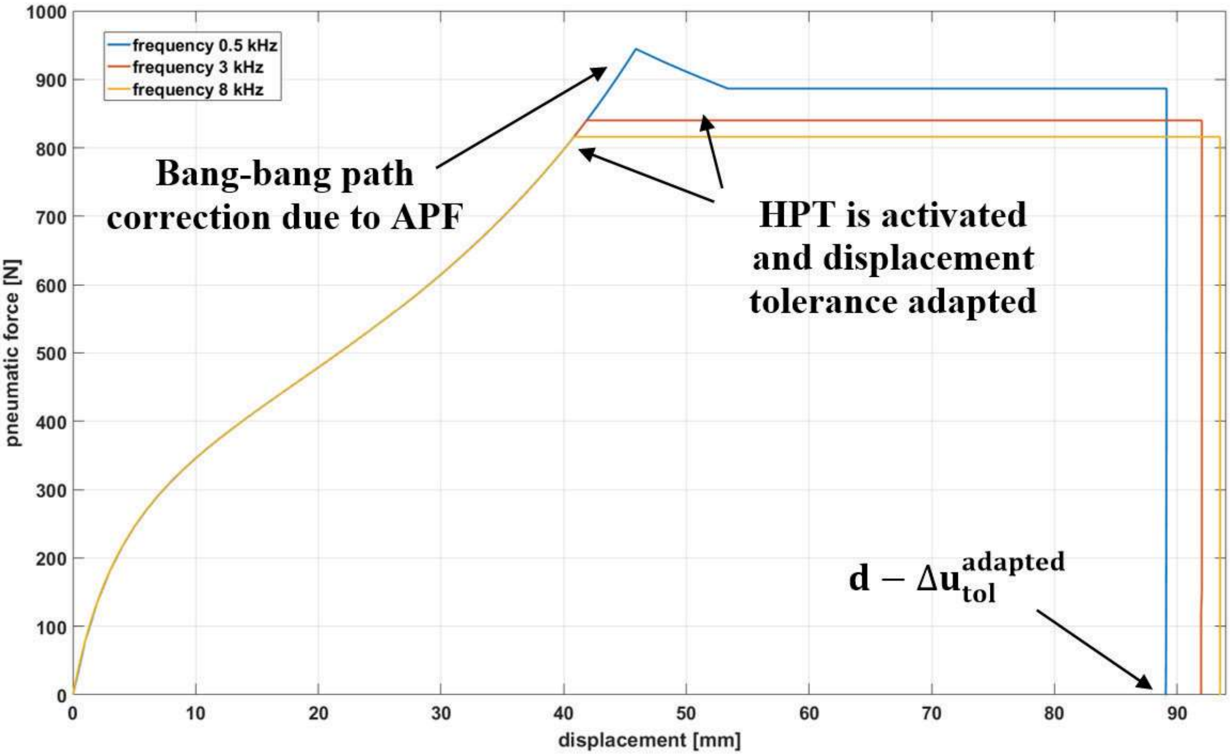
