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Short Communication

Position control of seat suspension with minimum stiffness

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Abstract

Quality of vibration isolation of vehicle drivers in the infra-narrow band is the most essential and simultaneously is a hard-hitting goal. Use of the "negative" stiffness' phenomenon is a unique concept to minimize stiffness of a vibration isolating device and improve the quality so that protected object becomes motionless in inertial space. Though control strategy for the device with minimum stiffness sufficiently differs from the approaches based on attenuation of extraneous resonant responses. An approach for positioning such a device is proposed focusing on motion stability in large. In the approach, a model of the device is structured to generate the control criteria in operating immanently unstable mechanism of "negative" stiffness. A control algorithm effecting variability of the device stiffness in terms of the position and velocity data evaluation is considered. The object protected is motionless under the vibration and motion becomes shock-free under non-vibrational excitation even if there is no external damper. The validity of the approach is assessed by numerical experiment and physical measurement for an actual seat pneumatic suspension restructured via the mechanism and controlled by the program.

1. Introduction

Vibration isolation is the most effective method of protection for a man especially vibrosensitive in the infra-frequency range, $f = 0.5-5\,\text{Hz}$ [1]. To achieve this goal one should have a resource in stiffness minimization of a vibration isolating mechanism (VIM). This results in a shift of the fundamental frequency spectrum of vibration isolation system close to zero Hz. One should then prevent end-stop impact of the "over-softened" VIM.

Among the alternatives [2], rodless air-springs seem to be the most suitable for the VIM with minimum stiffness due to advantages such as high self-compliance, high power density, ecological compatibility, low production cost, etc. [3]. Auxiliary wind chambers [4] and shear compliance capacity [5] are the prevailing approaches to minimize spring stiffness. Nevertheless, similar pneumatic suspensions are high-performance in the frequency range, $f \geqslant 10-12\,\text{Hz}$ [2,5,7], and have poor quality in the infra-frequency range.

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It was proven in Refs. [7–10] and many other explorations that coupling a stiffness control mechanism (SCM) containing a spring with adjustable "negative" stiffness enables reducing the stiffness of a VIM with no limit. Due to the chaotic behavior of similar springs [11] and a low mechanical Q-factor close to zero frequency [7,8], however, the VIM must be automatically controlled.

Apparently, the control strategy for the VIM with minimum stiffness essentially differs from well-known concepts of active vibration control based on the attenuation of extraneous resonant responses by using controlled fluid dampers [12–14]. But the vibration isolation quality of the active devices is determined through the compromising elastic-dissipative properties of passive springs that, as is well-known, fail to achieve vibration isolation in the infra-frequency range. In active control of a VIM containing the SCM one should not know the input signal characteristics but focus attention on stabilizing the VIM in operating an immanently unstable SCM.

In enhancing the author's ideas of vibration isolation [7,9–11], this paper presents an approach for the position control of a VIM with minimum stiffness that ensures (a) an immobility under vibrations beginning with the infra-frequency range and (b) shock-free motion under impulsive excitation for the protected object. In the context of this approach we have (1) shown a resource in the stiffness control of a pneumatic VIM by using the SCM, (2) formulated criteria of the variable structure of air-damping control, and (3) demonstrated an algorithm of position control for similar VIMs. The validity of the approach was assessed by numerical experiments and physical measurements applied to an actual seat pneumatic controlled suspension that was redesigned by removing the fluid damper, by coupling the SCM, and by changing the structure of air-damping.

2. Parameter control of a VIM with minimum stiffness

In general, a VIM with minimum stiffness is a system of interacting mechanisms including guide mechanism containing input and output links, load-carrying spring, SCM containing spring with "negative" stiffness, and connecting mechanism. Let us examine a resource of parameter control of the VIM when the controlled air-spring is the load-carrying one.

