Vibration Isolation Pneumatic System with a Throttle Valve

L. Pesik, A. Skarolek, and O. Kohl

Abstract The paper shows a possibility of the tuning mechanical system by means of two pneumatic springs in a differential configuration connected with a throttle valve. The springs are inserted into the lead mechanism and connected to its parts, and to its supporting platform. The vibrations, transferred from the kinematic excitation of the base, are intended to be minimized. The vibration isolation by means of pneumatic springs is available in many technical systems, e.g. in supports of heavy machinery as well as in systems characterized by the human interaction, such as driver seats, ambulance couchettes, etc. The pneumatic springs provide the option of adaption of the stiffness, and herewith the adaption of the natural frequency of the system according to the exciting frequency. In cases of application of the object vibration isolation, they can change the load characteristics in a relative large range. In the studied case of the differential spring configuration, the springs are connected with an air pipe to the throttle valve. The air being exchanged during the motion period comes through the valve, the cross-section of which determines the time delay of the pneumatic sub-system thus creating a hysteresis of load characteristic of the spring support. This brings an additional, controllable damping to such a system that is profitable in most vibration isolation cases.

Keywords Pneumatic spring • Vibration isolation system

1 Introduction

There are many technical problems that are connected with a necessity of the minimization of the vibrations. Some of them are present during the run of the machines when intensive dynamic forces are transmitted into their foundations. In these cases, the elastic supports are used for an effective dynamic isolation. Here, the basic principle is

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grounded on the sufficient difference between the force excitation frequency and resonant frequency of the dynamic system. For processing machines, the excitation frequency is usually constant and relates to the operation speed. The vibration isolation system supporting these objects is created by springs with unchangeable load characteristics. On the other hand side, especially in the automotive industry, there are many types of equipment which have an excitation through the movement of the foundation. The typical instance of this set is a driver seat. With vibration isolation of these objects, there are further problems that come from variable excitation frequency. It is clear that the condition of the sufficient difference between the force excitation frequency and resonant frequency of the dynamic systems cannot be ensured at all times. In these cases, it is necessary to use special supports with the possibility of the stiffness changing, and to tune the natural frequency of the system appropriately.

2 Principle of the Resilient Pneumatic Support

The resilient support is created by a lead mechanism, in which two pneumatic springs are inserted, see (Fig. 1). Here they have their ratios of transmission. One of the springs has a positive ratio and the other one a negative one. This means that the first of the pneumatic springs brings the support upwards and the second one downwards. The force effect of the second spring on the support has a similar effect of its loading. This configuration gives possibility to adapt the stiffness as required by the changing of the air pressure in the second spring. This change can be executed in the still stand or during the movement of the supported object.

In the studied case, the pneumatic springs are connected not only by the lead mechanism but also by pneumatic tubing with a throttle valve. This configuration is called differential. The effective cross-section of the valve can be adjusted standstill or can be controlled during the motion. In this paper, we deal only with the former situation, i.e. the selecting of the optimum throttle valve cross-section to achieve the most promising amplitude-frequency characteristic of the mechanism. Considering the two boundary cases of the throttle valve setup, the closed valve and the fully open valve with the infinite cross-section, the valve would have following effects. Fully closed valve does not permit any air exchange between the springs; the mechanism is stiff with no additional energy dissipation. The pressures inside the springs change in time with 180° phase shift. For a very large cross-section of

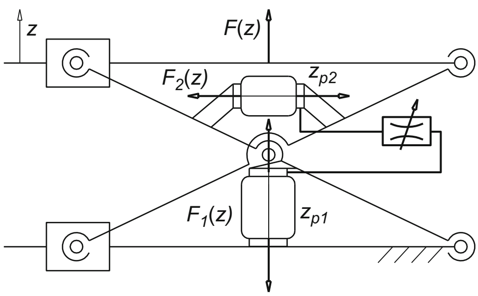

Fig. 1 Principled scheme of the differential pneumatic support with throttle valve

the valve, the stiffness of the mechanism is significantly reduced, the pressure inside the springs is changing in phase as they are now equal all the time, but again there is virtually no additional energy dissipation. The valve setup between the two extremes will allow for maximizing energy dissipation by creating a certain phase shift between spring pressures and thus spring forces [1–3].

