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EVALUATION OF A DRIVER’S SEAT’S DYNAMIC PROPERTIES

Summary. The springing support in a driver’s seat is a very serious issue, such that manufacturers are increasing their efforts to optimize the dynamic properties of this kind of seat. The main optimization criterion is vibration insulation efficiency with regard to health research and associated health standards and regulations. This article deals with the definition of the optimal driver’s seat properties in relation to two examples of springing systems with different kinds of damping. The first case involves the springing support of a driver’s seat that uses a pneumatic spring and a telescopic hydraulic damper. In the second case, the damping effect is achieved by two pneumatic springs whose forces effect a phase shift due to a throttle valve located in the connecting piping system.

Keywords: driver’s seat, vibration minimization, oscillation, stiffness, damping

1. INTRODUCTION

The springing support in a driver’s seat (Fig. 1) consists of a guide mechanism, in which is placed a pneumatic spring. Parallel to the air spring is a guide mechanism, which is connected to a damper (Fig. 2). Force effects $F_1(z_r)$ and $F_2(\dot{z}_r)$ are phase-shifted by $\pi/2$.

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Fig. 1 Vehicle driver’s seat

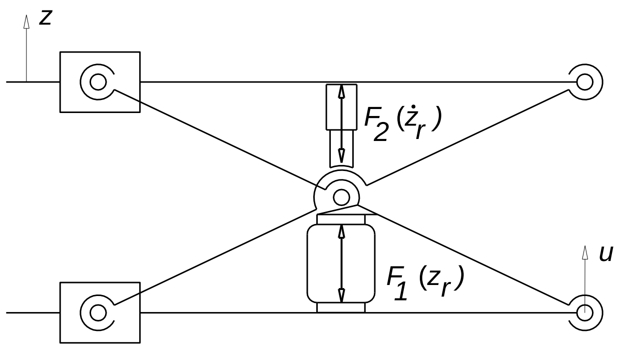

Fig. 2. Springing seat support with a hydraulic damper

In the original author’s solution, there are two air springs that, by themselves, create force effects that act against each together. They are pneumatically connected through the throttle valve, which achieves phase-shifting as a result of their force effects $F_1(z_r)$ and $F_2(z_r)$, and in turn a damping oscillatory motion (Fig. 3).

When the vehicle is moving, the seat is kinetically energized by the movement of the floor. The excitation function u is, in fact, in respect of frequency and amplitude, very diverse, while its character depends on the road surface, vehicle suspension and speed. Manufacturers of driver’s seats asses the dynamic properties of the springing support and the vibration insulation effect based on their own practices and policies. Typically, measurement and evaluation of the feedback z to the excitation signal u take place when a vehicle is driving within a test polygon. General evaluation, enabling an objective comparison of driver's seats from different manufacturers, in terms of vibration insulation efficiency, is practically non-existent.

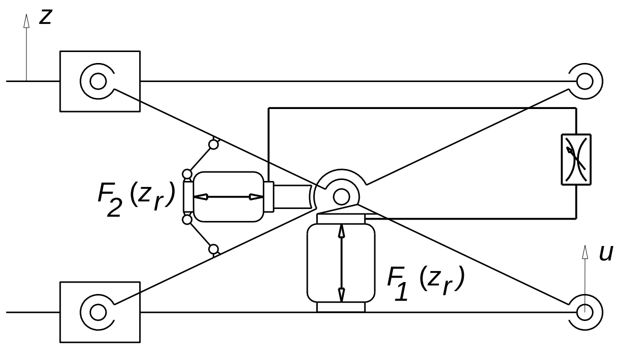

Fig. 3. Springing seat support with two air springs

2. MECHANICAL MODEL

The basis for evaluating the driver's seat suspension system is undoubtedly the creation and compilation of a mechanical model and solving equations of motion.

