KU Leuven
Department of Mechanical Engineering
LMSD (Mecha(tro)nic System Dynamics)
Celestijnenlaan 300 - box 2420
B-3001 Heverlee, Belgium

Proceedings of

ISMA2022

International Conference on

Noise and Vibration Engineering

USD2022

International Conference on

Uncertainty in Structural Dynamics

12 to 14 September, 2022
Editors: W. Desmet, B. Pluymers, D. Moens, S. Neeckx.

Switching stiffness approach on a semi-active pneumatic actuator dedicated to vibration isolation

G. Mikułowski
Polish Academy of Sciences, Institute of Fundamental Technological Research,
Department of Intelligent Technologies,
ul. Pawińskiego 5B, 02-106, Warsaw, Poland

Abstract

In this study a novel technique of switchable stiffness dedicated to structural vibration mitigation is presented. The approach is based on a semi-active pneumatic device, which enables the stiffness switching operation via controlled thermodynamic process. The process allows for adaptation of the system in response to the current vibration excitation amplitude and frequency. The proposed system is analysed under passive and semi-active modes of operation in an experimental survey. The study consists of the energy dissipation process analysis and a verification of effectiveness of the concept under varying operational conditions. As a result, experimental transfer functions of a one degree of freedom system revealed minimization of the magnitudes below unity for the complete resonant range. The presented concept introduces an innovative approach to the switching stiffness techniques dedicated to vibration mitigation.

1 Introduction

Pneumatic systems for vibration isolation are widely utilised solution, which can be considered as a classical technical design task. The modelling and analysis of dual chamber pneumatic isolators, which are the most popular configuration, is a commonly studied case [1]. On the other hand, there exist a significant interest in adopting modern controllable techniques for improvement of the pneumatic system's performance. The most promising systems are based on the semi-active operation principle. The semi-active solutions in general are proved to be more flexible and be more effective than passive systems. Moreover, they are more stable and require less energy to be operated in comparison to active systems. Thus, the semi-active techniques gather increased interest among scientists in the field. Recent publications reveal a series of studies that are dedicated to new concepts related to the field.

Researchers from the University of Castille-La Manche (UCLM) have proposed a method and a vibration control algorithm [2], which adopts the resonant frequencies of pneumatic systems. The system consists of a dual chamber isolator with inter connecting tubes and supplementary gas vessels. The control aim is to modify a dynamic stiffness in order to avoid the system's eigenfrequency. The objective is obtained through introduction of the following elements: switching between two connector tubes of various diameters, altering the length between the bellows and changing pressure in additional gas vessel with rigid walls. The presented control algorithm is based on the system's transmissibility observation and aims to switch between two connection tubes when a transition frequency is determined.

Pu et al. [3] have proposed a concept of a dual-chamber pneumatic vibration isolation system equipped with an interconnecting adjustable orifice. The proposed configuration aims at tuning stiffness and damping of the isolator by adjusting the orifice diameter that is kept constant in the time of operation. The procedure allowed for finding an optimal setting that minimizes transmissibility of the system. The concept is dedicated for a stationary vibration isolation of highly sensitive equipment, therefore the demanded displacements are very small in relation to the diameter of the device. Due to small vibration amplitudes it has been feasible for the authors to use an adiabatic model of constant parameters, which involves no energy exchange with the surroundings. Also, it allowed for neglecting energy losses on the flow through the orifice. In contrast, addressing systems with a capability of operation with larger range of strokes is more demanding in modelling and in carrying out. An example of such a system is the one discussed in this study.

Recently, an increased interest has been given to vibration isolation system based on quazi-zero-stiffness concept [4], requiring the use of negative stiffness elements. Authors show that introduction of this approach to isolation systems improves their performance in the range of low frequencies. Dahn et al. [5] have proposed an active pneumatic system obtained by vertical elements combined with a passive negative stiffness structure consisting of two horizontal spring elements. Introducing a system, which carries out the negative stiffness principle in a semi-active adaptable manner may improve the performance even further. A solution with a semi-active approach to the negative stiffness is presented in this work.

Palomares et al. [6] have proposed a vibration isolator with a negative stiffness system that is made of pneumatic double-acting actuators. An important feature of the system is that each isolation unit consists of three actuators: one vertical and two horizontal. The two horizontal ones are intended to carry out a negative stiffness operation in a similar way to passive quazi-zero-stiffness systems. The system is intended to be adaptive to variations in the operational conditions by changing pressure in the volumes of the actuators. The controlled parameter is dynamic stiffness of the system. The main disadvantage of the proposed solution is its limited compactness.