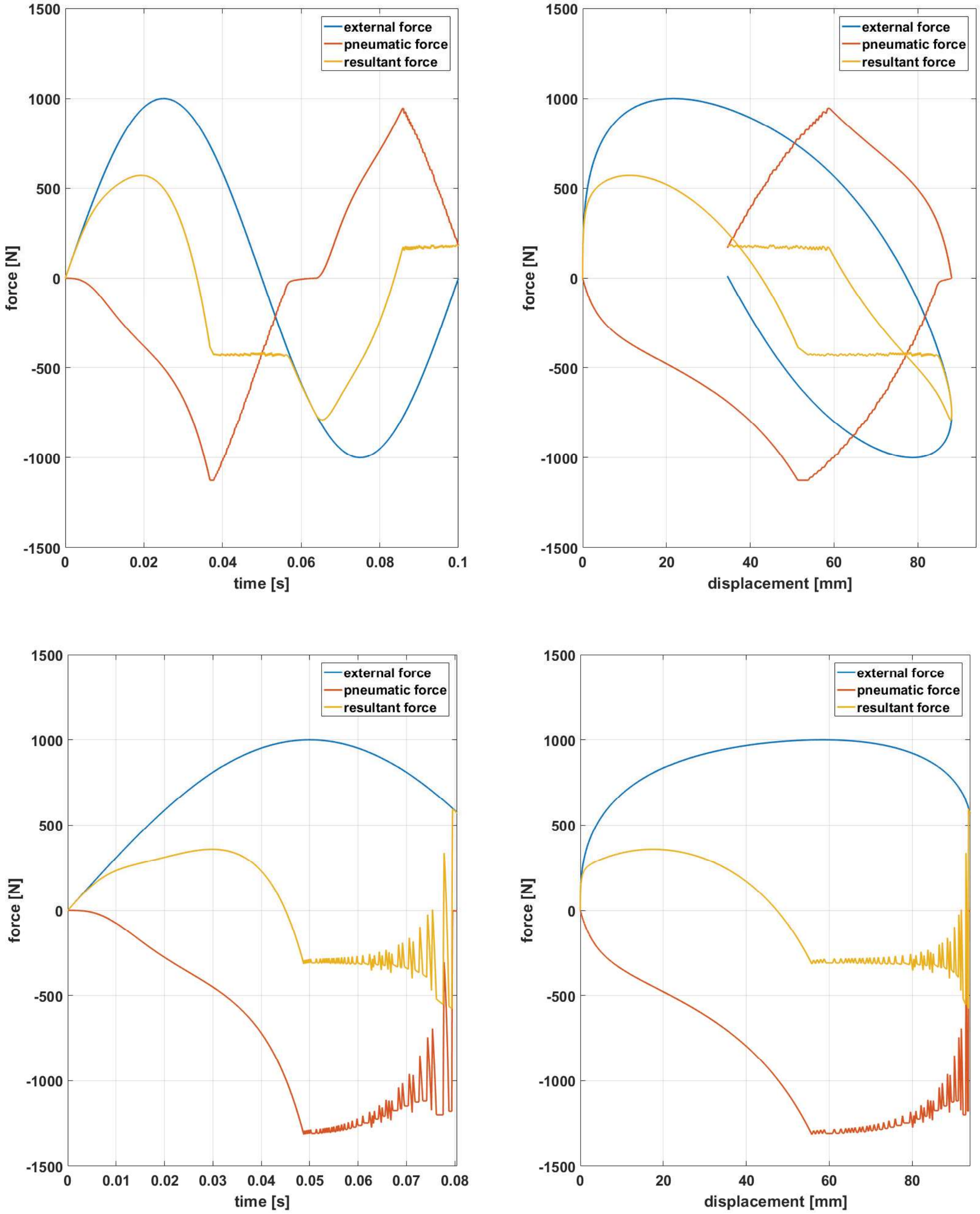