2.1. Stiffness control

Coupling an SCM and an initial VIM containing a load-carrying spring with "positive" stiffness k_1 enables one to minimize stiffness k of the redesigned VIM. Since a spring with "negative" stiffness has no load-carrying capacity, coupling should be in parallel resulting in alternatives [7]:

$$k(q^*) = k_1(q^*) + \Phi[-k_2(q)] \left\} \begin{array}{l} \to +0 \\ = 0 \\ \to -0 \end{array} \right. \quad \text{(1a–1c)}$$

where q^* is a coordinate in the translation or rotation of the output link in the direction of minimum stiffness; q is a generalized coordinate of the VIM; Φ is a law of connecting; k_2 is adjustable "negative" stiffness.

Conditions (1a–1b) mean that stiffness ratio k can be minimal but "positive" or equal to zero. Using the methods [9,10] for synthesis of the SCM, a designer can easily define a range of magnitude $|-k_2|$ according to k_1-change within stroke z_0 of the input link and payload capacity. By using condition (1a) one might evaluate a resource of stiffness minimization applied to, e.g., a seat pneumatic suspension (see Fig. 1) redesigned by coupling the spring with "negative" stiffness. In the indicated z-direction of the output link motion there will be

$$k(z) = \frac{1}{i_1^{\alpha_n}} \left(A \frac{\mathrm{d}p}{\mathrm{d}s} \pm p \frac{\mathrm{d}A}{\mathrm{d}s} \right) - \tilde{k}_2(\varphi_8)\mu_k\mu_l i_2, \quad (2)$$

where $i_1 = i_1^{\alpha_n}(\beta)$ is a ratio considering misalignment z and s (axe of movement of rubber-cord casing relative to the air-spring supports); α_n is nonlinearity factor, as a rule, $1 \leqslant \alpha_n \leqslant 2$ [6,7,15,16]; A is effective value of the section area of rubber-cord casing; p is working pressure in the air-spring; $\tilde{k}_2$ is dimensionless stiffness rate of

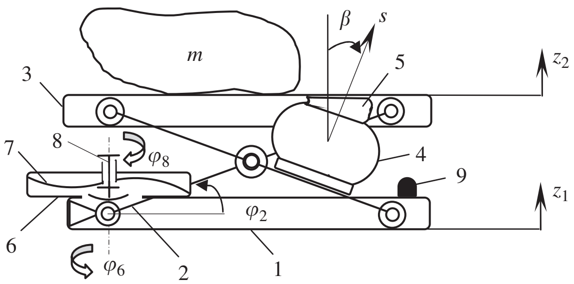

Fig. 1. Control object: schematic diagram of seat pneumatic suspension with minimum stiffness, where the mount excited (position 1), input and output links of guide mechanism (2–3), rubber-cord casing and supports of the air-spring (4–5), frame and spring of SCM (6–7), connecting mechanism (8), elastic end-stop (9).

the SCM spring; μ_k and μ_1 are the scale factors of stiffness, and transformation of rotation into translation, respectively; $i_2 = \varphi_{2/}\varphi_8$ is a gear ratio, here φ_2 and φ_8 are rotation angles of the input link and driving link of connecting mechanism.

As a result of VIM and SCM coupling, the stiffness of VIM can be reduced to an arbitrarily small value. In particular, we have shown in Ref. [7] that the stiffness rates of the most typical seat suspensions can be gradually reduced from $k_1 \in (5250;\ 10\,000]\,\mathrm{N/m}$ to $k = 87-100\,\mathrm{N/m}$.

2.2. Damping control

It is clear that damping in a VIM with minimum stiffness has to be widely varied. On the one hand, it should be light in steady-state vibration motion. In Ref. [7] we have experimentally shown that the relative damping ratio in a standard seat suspension redesigned by coupling the SCM and removal fluid damper can be reduced to $D \approx 0.025$. This might result in improving vibration isolation beginning with a range close to zero frequency. Meanwhile, in Ref. [11] we predicted that it is inadmissible to reduce the ratio to less than $D_{\min} \approx 0.06$ with regard to a seat pneumatic suspension containing the SCM if there is no active position control.