For the commonly known system shown in Fig. 2, the equation of motion can be written by simplifying assumptions of linearization dynamic parameters in the following form:

$$m\ddot{z} + i_b b(\dot{z} - \dot{u}) + i_k k(z - u) = 0, \tag{1}$$

where m is the reduced mass of the moving parts of the seat including the driver, b is the coefficient of vibration damper damping, k is the spring stiffness, i_b and i_k are the springs and dampers transfers, $u(t)$ is the displacement of the base under the kinematic excitation, and $z(t)$ is the absolute displacement of the object.

It is then possible to rewrite Equation (1) as follows:

$$m(\ddot{z}_r + \ddot{u}) + i_b b\dot{z}_r + i_k k z_r = 0, \tag{2}$$

where $z_r(t)$ is the relative displacement of the object and the base.

For pneumatic–mechanical systems, as shown in Fig. 3, it is possible to write the equation of motion in the following form:

$$m(\ddot{z}_r + \ddot{u}) + b_{constr}\dot{z}_r + i_1 F_1(z_r) + i_2 F_2(z_r) = -mg, \tag{3}$$

where b_{constr} is the construction damping coefficient, while i_1 and i_2 are transfers of springs in mechanism.

The springs of the resilient differential pneumatic support have their geometrical characteristics given by polynomial functions of effective surfaces $S_1(z_{p1})$ and $S_2(z_{p2})$, as well as volumes $V_1(z_{p1})$ and $V_2(z_{p2})$. The forces in the springs are also represented as follows:

$$F_1(z_{p1,}\, p_{p1}) = S_1(z_{p1})\, p_{p1} \tag{4}$$

and

$$F_2(z_{p2,}\, p_{p2}) = S_2(z_{p2})\, p_{p2} \tag{5}$$

z_{p1} and z_{p2} represent deformations of the springs, p_{p1} and p_{p2} represent air pressures in the springs, and $S_1(z_{p1})$ and $S_2(z_{p2})$ represent effective surfaces of the springs. The equation of motion in the support with the mass m of the supported object is simple:

$$m(\ddot{z}_r + \ddot{u}) + b_{constr}\dot{z}_r + F(z_r) = -mg, \tag{6}$$

where $z_r(t)$ is the relative displacement of the object and the base, $u(t)$ is the displacement of the base under the kinematic excitation, $z(t)$ is the absolute displacement of the object, b_{constr} is the construction damping of the mechanism, and mg is the static load of the mechanism. The function $F(z_r)$ is the equivalent force from the springs:

$$F(z_r) = i_1(z_r)F_1(z_r) + i_2(z_r)F_2(z_r), \tag{7}$$

where $i_1(z_r)$ and $i_2(z_r)$ are transmission ratios of the springs.

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

$$p_{p1} = \frac{m_{a1}\, r\, T_1}{V_1(z_{p1})}, \tag{8}$$

and

$$p_{p2} = \frac{m_{a2}\, r\, T_2}{V_2(z_{p2})} \tag{9}$$

where m_{a1} and m_{a2} are masses of the air enclosed inside the springs, r is the specific gas constant, and T_1 and T_2 are temperatures of the air inside the springs.

The air exchange between the springs is described by isentropic airflow 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 denotes the higher pressure of p_{p1} and p_{p2} at a given time, while p_B represents the remaining one. The sign of the flow rate is determined by the two pressures that are higher. The rate of air mass is then calculated thus:

$$\frac{\mathrm{d}m_{a1(a2)}}{\mathrm{d}t} = A_v c p_A \sqrt{\frac{2}{rT_A}\frac{\kappa}{\kappa-1}\left[\left(\frac{p_B}{p_A}\right)^{\frac{2}{\kappa}} - \left(\frac{p_B}{p_A}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{10}$$

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

$$\frac{dm_{a1(a2)}}{dt} = A_v c p_A \sqrt{\frac{2}{rT_A}\frac{\kappa}{\kappa-1}\left(\frac{p_B}{p_A}\right)^{\frac{2}{\kappa-1}}}. \tag{11}$$

Critical pressure ratio $\beta^*$is determined thus:

$$\beta^* = \left(\frac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}} \tag{12}$$

where κ is specific heat ratio for the air. In Equations (9) and (10), c is the discharge coefficient and A_v is the cross-section of the throttle valve.