The above mentioned techniques reveal a promising effectiveness of the switching stiffness and negative stiffness techniques in pneumatic systems. However, the presented solutions have numerous disadvantages that make them difficult in implementations. Firstly, they require an external source of power to adjust instantly the pneumatic pressure. Secondly, the proposed solutions are large complicated systems with the lack of compactness, which is very important feature in suspension design. Thirdly, most of them is designed in order to respond to a narrowed range of amplitudes, which limits significantly their versatility in applications. Therefore, in this work an adaptive technique is proposed which, overcomes the mentioned disadvantages.

In this work a pneumatic semi-active system utilising a switching stiffness technique for vibration control is presented [7]. It has a possibility of dissipating the accumulated mechanical energy via an irreversible thermodynamic process. It means that the stored energy is instantaneously changed into heat. The process ensures a situation of no energy transfer from the spring elements back to the mechanical system, allowing for enhanced performance of the vibration control procedure. The proposed system involves large displacements of the piston and intensive controllable flow of gas between cylindrical volumes, which both vary in time. This situation enforces an implementation of an advanced method of mathematical modelling. The modelling needs to take into account thermal energy exchange between the volumes and the surroundings, energy transfer by enthalpy and energy losses on the throttled flow [7]. In this work experimental results related to the concept are presented. This is a novel approach in the switching techniques dedicated to vibration mitigation, which enables implementations of more effective semi-active pneumatic suspension systems.

2 Vibration isolation concepts

The vibration isolation objective in this study is considered as minimising power of vibration, which results in lowering the corresponding displacements. The equation of mechanical power balance in a 1DoF vibrating system can be noted as:

$$\frac{dK}{dt} = -\frac{dP}{dt} - c\dot{z}^2, \tag{1}$$

where K is kinetic energy, P is potential energy and $\dot{z}$ is velocity of a mass [7]. Both of the energies are positive by definition:

$$K = \frac{1}{2}m\dot{z}^2 \geqslant 0, \tag{2}$$

$$P = \frac{1}{2}kz^2 \geqslant 0. \tag{3}$$

Lets consider a process of the kinetic energy decrease in the system. The contribution to the kinetic energy decrease comes from two sources. Firstly, damping, which contributes continuously by definition $c\dot{z}^2 > 0$.

Secondly, the further decrease of the kinetic energy can be obtained by an increase of the potential energy:

$$\frac{dP}{dt} \geqslant 0, \tag{4}$$

which is a phenomenon that takes place during compression phase of vibration due to the fact that energy P is accumulated in the spring. However, in a passive system it is not possible to achieve an exclusive increase of the potential energy as it is periodically accumulated and released in dependence on the phase of the vibration movement. Therefore, the change of the potential energy in the systems can be denoted as following:

$$\frac{dP}{dt} = kz\dot{z} > 0$$

or

$$\frac{dP}{dt} = kz\dot{z} < 0.$$

In the passive systems the potential energy increase corresponds to the mass deceleration phases, whereas the energy decrease occurs in the acceleration phases. Using a semi-active device with a pneumatic adaptive absorber equipped with a switchable stiffness functionality, allows for utilisation of an appropriate switching strategy that leads to vibration reduction. The aim of the strategy is to arrange a situation, in which the potential energy is increased most of the time. Therefore, the kinetic energy of the system is reduced at the same time.

2.1 Semi-active switching stiffness concept

The presented semi-active swithing stiffness is based on a concept of a controllable transferring gas between the volumes of a pneumatic piston-cylinder device. The gas is transferred via a dedicated precise valve positioned in the piston. The process has an important objective as it equalises pressure between the volumes and therefore releases the accumulated potential energy. After the gas transfer the system is reset for a subsequent stroke of the operation. Therefore, the pneumatic system allows for carrying out a stiffness of a negative sign.

Using the semi-active device with the pneumatic adaptive absorber characterised by the switchable stiffness, allows for utilisation of an appropriate switching strategy that leads to vibration reduction. The aim of the strategy is to arrange a situation, in which the potential energy is increased most of the time and the mass is decelerated.

2.2 Energy dissipation principle

The fundamental feature of the concept is that at the end of each deceleration phase, the accumulated potential energy is dissipated in a controlled irreversible therodynamic process. The energy dissipation is carried out by a double stage process. The potential energy is accumulated in a thermodynamic process of gas compression and then it is dissipated by transferring the gas between two volumes of the cylinder. The process has an important advantage in comparison to the switching stiffness concepts known from the literature, because during the gas transfer, the pressure difference between both sides of the piston drops. Eventually, the mechanical reaction of the system drops and the potential energy is dissipated instead of being accumulated.