On the other hand, a VIM with minimum stiffness must be stabilized in a transient to extend the time of shock-free motion. This is especially important if the reference height of the VIM is close to the bottom or top elastic end-stop. Then damping has to be gained many times. In this regard, the elastic properties of a pneumatic controlled suspension over-softened with the SCM and derestricted to the fluid damper provides a unique chance to use air-spring as a controlled air-damper along with its load-carrying capacity. Approximately, the air-damping force range might be evaluated as

$$F_{a-d} = i_1^{-\alpha_n} A \Delta p, \tag{3}$$

where Δp is the pressure difference in the air-spring and a vehicle network line to the seat suspension or atmospheric one. In consideration of i_1- and A-ranges, and a vehicle (heavy truck, bus etc.) network power [5–7,15,16], the difference can reach at least $\Delta p = 2-4\,\mathrm{bar}$. This allows generating the force F_{a-d} which is enough to reach even the critical value, i.e., $D = 1$.

3. Equations formulation

To demonstrate the control idea, e.g., with regard to the redesigned suspension (see Fig. 1), let us consider a single-degrees-of-freedom (DOF) dynamic model. It includes the equation of motion of a mechanical system under kinematic vibration or impulsive excitation, equation of air balance in the control valves responding to changes in the elastic and damping properties of the suspension, and the relationship among the thermodynamic parameters of the air-spring and vehicle network:

$$m\ddot{z}_2 = \frac{(p-p_a)A}{i_1^{\alpha_n}} - \tilde{k}_2\mu_k\mu_l i_2(z_1 - z_2) + b_2(\dot{z}_1 - \dot{z}_2) - mg,$$
$$\dot{p} = \frac{\gamma}{V}\left[R(G_{\rm in}T_{\rm in} - G_{\rm ex}T) - p\dot{V} - (T - T_a)k_T A^*\right],$$
$$\dot{T} = \frac{T}{pV}\left[G_{\rm in}R(\gamma T_{\rm in} - T) - G_{\rm ex}RT(\gamma-1) - (\gamma-1)p\dot{V} - (T - T_a)k_T A^*,\right. \tag{4a−4c}$$

where m is the mass of the seat sprung part including the occupant; z_1, z_2, $\dot{z}_1$, $\dot{z}_2$, and $\ddot{z}_2$ are the position, velocity, and acceleration, respectively, of the part and the mount excited in their relative motion; g is the free fall acceleration. The excitation law z_1 is entered in the form of the infra-frequency vibrations or impulses. Light damping b_2 is entered into the model as viscous because contact surfaces of kinematic pairs in the guide and the connecting mechanisms are protected with the fluorine-containing antifriction films [7,9]. Motion of working substance depends on the pressure and temperature velocities in the air-spring. The values $G_{\rm in}$ and $G_{\rm ex}$ of inlet and outlet air consumption are defined from the St.-Venant function [17] that in this case are

$$G_{\rm in} = \left(\sum_{j=1}^{N} a_{{\rm in}(j)}\right) p_{\rm in}\sigma_{\rm in}\sqrt{\frac{2\gamma}{RT_{\rm in}(\gamma-1)}}; \quad G_{\rm ex} = \left(\sum_{j=1}^{N} a_{{\rm ex}(j)}\right) p\sigma_{\rm ex}\sqrt{\frac{2\gamma}{RT(\gamma-1)}}, \tag{5a,5b}$$

where the inlet and outlet characteristics are the following:

$$\sigma_{\rm in} = \sqrt{\left(\frac{p}{p_{\rm in}}\right)^{2/\gamma} - \left(\frac{p}{p_{\rm in}}\right)^{(\gamma+1)/\gamma}} \quad \text{or} \quad \sigma_{\rm in} = \sqrt{\left(\frac{2}{\gamma+1}\right)^{2/(\gamma-1)} - \left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \tag{6a}$$

$$\text{if } \frac{p}{p_{\rm in}} > \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \text{ or } \frac{p}{p_{\rm in}} < \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)}, \text{ respectively.}$$

$$\sigma_{\rm ex} = \sqrt{\left(\frac{p_a}{p}\right)^{2/\gamma} - \left(\frac{p_a}{p}\right)^{(\gamma+1)/\gamma}} \quad \text{or} \quad \sigma_{\rm ex} = \sqrt{\left(\frac{2}{\gamma+1}\right)^{2/(\gamma-1)} - \left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \tag{6b}$$

$$\text{if } \frac{p_a}{p} > \left(\frac{2}{\gamma+1}\right)^{\gamma/\gamma-1} \text{ or } \frac{p_a}{p} < \left(\frac{2}{\gamma+1}\right)^{\gamma/\gamma-1}, \text{respectively.}$$