3 State Equation of the Pneumatic Mechanical System

Equation of motion of the mechanism platform with reduced mass m (including the part of the mechanism, seat and the passenger) is simple

$$m\frac{d^2}{dt^2}(z(t)+u(t))+b\frac{d^2}{dt^2}(z(t)-u(t))=-mg+F, \tag{1}$$

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$$m\frac{d^2}{dt^2}(z(t)+u(t))+b\frac{d^2}{dt^2}(z(t)-u(t))=-mg+F, \tag{2}$$

where $z(t)$ is the absolute displacement of the platform, $u(t)$ is the displacement of the base under the kinematic excitation, b is the construction damping of the mechanism, which can be observed experimentally, mg the static load of the mechanism [4, 5].

The function F is the equivalent force from the springs

$$F=i_{p1}(z)\cdot F_1\left(z_{p1},p_{p1}\right)+i_{p2}(z)\cdot F_2\left(z_{p2},p_{p2}\right), \tag{3}$$

determined by means of transmission functions $i_{p1}(z)$ and $i_{p2}(z)$

$$i_{p1}(z)=\frac{z_{p1}(z)}{z}, \qquad i_{p2}(z)=\frac{z_{p2}(z)}{z}. \tag{4}$$

Spring displacements $z_{p1},\ z_{p2}$ are also determined by these ratios

$$z_{p1}=i_{p1}(z)\cdot z, \qquad z_{p2}=i_{p2}(z)\cdot z. \tag{5}$$

Air pressures inside the springs obey the state equation of ideal gas

$$p_{p1}=\frac{m_{a1}r\ T}{V_1\left(z_{p1}\right)}, \quad p_{p2}=\frac{m_{a2}r\ T}{V_2\left(z_{p2}\right)}, \tag{6}$$

where m_{a1} and m_{a2} are masses of the air enclosed inside the springs, r and T is specific gas constant and temperature respectively [6–8]

The air exchange between the springs is described by isentropic air flow through the throttle valve. In the next two equations, the rate of air exchange depends on pressures p_A and p_B; the pressure p_A signs the higher pressure of p_{p1}, p_{p2} at given time, p_B is the other one. Which of the two pressures is higher determines the sign of the flow rate. The rate of air mass is then

$$\frac{\mathrm{d}m_{a1(a2)}}{dt} = A_v\, c\, p_A \sqrt{\frac{2}{rT}\frac{\kappa}{\kappa-1}\left(\left(\frac{p_B}{p_A}\right)^{\frac{2}{\kappa}} - \left(\frac{p_B}{p_A}\right)^{\frac{\kappa+1}{\kappa}}\right)}, \tag{7}$$

for subcritical flow conditions, where $p_B/p_A \geq \beta^*$ or

$$\frac{dm_{a1(a2)}}{dt} = A_v\, c\, p_A \sqrt{\frac{2}{rT}\frac{\kappa}{\kappa+1}\left(\frac{p_B}{p_A}\right)^{\frac{2}{\kappa-1}}}, \tag{8}$$

otherwise. Critical pressure ratio β^* is

$$\beta^* = \left(\frac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}}, \tag{9}$$

where κ is specific heat ratio for the air. In the Eqs. (7) and (8) are c discharge coefficient and A_v cross-section of the throttle valve.

Differential Eq. (6) supplemented by differential equation for air mass inside the springs fully describe the presented pneumatic-mechanical system. We have considered closed pneumatic system, so the air masses are bound by condition [9]

$$m_{a1} + m_{a2} = m_a\,. \tag{10}$$

4 Simulation Results

On the Fig. 2, there are depicted three load characteristics of the mechanism under kinematic excitation of 0.6 Hz (graph A) and 1.1 Hz (graph B). The selected cases are 1, 1.4 and 2 mm of the throttle valve nozzle [10].