The differential Equation (8), which is supplemented by a differential equation for air mass inside the springs, describes the pneumatic-mechanical system being presented in this article. As we have considered a closed pneumatic system, the air masses are bound by the following condition:

$$m_{a1} + m_{a2} = m_a \tag{13}$$

Given that thermal effects are within the scope of this article, temperatures T_1 and T_2 are considered as state variables. The evolution of the air temperatures in time is described by the following differential equations, which are derived from internal energy conservation (first law of thermodynamics):

$$c_v m_{a1} \frac{dT_1}{dt} = -\alpha_T (T_1 - T_\infty) - p_{p1} \frac{dz_{p1}}{dt}\left(S_1 + \frac{dS_1}{dz_{p1}} z_{p1}\right) + c_v \frac{dm_{a1}}{dt}(T_2 - T_1) \tag{14}$$

$$c_v m_{a2} \frac{dT_2}{dt} = -\alpha_T (T_2 - T_\infty) - p_{p2} \frac{dz_{p2}}{dt}\left(S_2 + \frac{dS_2}{dz_{p2}} z_{p2}\right) + c_v \frac{dm_{a2}}{dt}(T_1 - T_2) \tag{15}$$

where c_v is the isochoric thermal capacity of the air, T_∞ is the ambient temperature, and α_T is the coefficient of heat transfer. For the purpose of this article, the heat transfer coefficient is presumed to involve both the conductive and convective heat transfer.

3. EVALUATION OF SPRINGING SUPPORT

Based on the solutions of equations of motion in the preceding section, a transmission frequency characteristics can be obtained, which are usually concerned with the displacement amplitude of the oscillatory movement of the seat. According to these data, the function of the spring base, due to changes in the position of the driver in relation to the vehicle controls, can be evaluated, thereby allowing the appropriate size of the damping coefficient to be set.

Regarding the springing support with a hydraulic damper, as shown in Fig. 2, the transmission frequency characteristics are based on those presented in Fig. 4.

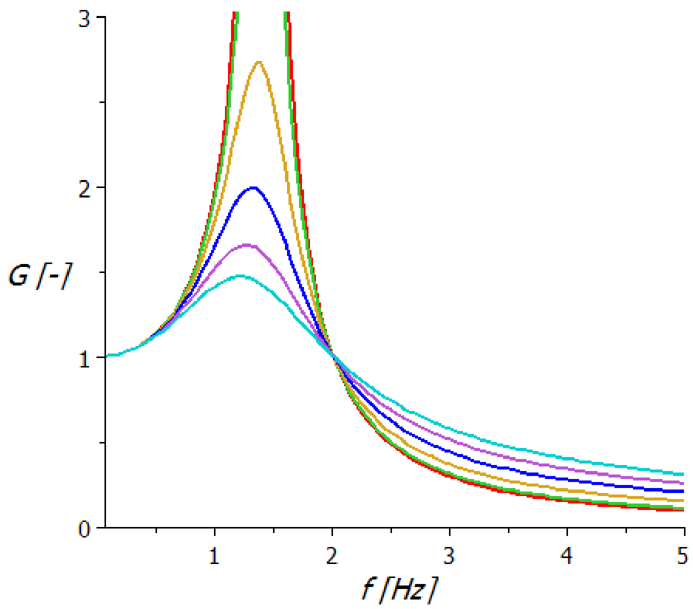

Fig. 4. Transmission frequency characteristics of springing support with a linear hydraulic damper

Particular damping characteristics correspond to the relative damping coefficient of 0.01, and 0.1 to 0.5 with step 0.1. Generally, it is possible to say that damping has a positive effect when the system is erected near to its natural frequency. Outside of this region, the damping importance, with respect to the displacement amplitude, is either small or rather negative.