In Eqs. (4)–(6), p_a, $p_{\rm in}$, $T_{\rm in}$ are the atmospheric pressure, the pressure and temperature in the network; R, T are the gas constant and absolute temperature in the air-spring; γ, k_T, Ta are the adiabatic exponent, heat-transfer factor, and ambient temperature; V and A^* are the current volume and heat-transfer area of the air-spring; $a_{{\rm in}(j)}$, $a_{{\rm ex}(j)}$ are the effective areas of the valve inlet and outlet orifices; N is a number of control valves, respectively.

4. Simulated results. Control criteria

Solution of Eqs. (4)–(6) yields a set of criteria for control of the suspension with minimum stiffness in steady-state relative motion and in a transient. Results obtained by the computational algorithm are then compared to the responses of a full-scale prototype of the suspension.

4.1. A range of stiffness minima

Using methods of dimensional and performance syntheses of springs with "negative" stiffness [7,9,10], one can design a compact SCM that enables minimizing the stiffness, e.g., of a standard seat pneumatic suspension in complete range of height control (see Fig. 2a). Then one must obtain an optimum of $|-k_2|$ in complete range of payload capacity. Otherwise, there exists a hypothetical opportunity for unstable motion on a certain section of stroke z_0 if $|-k_2| > k_1$ (see shaded area in Fig. 2b). At the same time, these magnitudes $|-k_2|$ can

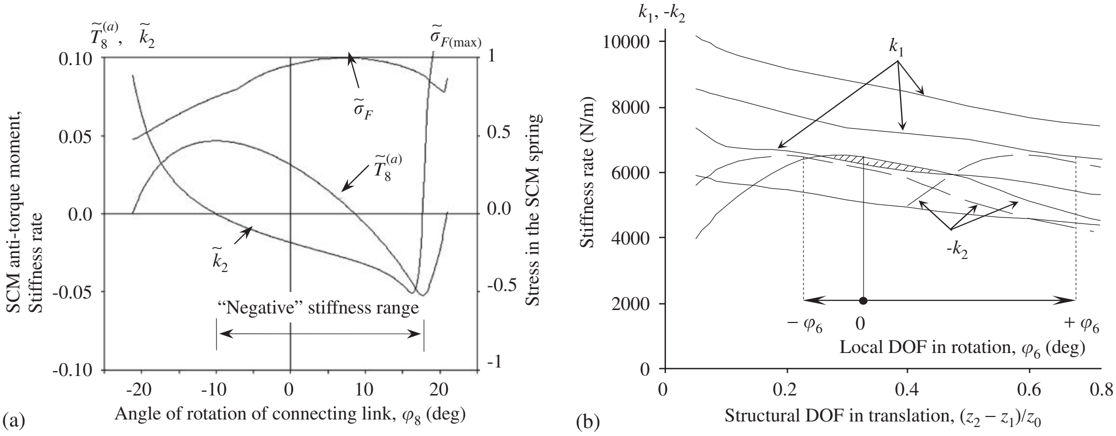

Fig. 2. Stiffness adjustment: (a) "negative" stiffness range of SCM for a seat pneumatic suspension; (b) ranging of an optimal stiffness ratio between spring of SCM and air-spring.

appear insufficient under maximal payloads. To optimize $|-k_2|$-range one may enter the local DOF for the SCM. In practice, this option consists, e.g., in discrete (or indefinite) rotation of the SCM frame 6 relative to mount 1 by an angle φ_6 in pre-adjusting (see Fig. 1). Such a shift of the SCM operation point is feasible because the $|-k_2|$-range is large. For instance, Fig. 2a shows that $\varphi_8 \geqslant 27^\circ$ meets the stroke $z_0 \approx 129-137$ mm in view of design of the suspensions considered [2,5–7,9,14–16]. Thus, a minimal but "positive" stiffness, $k = \rightarrow +0$, is an absolute criterion of a VIM position control in steady-state motion under operation of the SCM tending to chaotic motion.

