DETERMINATION OF VIBRATIONS TRANSMISSIBILITY CHARACTERISTICS OF AIR SUSPENSION SYSTEM FOR VEHICLE RIDE COMFORT AND SAFETY ASSESSMENT

Ryszard Dindorf*, Jakub Takosoglu

Kielce University of Technology, Faculty of Mechatronics and Mechanical Engineering, Department of Mechatronics and Armaund, Kielce, Poland

*E-mail of corresponding author: dindorf@tu.kielce.pl

Ryszard Dindorf 0000-0002-2242-3288, Jakub Takosoglu 0000-0003-0606-039X

Resume

The purpose of this study was to determine the vibrations transmissibility characteristics of an air suspension system (ASS) with self-damping air bellows to assess the comfort and safety of the ride. The conceptual design of a pneumatic self-damping air bellow is presented, which involves forced airflow through a damping orifice between the air bellow and an auxiliary reservoir. Two dynamic models, classical and simple, of the analyzed air bellow were used for numerical modelling solutions. Vibrations transmissibility charts as a function of the frequency ratio and at different damping ratios were determined using the accepted excitation forces. The different vibrations transmissibility states and dynamic characteristics of the air bellow for the frequency ratios and individual damping (inherent, pneumatic self-damping, and hydraulic absorber) in the ASS range were examined.

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1 Introduction

Vibrations in vehicles significantly reduce driving comfort and safety, reduce driver concentration, and cause discomfort for passengers. Depending on their frequency, amplitude, and duration, vibrations have a direct impact on the health, motion sickness, performance, and safety of drivers and passengers, potentially leading to accidents. Vehicle vibrations are caused by irregularities on the road surface, wheel imbalance, suspension conditions, and engine and powertrain operation [1]. Vehicle vibrations, especially low frequency, can cause physiological effects such as drowsiness, fatigue, and loss of alertness and affect the body's organs, leading to discomfort or even pain. For example, lower frequencies (up to 4 Hz) affect the chest, abdomen, and stomach, while the higher frequencies (above 10 Hz) affect the head and eyes. Vehicle vibrations also have psychological effects, reducing driver concentration and performance and increasing the risk of accidents, particularly during long drives. The sensitivity of drivers and passengers to vibrations may vary depending on individual factors. The human body cannot perceive or feel all vibrations equally. Human sensitivity to vibrations is covered by the ISO 2631-1 standard [2], which generally evaluates vibrations in the frequency range of 0.5 to 80 Hz and focuses on the whole-body vibrations (WBV), which is transmitted through supporting surfaces, primarily the seat (thigs and backrest) and floor (feet). The human body has "resonant frequencies" at which specific organs or skeletal structures vibrate more intensely [3]. The standard defines a Health Guidance Caution Zone (HGCZ). If the daily exposure of a passenger falls within or above this zone, there is an increased risk of long-term health problems, mainly degeneration of the lumbar spine, herniated discs, lower back pain, fatigue, and decreased reaction time. Vehicle manufacturers use ISO 2631-1 to design and tune suspension systems to ensure that the natural frequency of the vehicle is maintained away from human resonance. Seats are designed to act as secondary damping, filtering out frequencies to which the human body is most sensitive [4-5].

***Table 1** Comparison of vehicle ride comfort and safety of ASS with MS and HPS*

Feature	*MS*	*ASS*	*HPS*
Ride comfort	Moderate poor	Excellent	Superior
Safety handling	Average	High predictive	Extreme active
Reliability	Highest	Moderate	Moderate low
Payload capacity	Fixed	Self-adjusting	Self-adjusting
Cost	Low	High	Very high