Regarding the springing support, as shown in Fig. 3, the transmission frequency characteristics are based those presented in Fig. 5.

Individual characteristics correspond to different intensities throttling the flow of compressed air between the two air springs in the support. The diameters of the holes are graded from 1 mm to 3.5 mm in steps of 0.5 mm. The damping character of the oscillation in this case is favourable in the wider frequency range and the gain does not exceed the value of 1.5.

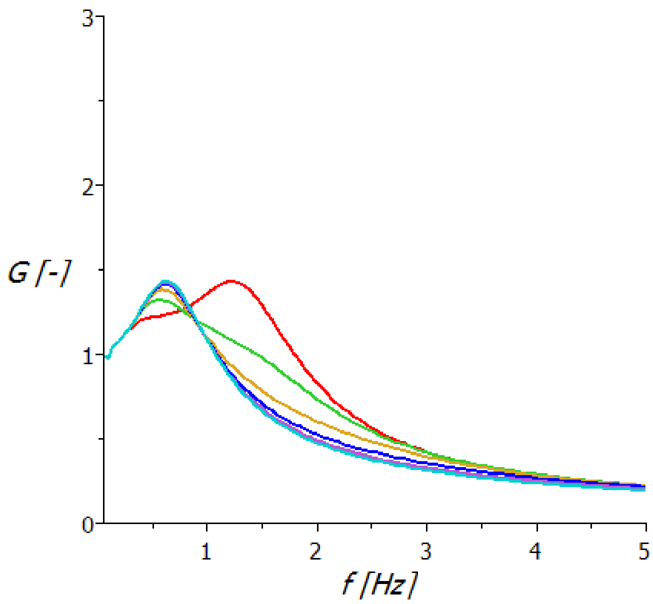

Fig. 5. Transmission frequency characteristics of springing support with two air springs

Another evaluation criterion relating to vibration insulation for a driver's seat concerns acceleration value and frequency, which act on the driver and adversely affect driving comfort.

Fig. 6 presents acceleration transmission frequency characteristics of springing support with a linear hydraulic damper. It shows that damping decreases displacement amplitude, but significantly increases the acceleration values. This fact means that vibration insulation system seats, located outside the resonance region, require negligible damping.

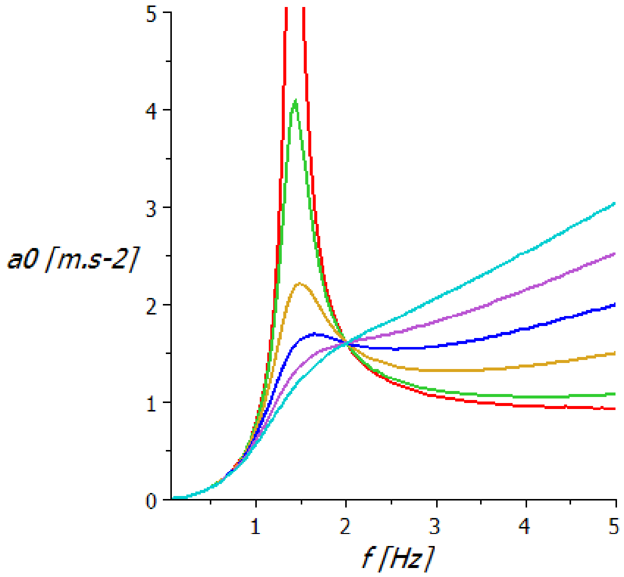

Fig. 6. Acceleration amplitude frequency characteristics of springing support with a linear hydraulic damper

Fig. 7 presents the acceleration amplitude frequency characteristics of springing support with two air springs.

This springing system significantly reduces amplitude acceleration values compared to the classic springing support design. Acceleration amplitudes increase almost linearly with excitation frequency.