4.2. *Variable structure of air-damping control and stability criteria in a transient*

Assuming that the velocity of the suspension relative to mount excited is constant, the equation of continuity at, e.g., inlet phase is

$$G_{\mathrm{in}} = A_0 \cdot \rho \cdot i_1^{-\alpha_n}(\dot{z}_2 - \dot{z}_1), \tag{7}$$

where A_0 is an initial value of the area A; ρ is air pressure in the wind chamber defined from equation of state, $\rho = \rho_0(p/p_{\mathrm{in}})^{\gamma^{-1}}$ in subcritical airflow, here ρ_0 is air pressure in the network.

From Eq. (7) in view of Eqs. (5) and (6) one may express a relation (a_{in}/A_0). In consideration of the pressure ratio (p/p_{in}) and payload capacity, one can obtain approximate values of second control criterion for the VIM with minimum stiffness:

$$a_{\mathrm{in}} \approx 10^{-4} A_0. \tag{8}$$

These values are optimized in terms of the particular valve parameters, relative velocity, and response time.

Simulation of motion for a vehicle driver shows that the object may remain motionless if the relative velocity is around $\dot{z}_2 - \dot{z}_1 < 5 \times 10^{-2}$ m/s. In these cases, the control system must only prevent the deviation amplitude out of a ξ-"corridor" determined for a reference height. This is a well-known AC qualitative concept [18] illustrated in Fig. 3a.

However, the AC concept becomes ineffectual under rising velocities. Fig. 3b shows that the deviation amplitude increased during a short period under impulsive excitation. As seen, the response is weak-damped. Evidently, a re-impulse may increase the amplitude resulting in a suspension impact against the end-stop. Thus, there is no stabilization by using position feedback containing only the control valves with the small values of $a_{\mathrm{in}(j)}$ and $a_{\mathrm{ex}(j)}$ as in the standard seat pneumatic controlled suspensions (see Eq. (8)).

Therefore, from the above model analysis, the large values of $a_{\mathrm{in}(j)}$ and $a_{\mathrm{ex}(j)}$ that ensure stabilization of the VIM with minimum stiffness in a transient have also been defined. Fig. 3c shows that such drastic air-damping results in stabilizing the suspension immediately. And continuity of command in the AC-mode enables