5 Conclusion

This solution presents a pneumatic spring system with a differential configuration and the throttling of the air flow between the springs. The springs act one against the other. The cross-section of the throttling element permits the changing of the amplitude-frequency characteristic with regards to the position and magnitude of the peak resonant amplitude. This system can be used with benefits to the vibration

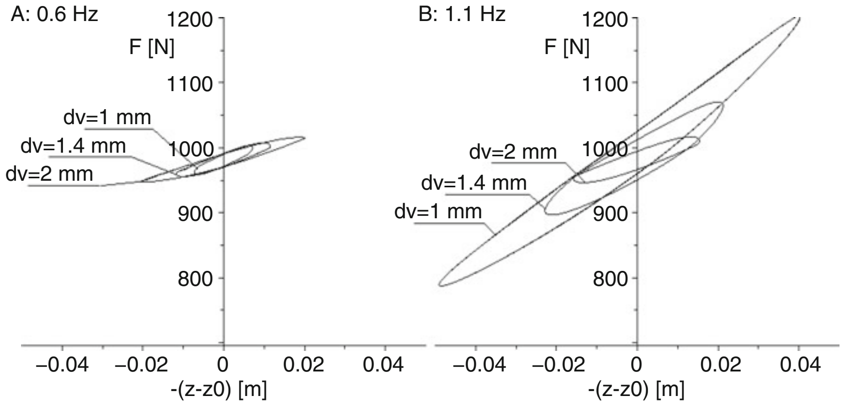

Fig. 2 Mechanism dynamic load characteristic, three valve setup at two excitation frequencies A: 0.6 Hz, B: 1.1 Hz

isolation of the objects of the systems with kinematic excitation, e.g. driver seats, ambulance couchettes, etc. Based on the frequency spectrum of excitation, it is possible to choose the optimum cross-section of the throttling element and achieve efficient vibration isolation in a relatively broad range of low excitation frequencies.

References

1. C.M. Harris, *Shock and Vibration Handbook*, 5th edn. (McGraw-Hill, NewYork, 2005)
2. R. Uhlir, P. Mossoczy, P. Maly, Experimental testing of agricultural trailer, in *Book of Proceedings of 54th International Conference of Machine Departments* (Technical University of Liberec, Liberec, 2013), pp. 353–358. ISBN 978-80-7372-986-8
3. D.J. Whitehouse, *Handbook of Surface and Nanometrology*, 2nd edn. (CRC Press, Boca Raton, 2011). ISBN 978-1-4200-8201-2
4. Y. Girfhar, A. Xu, B. B. Dey, M. Meghjani, F. Shkurti, I. Rekleitis, G. Dudek, MARE: Marine Autonomous Robotic Explorer, in *Mobile Robotics Lab* [online] (McGill University, 2011), [vid. 2011_10_10]. Available via: http://www.cim.mcgill.ca/~mrl/pubs/anqixu/iros2011_boat.pdf
5. V. Zeman, Z. Hlaváč, *Kmitání mechanických soustav* (Západočeská univerzita v Plzni, Plzeň, 2010). ISBN 978-80-7043-337-9
6. Z. Bradský, R. Vrzala, *Mechanika III. Dynamika* (Skripta VŠST, Liberec, 1986)
7. H. Dresig, F. Holzweißig, *Maschinendynamik*, 8th edn. (Springer, Berlin, 2008)
8. L. Pešík, A. Skarolek, Heat transfer effects on vibration isolation differential pneumatic system, in *Transactions of the Universities of Košice: Research Reports from the Universities of Košice* (Technická univerzita v Košiciach, Košice, 2012), pp. 109–114. ISBN 1335-2334
9. L. Pešík, A. Skarolek, Tuning of vibration isolation differential pneumatic system by means of trottle valve, in *Transactions of the Universities of Košice: Research Reports from the Universities of Košice* (Technická univerzita v Košiciach, Košice, 2011), pp. 191–196. ISBN 1335-2334
10. M. Mansouri, Q. Ilham, H. Snoussi, C. Richard, Adaptive quantized target tracking in wireless sensors networks. Wirel. Netw. **17**(7), 1625–1639. ISSN: 1022-0038. 2011