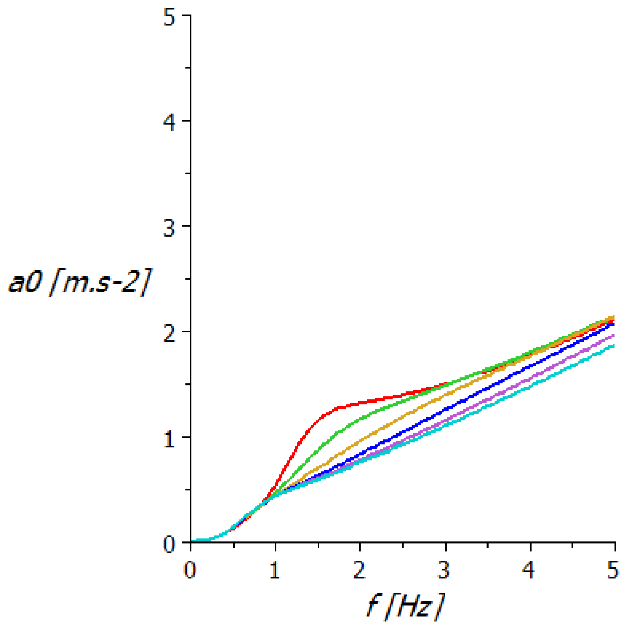

Fig. 7. Acceleration amplitude frequency characteristics of springing support with two air springs

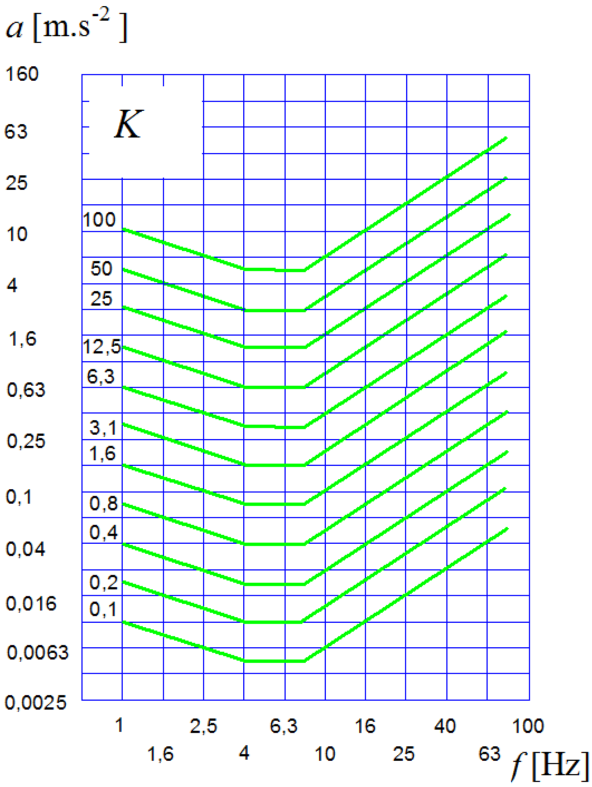

Fig. 8. Amplitude and frequency dependence in relation to the comfort of the seated human on acceleration

Vibration isolation properties of the driver's seat can be quantified on the basis of hygiene standards. Medical research has indicated that the result is defined by the sitting person's comfort under the influence of acceleration with a certain amplitude and frequency. Fig. 8 presents graphs taken from the internationally valid standards ISO 2631, ISO 5982 and VDI 2057.

Using these findings could contribute to an objective evaluation of vibration insulation properties of driver's seats, as well as remove the element of subjective assessment, which currently dominates among the manufacturers of driver's seats.

Using the coefficient K, the vibration insulation level of the driver's seat properties can be deduced. In Fig. 9, dependence on the coefficient K is calculated according to the seat's excitation frequency with a hydraulic telescopic shock absorber, while Fig. 10 presents this dependence on the seat involving two pneumatic springs.