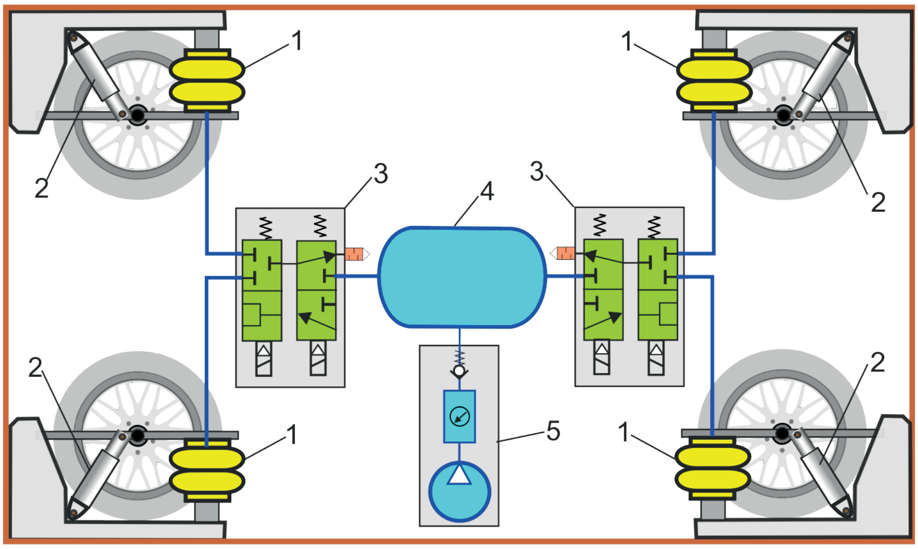

***Figure 1** Schematic diagram of the pneumatic circuit for the ASS control in a two-axle vehicle: 1 - air springs (air bellows), 2 - shock absorbers (dampers), 3 - control valve blocks, 4 -compressed air reservoir, 5 - compressed air supplies*

The ASSs offer the best balance of plush ride comfort and safety, ensure adjustable height and modern safety technology. In Table 1 is given the comparison between the ASS with mechanical suspension (MS) and hydropneumatic suspension (HPS) in terms of comfort and safety of the vehicle ride [6].

Vibrations transmissibility affects the comfort and handling of a vehicle with air suspension, which considers the following: comfortable ride, air springs better absorb road imperfections, reducing noise, vibrations, and harshness (NVH) for a smoother ride, particularly on long trips; reduced body roll, electronic controls counteract body lean during cornering and sudden maneuvers to improve stability; auto-leveling control system maintains a consistent vehicle height regardless of the passenger or cargo load, preventing squatting or sagging. The vibrations transmissibility level affects the safety ride, stability, load control, and road conditions, ensuring optimal tire contact and thereby improving grip and handling. Ensuring optimal aerodynamics by lowering the vehicle at higher speeds, improving its fuel efficiency and stability, improving braking and traction, and ensuring consistent tire contact improves brake performance and overall road holding; consistent safe load distribution prevents tire wear and cargo shifting [7-9].

2 The ASS and ride comfort and safety

The ASS is an advanced vehicle suspension consisting of the following elements: air spring, shock absorber, pneumatic equipment (compressor, reservoir, block valves, fittings, and pipes), electronic control unit (ECU), and sensors [10]. Figure 1 shows a schematic diagram of the pneumatic circuit for the ASS control in a two-axle vehicle [11]. The main load-bearing components of the ASS are air springs and shock absorbers. The air spring works in conjunction with the shock absorber to provide ride stability. While the air spring provides a cushion, supports the weight, and sets the vehicle height, the shock absorber (damper) dampens the movement, stops the bounce, roll, and oscillation of the vehicle body, and ensures constant wheel traction. In many modern luxury vehicles, shock absorbers and air springs are not separate parts; instead, they are combined into a single unit, called an air strut. No known commercial vehicles use suspension systems that are exclusively based on air springs without shock absorbers or self-damping air springs. In a self-damped air spring (without an external damper), the damping is significantly lower because it results from the pneumatic damping. All the available sources indicate that the air spring must be paired with a shock absorber.

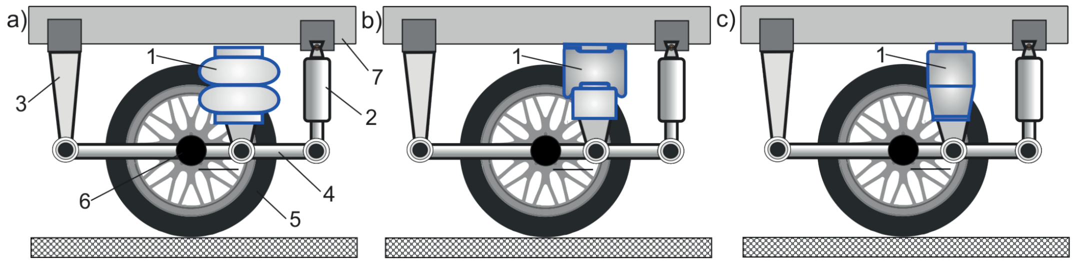

Figure 2 *Diagrams of the basic ASS with various types air spring: (a) double convoluted, (b) rolling lobe and (c) tapered sleeves: 1 - air spring, 2 - damper, 3 - rotation arm hinge bracket, 4 - twin rotation arm, 5 - wheel, 6 - wheel hub, 7 - vehicle body*