In the concept, the decelerating reaction of the system can be obtained for the whole period of vibration, both in the accelerating and decelerating phases of the mass movement. As a result, the system is able to decelerate mass in presence of the both following conditions:

$$\frac{dP}{dt} > 0 \wedge \frac{dP}{dt} < 0. \tag{5}$$

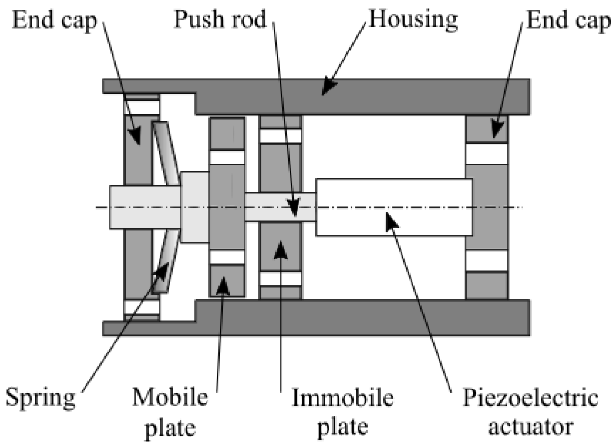

Figure 1: High performance valve - scheme.

2.3 Features of the system

The concept consists of two essential features, which are: double gas volumes and a controllable valve in the piston. The first involves utilisation of a double volume configuration, which improves the mechanical response of the conceptual device [7]. The second employs introduction of a fast operable valve that allows for a controllable transferring gas between the volumes [8].

A schematic view of the valve is depicted in Fig. 1. It is assumed that the valve allows for a rapid modification of the pressure difference between the volumes. This enables an on-line control of the reaction force generated by the device. In this meaning it can be considered semi-active, since the low energy used for the control process results in modification of the total reaction force generated by the absorber. In order to carry out the assumed operation of the piston-cylinder device, the valve has to fulfil two requirements. The first one is providing an adequate mass flow rate, which corresponds to the frequency of the system vibration – analysed in detail in [8]. The second one is a short time delay during operation, which is required to be operable on the vibration control. A specific shape of the orifices is a multi-path complex geometry coming from the utilized plates' design. The shape originates from their purpose, which is intended to be operated by a piezoelectric actuator.

3 Control strategy

The control strategy dedicated for the vibration isolation case is based on the principle of the switching stiffness. The main objective of the approach is to decrease the transmissibility of the system by introducing the permanent deceleration situation, for both increasing and decreasing the potential energy as stated in Eq. (5). The strategy of the stiffness modification aims at a synchronous energy dissipation operation. The maximum point of the potential energy accumulation takes place at the time instant of the maximal displacement. At the same time instant velocity is equal zero. Therefore, the heuristic control algorithm is based on monitoring the velocity signal and recognises the zero crossing events.

The control signal C_{ctrl} is equal 0 for the state of closed valve and it is equal 1 for the state of an opened valve, thus the control algorithm can be noted as:

$$C_{ctrl} = \begin{cases} = 0 & when \quad \dot{z} < u_{c1}, \\ = 1 & when \quad u_{c1} < \dot{z} < u_{c2}, \\ = 0 & when \quad \dot{z} > u_{c2}, \end{cases} \tag{6}$$

where u_{c1} and u_{c2} are upper and lower offset values for the velocity signal.

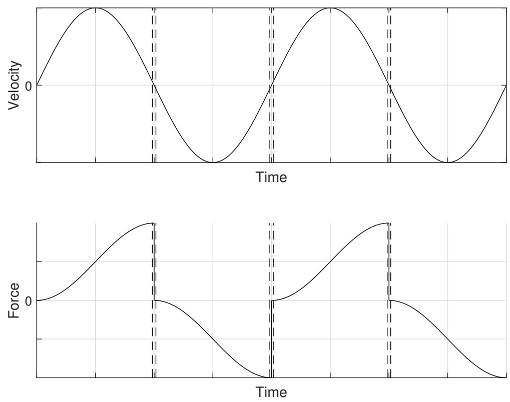

Figure 2: An illustration of the semi-active system operation under a harmonic excitation. Switching instants are depicted with dash lines

To illustrate the operation of the switchable device it is studied on an example of a harmonic sinusoidal excitation, depicted in Fig. 2. The velocity plots reflect the excitation history and the force plot refers to the reaction generated under the control strategy. Moreover, time instants of the valve operation are represented with dash lines. The presented control of the device assumes short lasting openings of the valve at the time instants when the velocity is close to zero. It means that the valve is open at the moment when the potential energy accumulated by the gas is maximal. The process of the gas free expansion between the volumes occurs, and results in an irreversible dissipation of the accumulated energy, which takes place on a thermodynamic principle. This results in increasing the effective damping effect produced by the device.