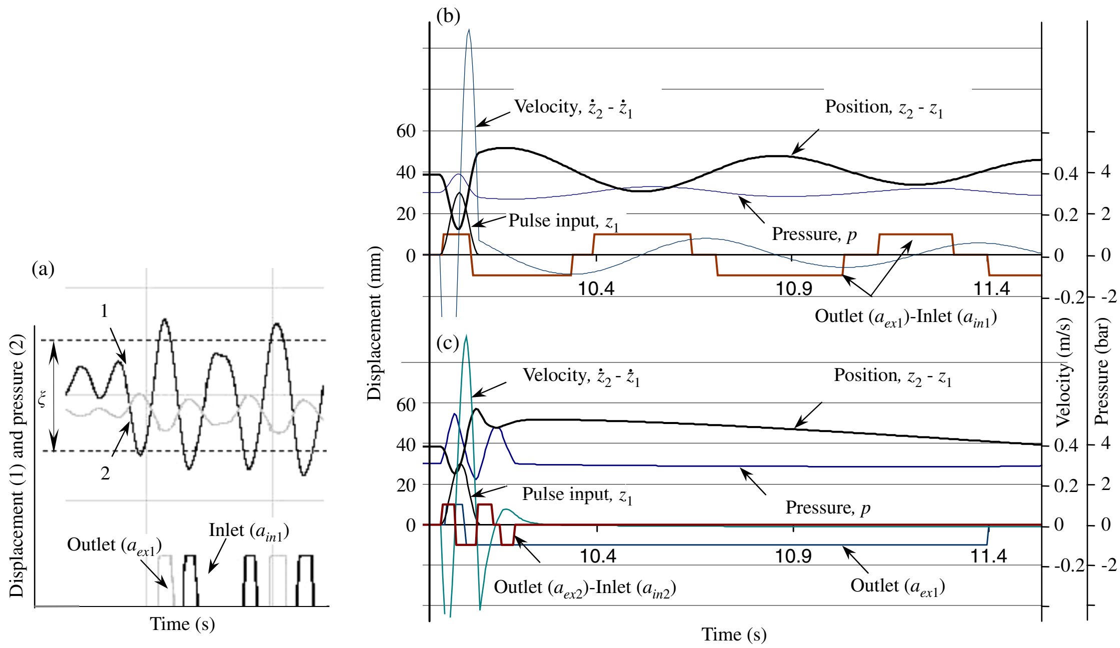

Fig. 3. Methods of position control: (a) AC concept for a VIM, in steady-state vibration motion; (b) and (c) simulation analysis of position controllability of a seat pneumatic suspension with minimum stiffness and light air-damping or variable structure of air-damping, respectively, in a transient.

detecting position errors due to drastic air inlet or outlet. Thus, one may express the control criteria, in addition

(a) To stabilize a seat pneumatic suspension with minimum stiffness one can organize a pneumatic control system containing at least two control valves operating in parallel, i.e., $N \geqslant 2$ ($j = 1, 2 \ldots$).
(b) Approximate ratio for the orifice effective areas in these control valves is to be around

$$\frac{\{a_{\text{in2}}\}, \{a_{\text{ex2}}\}}{\{a_{\text{in1}}\}, \{a_{\text{ex1}}\}} \succ 20 - 30. \tag{9}$$

5. Measurement

5.1. Organization of the pneumatic control system

Electro-pneumatic valves ensure the position control of a pneumatic VIM with minimum stiffness. In practice, one can use commercial off-the-shelf pneumatic components. By specification, in consideration of control problems and cost criteria, the pneumatic control system (see a circuit in Fig. 4a) may include two 3/3-way control valves [19]. And due to a relatively small number of controlled parameters one could also use a simple linear microcontroller [20].

5.2. An experimental rig

The approach of position control was approved in the development of the seat pneumatic controlled suspension for a heavy truck driver (see Fig. 4b). In the development process, initial arrangement of the suspension was gradually restructured: (a) fluid damper was removed; (b) the SCM containing spring with adjustable "negative" stiffness was connected to the input link of the guide mechanism by using the bevel gear.

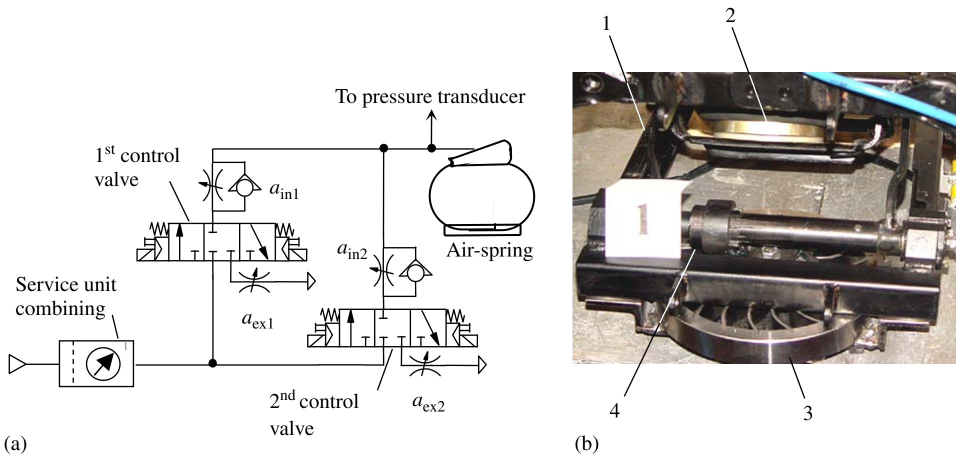

Fig. 4. Measurement instrumentation: (a) circuit of pneumatic control system; (b) layout of seat pneumatic suspension containing the SCM, where are the guide mechanism (position 1), air-spring (2), SCM (3), connecting mechanism (4).