The ASS uses air springs (rubber cushions filled with compressed air) instead of traditional mechanical elements. The air springs are flexible and airtight components consisting of reinforced rubber and fabric bellows that use compressed air to support vehicle loads, while providing effective shock and vibrations isolation [12]. Various air springs are used in vehicles, adopted to vehicle performance, driving comfort, and safety requirements. Figure 2 shows the basic air suspension with various types of air springs. The three types of air springs were distinguished:

Convoluted springs (air bellow): Available with one, two, or three convolutions, this type is the most widely used in heavy-duty trucks, trailers, and rally cars. Their multi-section bellow design offers robust load-bearing capability and effective vibrations damping.

Rolling lobe springs: These are constructed with a flexible rubber sleeve and a piston at the bottom, allowing the sleeve to roll over the piston as the spring is compressed. This design offers a longer stroke and superior ride comfort, making it popular in luxury cars and commercial buses with air suspensions.

Sleeve springs: With a slender cylindrical design and flexible cover, these springs are ideal for passenger vehicles and light trucks, as they are compact enough to fit in confined areas and ensure a comfortable ride. They are frequently used as isolators in the suspension systems of cabs and seats in trucks and buses.

The key air suspension parameters affect the comfort and safety of the ride [13-15].

1. The stiffness of the air springs depends on pressure. The pressure control determines the softness or hardness of an air spring. Excessive softness of the air spring at low pressure can lead to excessive body roll, sway, and poor handling, particularly in turns, thus reducing control over the vehicle and causing sudden loss of steering control or permanent damage to the suspension mounts and frame. High air spring stiffness at high pressure can make the ride harsh, causing driver fatigue and potentially reducing tire contact on uneven surfaces, thus increasing the braking distance and the risk of losing control on uneven surfaces.
2. The stiffness of the air springs can be adjusted according to the degree of air compression. A critical safety feature is the ability to increase the stiffness as the load increases. This keeps the vehicle's natural frequency constant, which prevents the vehicle from feeling sluggish or difficult to recover from a sway when heavily loaded. By adjusting the spring rates, the air suspension can minimize the nose dive during emergency braking. This maintains a weight distribution even across the four tires, allowing the ABS to operate more efficiently and reducing the total stopping distance.
3. The damping coefficient of air suspension, with inadequate damping caused by a low damping coefficient, can cause excessive bouncing, wallowing, or excessive oscillation of the vehicle body, where the grip of the tire varies significantly, thus compromising safety on wet or loose terrains. A high rubbing coefficient can deteriorate ride stability and comfort.
4. The height of the air suspension affects the posture of a vehicle and the ground clearance. An incorrect height air suspension affects the ride stability, particularly during cornering, and can interfere with suspension travel, thereby affecting the wheel-road contact. An air suspension height that is set too high raises the vehicle center of gravity (CoG), directly increasing the risk of rollover during cornering or evasive maneuvers.
5. Air suspension sensors and control systems are essential for the proper functioning of air suspensions. Inaccurate sensors (load, height, and accelerometers) lead to incorrect adjustments, compromising stability and ride control. Sensors are used in an Electronically Controlled Air Suspension System (ECAS) to improve ride comfort and safety [16]. The ECAS automatically manages the lifting or lowering of the vehicle body based on the current weight of the vehicle. The ECAS monitors road irregularities and adjusts vibrations damping to minimize vibrations and jerks, resulting in a smoother ride for both cargo and passengers. The integrated ECAS with the service and maintenance system monitors pressure, calculates the current axle load to prevent overloading, and ensures that the weight is distributed according to the legal limits.