There are two essential components of the control strategy concept. The first is the fact that after each opening of the valve the system resets for another stroke of the energy accumulation and dissipation. Hence, it performs dissipation two times per period (Fig. 2). Therefore, it is free from disadvantages of switching stiffness concepts presented in the literature. In this sense its dissipation effectiveness is increased. The second important feature is the fact that the system is fully adaptive, as it follows the current velocity of the system. With the velocity change in a damped mechanical system, the controlled device follows the change and adapts to the system's response. Regarding the vibration mitigation, the absorber can be treated as a semi-active switchable stiffness device with an ability to dissipate the energy in a synchronisation with the mechanical response of the vibratory system.

4 Experimental stand

A laboratory system performing the stiffness switching strategy has been tested on a vertical testing stand with kinematic excitation in order to obtain the elementary mechanical response, as depicted in Fig. 3. The excitation setup is based on a hydraulic actuator (manufacturer MTS Corporation) mounted vertically on a loading frame. The actuator's operational bandwidth is 0.01 – 30 Hz at 100 mm displacement, which is relevant for the investigation as the natural frequency of the considered system is below 10 Hz. The semi-active absorber is mounted on the top of the actuator and from above it is loaded with an inertial mass guided along a guiding frame. Parallel to the absorber, linear springs support the inertial mass.

The setup is equipped with sensors divided into two groups. The first is dedicated to monitoring the state of the absorber and contains: gas pressure sensors, gas temperature sensors, piston position linear encoder. The second group deals with acquiring data from the inertial mass including: a linear encoder for measuring the mass displacement, a load cell to measure the input force, a load cell to measure the mass reaction. The precise definition of the particular equipment parameters is given in Tab. 1.

The pneumatic absorber laboratory model is a 32 mm inner diameter cylinder of 200 mm length with a piston

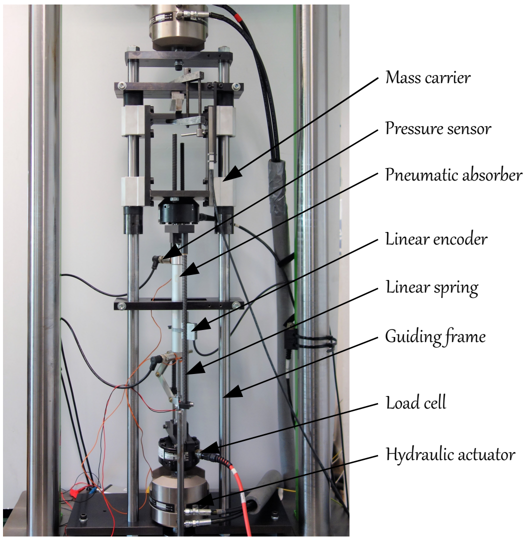

Figure 3: Experimental stand

Table 1: Data on the experimental stand

	$Manufacturer$	$Type$	$Range$
Hydraulic actuator	MTS	244	15 Hz
Pressure sensor	AS	$A3502$	35 bar Abs
Load cell	MTS	661	5 kN
Linear encoder	$Limes$	$B.2$	150 mm
Linear encoder	$Limes$	$B.2$	500 mm

equipped with a piezoelectric valve. The valve provides a possibility of a controllable transfer of the gas between the control volumes in the cylinder. The precise dosing of the gas is ensured via application of a fast responding piezoelectric stack element, which is characterised by a micron range strain resolution [8]. The response delay of the valve opening equals 1.2 ms. An important feature of the setup is that any modifications of the operation conditions are attainable i.e. inertial mass, initial pressure, stiffness of elements.

5 Energy dissipation

The main objective of the investigation is to analyse the capacity for the energy dissipation and vibration mitigation of the semi-active system. Complementarily, results of a numerical modelling are presented.