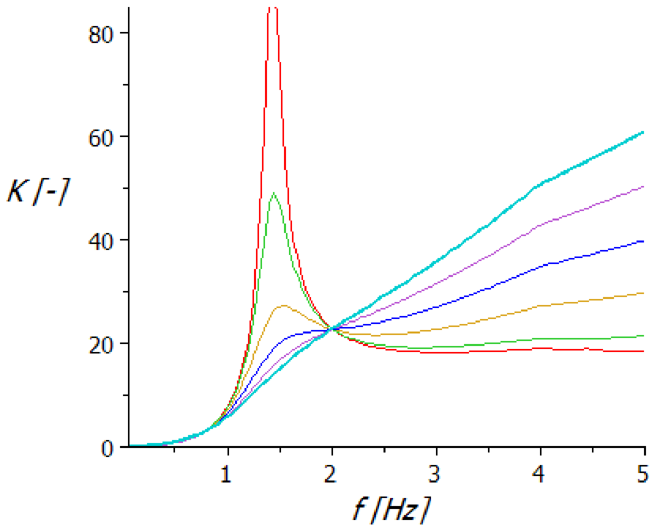

Fig. 9. Frequency dependence the coefficient of seated human loading on the acceleration of a seat with a hydraulic telescopic shock absorber

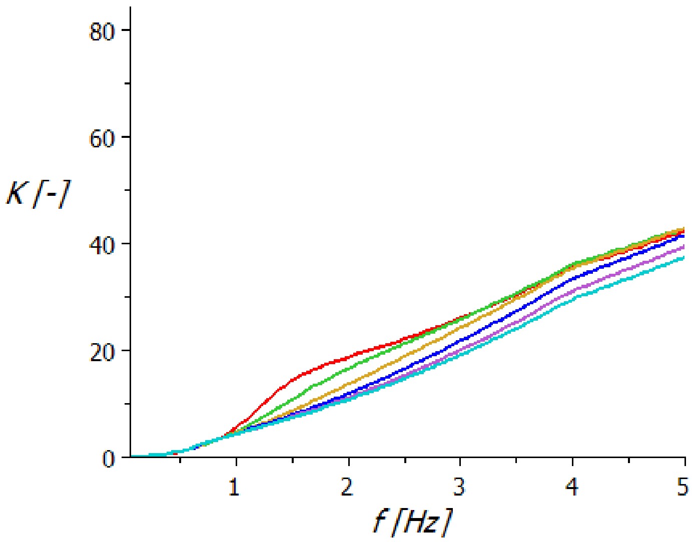

Fig. 10. Frequency dependence the coefficient of a sitting human loading on the acceleration of a seat with two air springs

4. CONCLUSION

There are many technical problems that are connected with the need to minimize vibrations, especially in the automotive industry. There are many types of equipment that produce an excitation through the movement of the foundation. A typical example in this respect a driver's seat. Although vibration isolation is possible for these objects, there are further problems that come from a variable excitation frequency. It is clear that the condition in terms 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 changing the stiffness, as well as tune the natural frequency of the system appropriately. In this article, two possible designs for a driver's seat were presented. One of them is the classical solution with a hydraulic damper.

The second one is an original solution by the authors, involving two pneumatic springs in the guide mechanism.

This original solution comes in the form of a pneumatic spring system with a differential configuration and throttling of the airflow between the springs. The springs act against each other. This system can be used in order to produce benefits to the vibration isolation of objects by using kinematic excitation, e.g., driver's seats, ambulance couchettes etc. Based on the frequency spectrum of excitation, it is possible to choose the optimum cross-section of the throttling element, along with achieving efficient damping of vibrations in a relatively broad range of low excitation frequencies. Similar results can be seen in the measurements obtained in a real system.

Evaluation of both seats is made on the basis of simulating vibration excitation at constant amplitude support displacements at frequencies between 0 and 5 Hz. The obtained frequency characteristics of kinematic quantities are further processed in terms of vertical acceleration coefficient determination regarding the seated human loading.

Calculation results showed that a qualitatively higher level of vibration insulation properties was found in the seat with two air springs.

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