Figure 5: Operation of the system with automatic adaptation of displacement tolerance

5. System operation under harmonic loading

In this section the proposed self-adaptive control system is applied for mitigation of the harmonic excitation. Three different frequencies of excitation are investigated, whereas the control is always performed with frequency of 5 kHz. The conducted numerical simulations indicate that response of self-adaptive system in three considered cases is substantially different. In case of the lowest frequency of harmonic excitation equal to 5Hz the impacting object is stopped at the end of cylinder stroke before the cycle of harmonic excitation is accomplished. This indicates unfavorable situation when energy is still submitted to the system but there is no remaining stroke left to dissipate this energy (Fig. 6a). In contrast, in case of the largest frequency of impact excitation equal to 10 Hz the excitation drops below zero before the impacting object is stopped. This indicates non-optimal situation in which deceleration of the object suddenly rises and the entire stroke of the absorber is not utilized (Fig. 6b). Therefore, we can expect that there exists the frequency of excitation for which the object is stopped exactly at time instant when single excitation cycle is accomplished. Indeed, the numerical simulation shows that such situation occurs for the excitation frequency of 6.7 Hz. In this case the entire impact energy is dissipated and the object is stopped almost at the end of cylinder's stroke.

We conclude that proposed self-adaptive system does not provide optimal mitigation of the excitation of various frequencies. In order to improve the operation of the system the level of deceleration, which is maintained constant should take into account compatibility with time when single cycle of the harmonic excitation finishes. Development and testing of such system is the objective of current research.


external force
pneumatic force
resultant force
force [N]
time [s]
displacement [mm]
1500
1000
500
0
-500
-1000
-1500
0 0.02 0.04 0.06 0.08 0.1
0 20 40 60 80
0 0.01 0.02 0.03 0.04 0.05 0.06 0.07 0.08

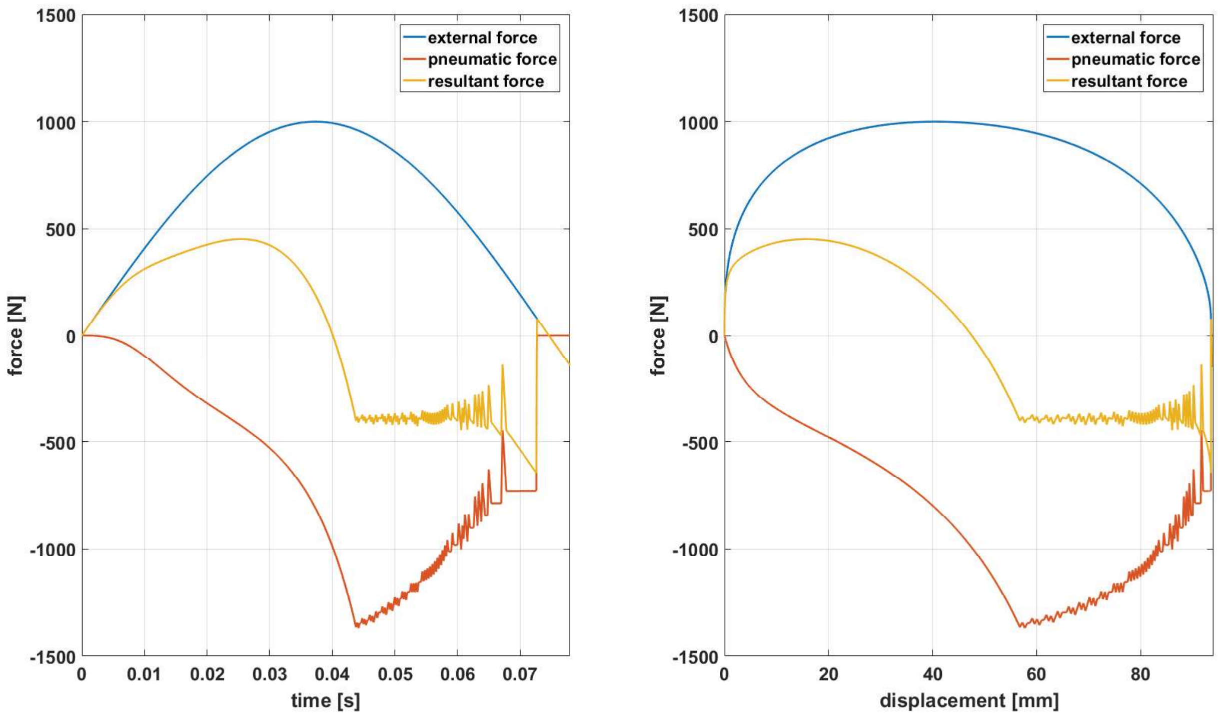


Figure 6: Response of self-adaptive system to harmonic excitation of various frequencies: a) 5 Hz (unacceptable response), b) 10 Hz (non-optimal response), c) 6.7 Hz (optimal response)

6. Conclusions

Performance of the innovative self-adaptive control system was thoroughly tested by using double-chamber pneumatic absorber subjected to impact excitation and harmonic loading. It was proved that the control system equipped based on APF, HPT and APU has the ability of automatic determination of the optimal system path, its robust realization and appropriate on-line modification as a response to disturbance. In the case of impact loading the requirements of energy dissipation and force minimization are always successfully fulfilled so it can be concluded that proposed control system is fully self-adaptive and it is superior to standard passive and typical adaptive absorbers used for impact and vibration mitigation. However, in case of harmonic excitation the current self-adaptive operation does not lead to optimal energy dissipation and some improvements of the control strategy have to be introduced.

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