5.1 Energy and power dissipation

The adopted control strategy ensure the irreversible dissipation of the energy accumulated in the gas, which is preferable from the energy management point of view. At the end of each cycle of the vibration process the reaction force of the absorber is minimised. Therefore, the potential energy is instantaneously dissipated in the way which does not involve any energy injected back to the system. An exemplary energy dissipation and management processes are illustrated in Fig. 4. Energy dissipated in a single cycle of operation can be noted as an integral in the following form:

$$E = \oint F_p(z)\, dz, \tag{7}$$

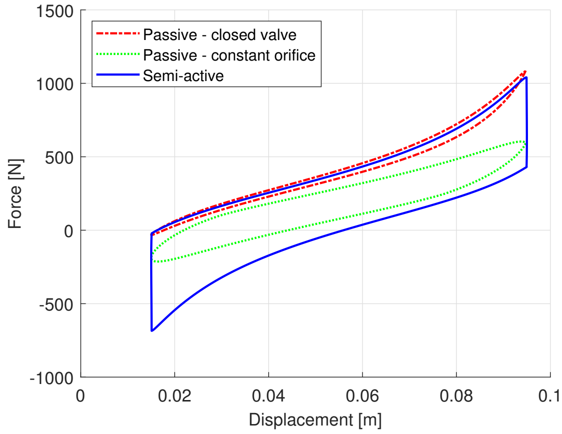

Figure 4: Numerical model response in displacement domain in three modes of operation

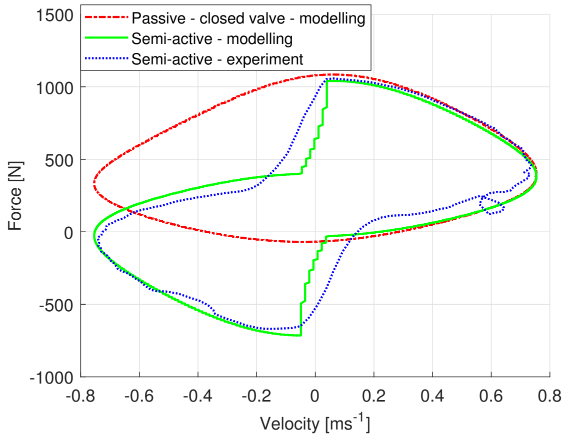

Figure 5: Numerical model versus experimental response in velocity domain

where F_p is reaction of the pneumatic absorber and z is displacement of the piston.

The response of the absorber to a sinusoidal excitation in three modes of operation is depicted in Fig. 4. The first plot describes a passive response with the valve continuously closed. In this mode the absorber reacts like a non-ideal pneumatic spring. The plot is characterised with a narrow hysteresis loop of the reaction force, which indicates that there exists a dissipation process characterised by a small magnitude. The dissipation results from internal friction and gas leakages in the cylinder device. The second plot depicts a reaction of the absorber when there is a constant orifice in the piston. In this case the device can be considered as a passive pneumatic damper. The opening of the valve is carried out to the full throttle, which is characterised in this particular case by 15 g/s of the mass flow rate [8]. The plot indicates that the introduced orifice significantly increases the work done on the dissipation (by 72%). The third plot illustrates a response of the absorber in the semi-active control mode. The dissipated energy has the highest value in this mode, which comes from the enhanced management of the gas transfer between the volumes. The introduced control strategy allowed for increasing the maximum levels of the achieved reactions. Therefore, the dissipated energy is 147% larger in comparison to the passive absorber case. The exact numerical data on the results is provided in Tab. 2 containing calculated (eq. 7) energies dissipated by the device in the three modes of operation.

Due to the fact that the controlled valve openings are programmed to take place twice per vibration period, the efficiency of dissipation is additionally increased in the system. The mentioned control effect is demonstrated on the force-velocity graph of the system response depicted in Fig. 5. The graph illustrates two modes of operation. The first is passive with constant orifice in the valve. The second is the semi-active device response

Table 2: Data on the experimental stand

Mode of operation	Energy dissipated	Percentage
Passive, closed valve	4.33 J	28 %
Passive, constant orifice	15.26 J	100 %
Semi-active	37.64 J	247 %

with the data from the simulation and the experiment. The plots indicate that the control procedure results in an increment of the negative reaction magnitude when the velocity takes the negative values. The depicted sudden changes of the reaction force close to value zero of velocity occur due to the control algorithm operation. The result reveals that the system dissipates the energy for both positive and negative velocity range which is the presumed advantage of the system. Moreover, it demonstrates the effect of the adopted semi-active switching stiffness approach, which increases the dissipation capability of the system. The graph also highlights the accuracy of the simulation technique proposed in this case via the depicted experimentally obtained curve.