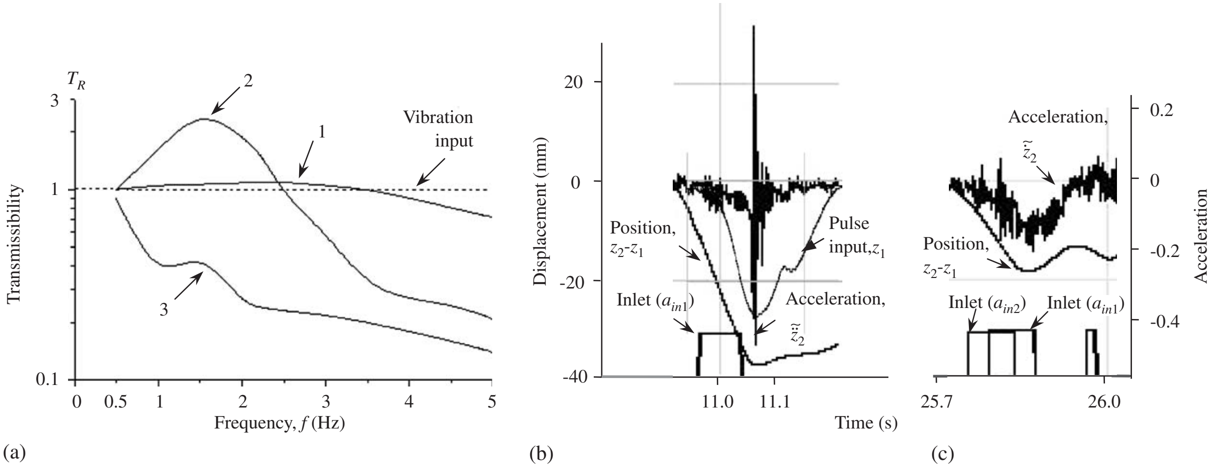

Fig. 5. Performance evaluation in improvement of seat pneumatic controlled suspension: (a) vibration isolation attainable with initial (standard) arrangement (1), if no fluid damper (2), after connecting the SCM; (b) and (c) accelerative forces attenuation by using the suspension with minimum stiffness and light air-damping or variable structure of air-damping, respectively.

In addition to standard test facility [21] representative in the frequency range $f \geqslant 1$ Hz, a pneumatic shaker was designed to simulate vibrations and impulsive excitations in the infra-narrow band, $f = 0.25-1.25$ Hz.

5.3. Estimating vibrations and impulsive responses

Under sinusoidal vibrations in the infra-frequency range, the input signal is $L_a \approx 60$ dB if $f \in [0.5; 1.2)$ Hz and $L_a \approx 70$ dB if $f \in [1; 5]$ Hz. Fig. 5a demonstrates evident progress in improving vibration isolation. The seat pneumatic suspension with minimum stiffness, in comparison with the initial, is non-resonant in the whole frequency range and reduces input vibration by at least $\tilde{T}_R(f \geqslant 1\,\text{Hz}) > 300 - 500\%$.

Figs. 5b and c show the behavior of the suspension under impulsive excitation. In the sample, the displacement amplitude of excitation is 29–30 mm. The pulse edge length is about of 0.1 s. In air-damping control by orifices with small effective areas (in operating of 1st control valve only), the pulse caused collision of the suspension against the bottom elastic end-stop resulting in accelerative forces (see Fig. 5b). At the same time, drastic air inlet–outlet by orifices with large effective areas (in operating of 2nd control valve) prevented these collisions. In addition, the work of the 1st control valve minimized position error due to drastic air inlet–outlet. Finally, this led to shock-free motion and a reduction in the amplitude of accelerations (see Fig. 5c).