The main purpose of this study was to determine and analyze the vibrations transmissibility characteristics of an ASS equipped with a self-damping air bellow and to assess its influence on vehicle ride comfort and driving safety.

The purpose of this study was to:

1. Develop the ASS dynamic models with a self-damping air bellow.
2. Derive analytical expressions for the vibrations' transmissibility as a function of the frequency ratio and the damping ratio.
3. Compare different damping mechanisms (inherent air bellow damping, pneumatic self-damping, and hydraulic shock absorber damping).
4. Identify the operating conditions under which a self-damping air bellow can replace or complement conventional dampers.
5. Design-oriented conclusions are provided for the selection and adaptation of ASSs to meet the comfort and safety requirements of vehicles.

3 The ASS with a self-damping air bellow

In an air suspension system, the shock absorber (damper) is almost always a hydraulic or gas-charged unit, similar to those in standard vehicles. However, they come in different types, depending on their construction and integration with the air bellows. There are two basic types of shock absorbers: passive shock absorbers with constant firmness and commonly used adaptive (electronic) shock absorbers with continuous damping control (CDC) that adjust firmness. In modern ASS, dampers typically operate using a combination of fluid dynamics and electronic management systems. Dynamic chassis control (DCC) is a Volkswagen system that uses an ECU to modify the damper flow valves and damping force based on road conditions, steering, braking, and acceleration. Popular adaptive suspension systems include MagneRide (Ford), Adaptive Dynamics (Jaguar), Porsche Active Suspension Management (PASM) (Porsche), Adaptive Variable Suspension (AVS) (Lexus) and Four-C Chassis Control (Volvo).

The ASS also uses self-damping air bellows with pneumatic damping, excluding external shock absorbers. Self-damping air bellows, often referred to as integrated air bellows, are used in vehicles that require superior ride comfort, load leveling, and reduced maintenance, with their primary applications in commercial, railway, truck vehicles, trailer, and tractors. The self-damping air spring that features Hendrickson's patented Zero Maintenance Damping (ZMD ®) technology eliminates the need for traditional shock absorbers by integrating the damping function directly into the air spring [17-18]. The ASS Hendrickson technology is primarily used in trailers, trucks, highway tractors, vocational trucks such as dump trucks, mixers, refuse haulers, and fire and rescue vehicles. ASSs, such as VANTRAAX and NTRAAX®, are used in trailers, and ROADMAAX® Z is used in the drive axes of heavy-duty trucks and tractor rears. The self-damping air springs can be installed in parallel with shock absorbers in vehicles with air suspensions. This is applicable to vehicles with factory installed air suspension, such as Mercedes-Benz, BMW, Audi, Lexus, Porsche, Tesla, Bentley, Jaguar and Maserati, and SUVs such as Land Rover, Volvo XC 90, and Ford Raptor.

A self-damping air bellow is often referred to as a low-frequency damping air bellow. From a technical and design perspective, the self-damped air bellow is referred to as a two-chamber and orifice damping [19]. The development of air suspensions with self-damping air bellows has become more aligned with the requirements of new electric vehicles in terms of handling, comfort, and safety. Although the self-damped air bellow eliminates the need for shock absorbers in suspensions and vibrations isolation systems, it can provide variable stiffness and damping. Air suspension can be soft for cruising or firm for turning. When assessing self-damped air bellows, vibrations transmissibility is considered. Vibrations transmissibility is a crucial indicator for evaluating the performance of air suspension systems. Vibrations transmissibility is determined by the influence of wheel vibrations on the air suspension and chassis. The vibrations transmissibility characteristics is the basis for directly indicating the effectiveness of self-damping air bellows.

In this study, a conceptual solution of self-damping air bellow with a damping orifice and auxiliary reservoir is considered, the schematic of which is shown in Figure 3.