As additional remarks, it should be mentioned that the proposed control strategy is characterised by a unique feature of a self adaptation to the frequency and amplitude. As it is controlled in reference to the velocity signal, the response of the absorber is activated when there occurs a movement in the system. The power dissipated by the absorber is also related to the magnitude of displacements.

6 Testing in frequency domain

This stage of the research consisted of experimental works:

- demonstration of the frequency domain response of a 1 degree of freedom system with the semi-active absorber,
- proving feasibility of the concept in laboratory environment
- verification of the adaptivity of the system
- testing a robustness of the system to variation of the operational parameters i.e. excitation frequency, mass mgnitude, internal pressure of the operating gas.

The analyses are conducted experimentally on the prepared experimental stand. For purposes of this work a performance index is adopted that describes displacement transmission in the system and is taken as the measure of effectiveness. The function is defined as:

$$H(f) = \frac{S_{z_1 z_2}(f)}{S_{z_1 z_1}(f)} \tag{8}$$

where $S_{z_1 z_2}(f)$ is Cross Spectral Density of the output to the input and $S_{z_1 z_1}(f)$ is Auto Spectral Density of the input.

6.1 Configuration of experiment

The experimental setup is configured to test forced vibration of the 1DoF oscillator under a kinematic excitation, where the pneumatic adaptive absorber and the linear springs play the role of the isolation system.

The configuration of the stand is adopted in accordance with the presented scheme. The mass element guided on the frame is supported on the pneumatic absorber and the coil springs. The sinusoidal excitation is applied from the bottom side with its amplitude at 2 mm and the frequency of a sinusoidal sweep ranging from 2 to 6 Hz. The tested bandwidth is defined in order to cover the eigenfrequency of the setup. Moreover, the system is tested with a set of masses and with varied initial pressures in the cylinder. The precise data on the experimental settings is given in Tab. 3.

Table 3: Parameters of experimental excitation

Excitation type	Sinusoidal chirp			
Excitation amplitude	A [mm]	2		
Excitation bandwidth	f [Hz]	$2-6$		
Inertial mass	m_u [kg]	17	27	37
Stiffness of springs	k [N/m]	3000		
Initial pressure	p [kPa]	250	350	450

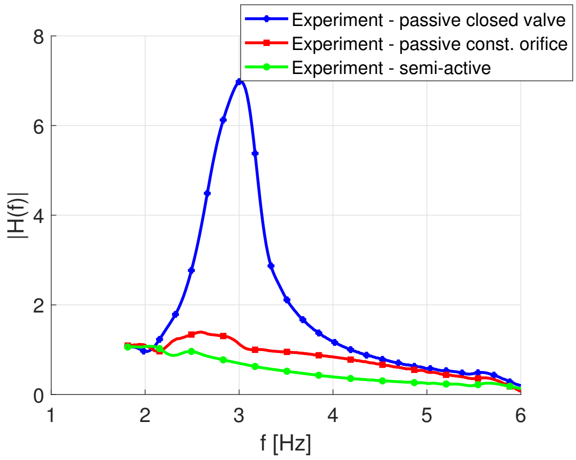

Figure 6: Experimentally obtained Transfer Functions of the system in three modes of operation: passive with continuously closed valve, passive with constant orifice, semi-active.

6.2 Proof of concept

Three operational modes of the demonstrator are verified by means of experiments. The testing programme comprises a frequency sweep of the excitation in the range given in Tab. 3. The range is adapted to cover the eigenfrequencies of the system, which vary due to changes of the mass and the initial pressure. The programme consists of the following three modes of operation: 1 - with permanently closed valve, 2 - with a constant orifice, 3 - with the valve open in accordance with the control algorithm. The first set of results contains response of the system configured with $m = 37kg$ and initial pressure in the absorber $p_0 = 250kPa$. A comparison between the responses of the system in three modes of operation is depicted on a graph in Fig. 6. The first plot indicates that the eigenfrequency in the first mode of operation is 3 Hz (Fig. 6). It also reveals small damping in the system and a significant resonant response. The second plot, reflecting the second mode of operation, is characterised by eigenfrequency 2.2 Hz and by enhanced passive damping due to loses associated with the gas flow between the cylinder volumes. The change in the eigenfrequency value comes from lowering of the effective stiffness in the system. It results from lower compression in the cylinder volumes. The third plot on the graph depicts the response of the device under the controlled mode and is characterised with the highest damping, which allows for keeping the amplification of the system below unity.