6. Conclusion

The paper proposes an approach of position control for seat suspensions with a focus on the vibration isolation of vehicle drivers in the infra-frequency range within which the standard controlled devices are ineffective in normal gravitation. The approach is based on stiffness minimization by coupling a mechanism containing the spring with adjustable "negative" stiffness in the large and organization of variable structure of air-damping control. A control algorithm corresponding to this approach can be used with a multi-channel pneumatic control system designed with simple commercial off-the-shelf pneumatic components, if high-precise positioning is not required. A test of the approach indicates that a standard seat pneumatic controlled suspension re-designed by coupling the above mechanism, removing the fluid damper, and organizing the variable structure of air-damping ensured the following for the driver: (a) an immobility under vibrations beginning with infra-frequencies close to zero Hz; (b) stabilization and shock-free motion under impulsive excitation.

Acknowledgments

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References

[1] M.J. Griffin, *Handbook of Human Vibration*, Academic Press Ltd., London, 1990.
[2] A.V. Andreytchikov, O.N. Andreytchikov, An intelligent decision making system under a uncertainty, *Journal of Problems in Engineering and Reliability of Machines* 4 (2002) 3–10 (in Russia).
[3] S.K. Majumdar, *Pneumatic Systems: Principles and Maintenance*, McGraw-Hill, New York, 1995.
[4] K.V. Frolov, F.A. Furman, *Applied Theory of Vibration Protecting Systems*, Taylor & Francis Group, New York, 1989.
[5] O. Krejcir, *Pneumaticka Vibroizolace, Doctorska disertacna prace*, Liberec, Czech Republic, 1986.
[6] http://www.motion-master.com/.
[7] C.-M. Lee, V.N. Goverdovskiy, Alternative vibration protecting systems for men-operators of transport machines: modern level and prospects, *Journal of Sound and Vibration* 249 (2002) 635–647.
[8] P.M. Alabuzhev, et al., *Vibration Protecting and Measuring Systems with Quasi-Zero Stiffness*, Taylor & Francis Group, New York, 1989.
[9] C.-M. Lee, V.N. Goverdovskiy, Method of stiffness control of vibration isolating suspension of a compact seat for a vehicle driver and the suspension to put the method into practice, RU Patent No. 2,214,335, 2003.
[10] V.N. Goverdovskiy et al., Stiffness control mechanism for a compact seat suspension, RU Patent No. 2,216,461, 2003.
[11] C.-M. Lee, V.N. Goverdovskiy, S.B. Samoilenko, Prediction of non-chaotic motion of the elastic system with small stiffness, *Journal of Sound and Vibration* 272 (2004) 643–655.
[12] R. Herzog, Active versus passive vibration absorbers, *Guidance, Control and Dynamics* 117 (1995) 116–122.
[13] J.Q. Sun, M.R. Jolly, M.A. Norris, Passive, adaptive and active tuned vibration absorbers: a survey. Special Fiftieth Anniversary Design Issue, *Transactions of ASME* 117 (1995) 234–242.
[14] S.J. McManus, et al., Evaluation of vibration and shock attenuation performance of a suspension seat with a semi-active magnetorheological fluid damper, *Journal of Sound and Vibration* 253 (2002) 313–327.
[15] Pneumatic controlled suspension for automotive seat, JP Patent No. 2,855,479, 1990.
[16] Controlled suspension of a seat, GE Patent No. 19,848,603, 1998.
[17] H.V. Hertz (Ed.), *Pneumatic Drives Analysis*, second ed., Engineering, Moscow, 1987 (in Russia).
[18] N.V. Joshi, P. Murugan, R.R. Rhineahrt, Experimental comparison of control strategies, *Journal of Sound and Vibration* 5 (1997) 885–896.
[19] W. Deppert, K. Stoll, *Pneumatic Control*, English edition, Vogel Buchverlag, Wurzburg, 1987.
[20] http://www.microcontroller.com/.
[21] ISO 8041, Human response to vibration: measuring instrumentation, 1990.