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Damping Properties of Driver Seat Differential Pneumatic System

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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.

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 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.

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.

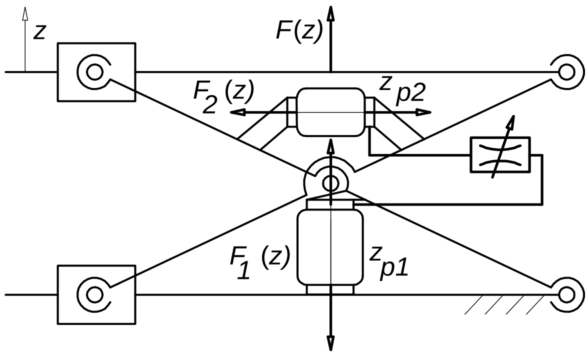

Fig. 1. Principled scheme of the differential pneumatic support with throttle valve.

State Equation of the Pneumatic Mechanical System

The parameters given are the geometrical characteristics of the both pneumatic springs. They are polynomial functions of effective surfaces $S_1(z_{p1})$ and $S_2(z_{p2})$ and volumes $V_1(z_{p1})$ and $V_1(z_{p1})$.

Having these characteristics, the spring forces are simply

$$F_1(z_{p1}, p_{p1}) = S_1(z_{p1}) \cdot p_{p1}, \quad F_2(z_{p2}, p_{p2}) = S_2(z_{p2}) \cdot p_{p2}. \tag{1}$$

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

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

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. The function F is the equivalent force from the springs

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

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

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

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

$$p_{p1} = \frac{m_{a1}\, r\, T}{V_1(z_{p1})}, \quad p_{p2} = \frac{m_{a2}\, r\, T}{V_2(z_{p2})}, \tag{5}$$

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.

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

$$\frac{\mathrm{d}\, m_{a1(a2)}}{\mathrm{d}t} = A_v\, c\, p_A \sqrt{\frac{2}{rT} \frac{\kappa}{\kappa - 1} \left(\left(\frac{p_B}{p_A} \right)^{\frac{2}{\kappa}} - \left(\frac{p_B}{p_A} \right)^{\frac{\kappa+1}{\kappa}} \right)}, \tag{6}$$

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

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

otherwise. Critical pressure ratio β^* is

$$\beta^* = \left(\frac{2}{\kappa + 1} \right)^{\frac{2}{\kappa - 1}}, \tag{8}$$

where κ is specific heat ratio for the air. In the equations (Eq. 6) and (Eq. 7) are c discharge coefficient and A_v cross-section of the throttle valve.

Differential equation (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

$$m_{a1} + m_{a2} = m_a. \tag{9}$$

Simulation and Measurement Results

The presented system has been solved in the time domain for a range of throttle valve setup. The aim is to show the effect of the throttle diameter dv to the frequency response function (FRF) of the mechanism. The FRF is here defined by means of Fourier transformation (FT) as

$$G(f) = \frac{FT(z(t))}{FT(u(t))}. \tag{10}$$

In Fig. 2, there are depicted modules of transfer function of mechanism having throttle valve setup ranging from 1 mm to 2 mm in throttle nozzle diameter dv. This figure shows how resonant peak frequency can be changed by air throttling. The most interesting is the case of 1.4 mm of valve nozzle, which seems to be the transition between two previously discussed extremes; the fully close valve and the valve with a very large cross-section. This case exhibits both resonant peaks, the 0.6 Hz and the 1.1 Hz, but with very limited magnitude. These results suggest that no additional damper may be needed in some vibration isolation problems. Similar results can be obtained by measurement of the real designed system (Fig. 3).

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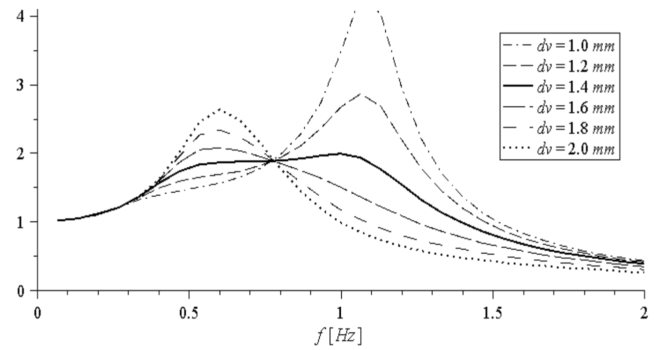

Fig. 2. Simulation results of the transfer function of the system versus diameter of throttle valve dv.

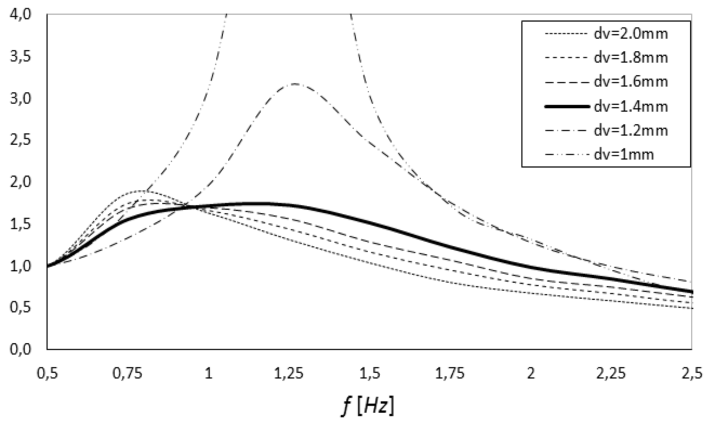

Fig. 3. Measurement results of the transfer function of the system versus diameter of throttle valve dv.

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. This system can be used with benefits to the vibration 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 damping of vibration in a relatively broad range of low excitation frequencies. Similar results can be seen by measurement of the real system.

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