The most significant result is that the system controlled with the proposed method does not amplifies the amplitude of vibration. The effect is valid for the whole range of the excitation frequency. The responses indicate that in the resonant range the application of the control strategy allows for obtaining an 85 % decrement of the vibration amplitude in comparison to the closed valve case. An important advantage of the system is that, the semi-active system has a possibility of adaptation to the changing environment and sustain its performance. It can also be noticed that in the post-resonant frequency range the passive system provides the best isolation.

Therefore, a conclusion can be drawn that a potential control strategy should be hybrid and combine the

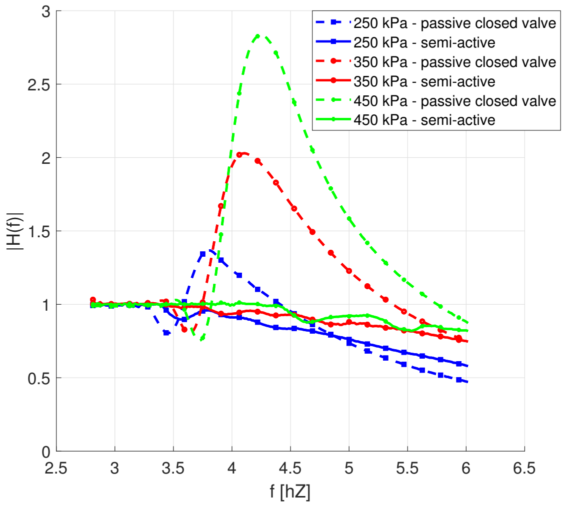

Figure 7: Transfer Functions of the PAA absorber on a 1DoF kinematic vibration test stand. Suspended mass 17 kg. Comparison of passive and controlled responses of the system with a variety of the initial pressures.

semi-active mode for the resonant regions and the passive mode for the post-resonant ranges of the frequency. Such a solution allows for obtaining the most gain from adopting the control strategy for the isolation task. A convenience of this control concept is that the mentioned hybrid system could be operated as passive by default and would need to be switched to semi-active periodically.

6.3 Robustness to variations in conditions

The objective of this experimental study is to verify the adaptivity of the proposed isolation system to variations in operating conditions, like modification of the suspended mass, frequency of excitation or gas pressure in the device. The investigation covers a comparison between two types of responses: 1) passive case with closed valve and 2) controlled case. The responses of the system are collected for set magnitudes of mass and values of the initial pressure as denoted in Tab. 3.

The results are depicted in the form of Transfer Functions graphs consisting of the six operation cases each. Data referring to passive responses presented in Fig. 7 (m=17 kg) indicates that the increase in the pressure affects the eigenfrequency of the system. It is a consequence of an increment of the effective stiffness of the absorber. The modification is noticeable on the plots as different corresponding frequencies of the peaks: 3.8 Hz, 4.1 Hz and 4.3 Hz. Moreover, the graph depicts a comparison between cases with and without the control. All the experimental responses reveal that amplitude is significantly reduced in the resonant region. Moreover, the data depicted on the graph reveals that the change of mass does not influence the amplification of the system when the control is on. In all tested cases the Transfer Function values were below unity.

Fig. 8 depicts Transfer Functions of the system acquired for the mass equal 27 kg. The presented results do not allow for observation of the eigenfrequency modification due to change in the initial gas pressure. In the case of the controlled process the Transfer Functions remain below unity for the whole bandwidth. The results demonstrate robustness of the system to the introduced disturbing factors.

The above mentioned results suggest a potential for a robust operation of the analysed system.

7 Discussion

The presented study provides opportunity of getting gain from the switching stiffness concept application and advantages of the pneumatic systems. It should be highlighted that the proposed pneumatic isolation system

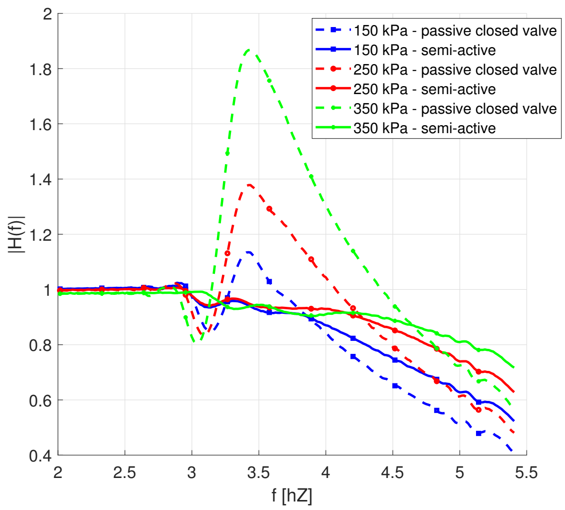

Figure 8: Transfer Functions of the PAA absorber on a 1DoF kinematic vibration test stand. Suspended mass 27 kg. Comparison of passive and controlled responses of the system with a variety of the initial pressures.

is substantially different form the concepts known from the literature. The presented concept introduces adaptive response of the system, which allows for carrying it on large displacements. Such a configuration significantly increases the number of potential applications.

The introduced vibration isolation concept is dedicated to be operated on semi-active devices. The same category is represented by a Sky-Hook semi-active control approach [9], which may be recognised as based on similar principles. The Sky-Hook strategy is intended to control the semi-active damping elements that enable for switching between the low and high damping intensity in real time. The principle of the strategy is to set damping high when the damping force is acting with regard to the kinetic energy of the suspended mass. Otherwise, the damping is set to low value. The on-off strategy is carried out by this method. Both the Sky-Hook damping modification method and the switching stiffness on pneumatics are based on the kinematics state monitoring and the velocity zero-crossings. Both of the strategies refer that control signal generation based on the energy state of the vibrating system. The important difference is that the presented concept considers switching stiffness. The introduced semi-active technique enables to command a damping effect in pneumatic systems without necessity of an additional external damping device employment.

8 Concluding remarks

The study presents a concept of a semi-active vibration isolation system and an appropriate control algorithm. The conclusions from the work can be formulated as follows:

1. The presented concept exhibits two important advantages in comparison to passive systems. The first is ability of adaptation to varying operational conditions and the second is introduction of dissipative functionality by virtue of the switching stiffness application.

2. The presented semi-active concept outperforms a active systems as it manages the energy in the pneumatic system and avoids the necessity of the external pneumatic energy utilisation. The presented system amplification examples reveal that the proposed control strategy allows for preserving its value below unity.

3. The appropriately designed switching stiffness strategy enables controlled dissipation of the mechanical energy during the complete vibratory movement.

Acknowledgements

The author thankfully acknowledges the support of the National Science Centre, Poland, granted under grant agreement 2020/39/B/ST8/02615.

References

[1] V. M. Ryaboy, “Static and dynamic stability of pneumatic vibration isolators and systems of isolators,” *Journal of Sound and Vibration*, vol. 333, pp. 31 – 51, 2014.

[2] A. J. Nieto, A. L. Morales, J. M. Chicharro, and P. Pintado, “Unbalanced machinery vibration isolation with a semi-active pneumatic suspension,” *Journal of Sound and Vibration*, vol. 329, pp. 3 – 12, 2010.

[3] H. Pu, X. Luo, and X. Chen, “Modelling and analysis of dual-chamber pneumatic spring with adjustable damping for precision vibration isolation,” *Journal of Sound and Vibration*, vol. 330, pp. 3578 – 3590, 2011.

[4] Q. Meng, X. Yang, W. Li, E. Lu, and L. Sheng, “Research and analysis of quazi-zero-stiffness isolator with geometric non-linear damping,” *Shock and vibration*, vol. 2017, p. ID 6719054, 2017.

[5] L. T. Dahn and K. K. Ahn, “Active pneumatic vibration isolation system using negative stiffness structures for a vehicle seat,” *Journal of Sound and Vibration*, vol. 333, pp. 1245 – 1268, 2014.

[6] E. Palomares, A. Nieto, A. Morales, J. Chicharro, and P. Pintado, “Numerical and experimental analysis of a vibration isolator equipped with a negative stiffness system,” *Journal of Sound and Vibration*, vol. 414, pp. 31 – 42, 2018.

[7] G. Mikułowski, “Vibration isolation concept by switchable stiffness on a semi-active pneumatic actuator,” *Smart Materials and Structures*, vol. 30, p. 075019, 2021.

[8] G. Mikułowski, R. Wiszowaty, and J. Holnicki-Szulc, “Characterization of a piezoelectric valve for an adaptive pneumatic shock absorber,” *Smart Materials and Structures*, vol. 22, no. 12, p. 125011, nov 2013. [Online]. Available: https://doi.org/10.1088%2F0964-1726%2F22%2F12%2F125011

[9] J. N. Potter, S. A. Neild, and D. J. Wagg, “Generalisation and optimisation of semi-active, on-off switching controllers for single degree-of-freedom systems,” *Journal of sound and vibration*, vol. 329, pp. 2450 – 2462, 2010.