The concept of a self-damping air bellow uses the forced flow of air through a specifically designed damping orifice (restriction) between an air bellow and

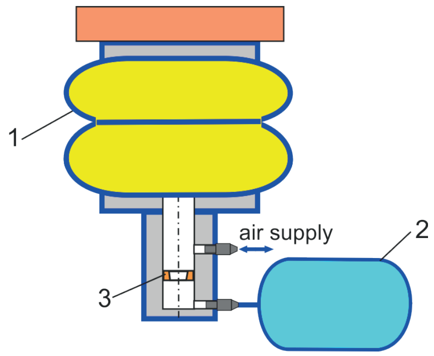

Figure 3 *Diagrams of the conceptual solution of self-damping air bellow with a damping orifice and auxiliary reservoir: 1 - air bellow, 2 - auxiliary reservoir, 3 - damping orifice*

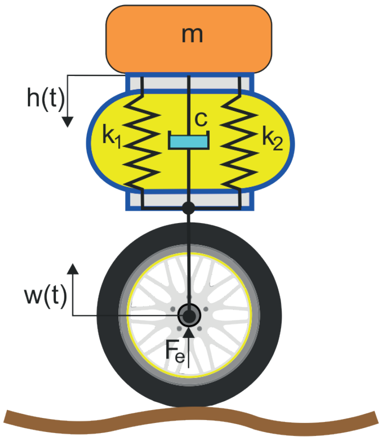

Figure 4 *Diagrams of the dynamic model of the air bellow: m is the sprung mass, h is the deflection of the air bellows, k_1 is the stiffness of the air bellow due to compression and expansion of the air, k_2 is the stiffness of the air bellow due to its deformation, c is the damping coefficient of the self-damped air bellow, F_e is the excitation force, w is the vertical displacement of a wheel*

an auxiliary reservoir to dissipate vibrational energy and heat exchange between the air and surroundings during air compression and expansion when the air suspension is subjected to vibrations. As the axle moves upward (absorbed), compressed air is forced from the air bellow into the auxiliary reservoir. As the axle moves downward (rebound), the compressed air returns from the auxiliary reservoir to the air bellow. The shape of the damping orifice significantly affects the damping coefficient, which depends on the perimeter-to-area ratio of the orifice. The effectiveness of the damping in the orifice is also dependent on the pressure differential between the air bellow and auxiliary reservoir chambers. In the advanced solution of self-damping air bellow, solenoid-controlled variable-area damping orifices are used, which can adapt the suspension stiffness and damping to varying loads and road conditions. By controlling the airflow in the damping orifice, compact and effective vibrations isolation can be achieved in the road and rail vehicle suspensions.

The inherent damping of air bellows, which refers to the natural damping effect produced by the air bellow, owing to the elastic structure of the reinforced elastomeric materials (rubber + fabric), is considered. The deformation of the rubber layers generates internal friction in the polymer and cord layers and hysteresis, which is the inner damping between 2% and 5% of the total damping. The ASS regulations and safety requirements render the standalone air bellow inherent damping insufficient. Inherent-damping air bellows are often incorrectly identified as self-damping air bellows.

4 Dynamic models of the air bellow

A diagram of the dynamic model of the air bellow is shown in Figure 4.

Based on the diagram of the dynamic model of the air bellow shown in Figure 4, a classical dynamic model of an undamped air bellow and a simple dynamic model of a self-damped air bellow are presented. The classical dynamic model of an air bellow was created using a mass-spring system with sprung mass and excitation force, without considering unsprung mass, tire stiffness, and road excitation. A simple dynamic model of an air bellow with self-damping was created using a mass-spring-damper system with sprung mass, excitation force, and road excitation, without considering the unsprung mass and tire stiffness.

4.1 Classical dynamic model of undamped air bellow

The classical dynamic model of an undamped air bellow under a fixed excitation force F_e is described as follows:

$$m\ddot{h} + (k_1 + k_2)h = m\ddot{h} + kh = F_e \Rightarrow \\ \Rightarrow \ddot{h} + \omega_n^2 h = \frac{F_e}{m}, \tag{1}$$

where m is the sprung mass of a quarter of vehicle body, h is the deflection of the air bellows, the change in the height/stroke of the air bellows, F_e is the structural fixed excitation force generated by the wheel - suspension - vehicle body interaction. k is the total stiffness of the air bellow, k_1 is the stiffness of the air bellow due to compression and expansion of the air, k_2 is the stiffness of the air bellow due to its deformation, ω_n is the natural angular frequency of the air bellow,

$$\omega_n = \sqrt{\frac{k}{m}}. \tag{2}$$

The air bellow has a following natural frequency:

$$f_n = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k}{m}}. \tag{3}$$

The stiffness of the air bellow for the classical dynamic model is determined as follows:

$$k = \frac{F}{dh} = \frac{d(p,A_{ef})}{dh} = A_e\frac{dp}{dh} + p\frac{dA_{ef}}{dh} = k_1 + k_2\,, \quad (4)$$

where p is the instantaneous pressure inside the air bellow,

A_{ef} is the effective area of the air bellow,

$$A_e = \frac{\pi D_{ef}^2}{4}\,, \quad (5)$$

where D_{ef} is the effective cross-sectional area,

k_1 is the stiffness of the air bellow due to compression and expansion of the air,

$$k_1 = A_{ef}\frac{dp}{dh} = A_{ef}^2\frac{dp}{dV}\,, \quad (6)$$

k_2 is the stiffness of the air bellow due to its deformation,

$$k_2 = p\frac{dA_{ef}}{dh}\,. \quad (7)$$

For the air compression polytropic process, the pressure and pressure derivative in the air bellow are as follows:

$$pV^n = p_H V_H^n \;\Rightarrow\; p = p_H\left(\frac{V_H}{V}\right)^n \;\Rightarrow$$
$$\Rightarrow\; \frac{dp}{dV} = \kappa p_H\frac{V_H^n}{V^{n+1}}, \quad (8)$$

where V is the instantaneous volume of the air bellow, p_H and V_H are the initial pressure and the initial volume of the air bellow, respectively, and n is the polytropic exponent.

At high frequencies (> 3 Hz), an adiabatic process with adiabatic exponent $n = \kappa = 1.4$ is justified since the air is compressed too quickly for the heat exchange and the air bellow becomes significantly stiffer [20]. At very low frequencies (< 0.1 Hz), air compression is close to the isothermal process with $n = 1$, heat has time to dissipate through the walls of the air bellow, and the air bellow is softer.

The total stiffness $k(h)$ of the air bellow, after considering Equations (5), (6), and (7), as well as the geometrical relationships $V = V_H - \Delta V$, $V_H \approx A_e H$, $\Delta V \approx A_{ef} h$, were determined as a function of the stroke h.

$$k(h) = k_1 + k_2 = np_H A_{ef}^2\frac{V_H^n}{(V_H - A_{ef}h)^{n+1}} +$$
$$+ p_H\frac{V_H^n}{(V_H - A_{ef}h)^n}\frac{dA_{ef}}{dh} = \frac{np_H A_{ef}}{H}\left(\frac{H}{H-h}\right)^{n+1} + \quad (9)$$
$$+\left(\frac{H}{H-h}\right)^n\frac{dA_{ef}}{dh}.$$

Based on the measurement experiment [21] of a single convolution air bellow Aventics 0822419002 Type, at the initial height of $H = 100$ mm and stroke $h \in (0, 50)$ mm, the approximation function of the effective area $A_{ef}(h)$ in the function of h in the range $H \in (100, 50)$ mm is as follows.

$$A_{ef}(h) = -0.0155h^2 + 1.06h + 118\,. \quad (10)$$

Then, the derivative of the effective area $A_e(h)$ with respect to h is

$$\frac{dA_{ef}(h)}{dh} = -0.031h + 1.06\,. \quad (11)$$

A simplified static air bellow model was used to determine the initial conditions for the differential dynamic model. The static state of air bellow results from assuming in Equation (9) that the isothermal process $n = 1$ and dA_{ef}/dh is minimal for the effective area A_{ef} = constant. Then, for the nominal diameter D_n, the nominal area A_n is determined as follows:

$$A_n = \frac{\pi D_n^2}{4}\,. \quad (12)$$

The stiffness of an air bellow $k(h)$, can also be determined based on the load charts for a particular type of air bellow.

$$k(h) = p_H A_n\frac{H}{(H-h)^2}\,. \quad (13)$$

The deflection (stroke) h of the air bellow depends on the weight force W, as follows:

$$W(h) = mg = k(h)(H-h) = p_H A_n\frac{H}{(H-h)}\,, \quad (14)$$

where W is the weight force, and g is the acceleration of gravity.

For the given sprung mass m_x, the stroke h_x is expressed as follows:

$$h_x = H\left(1 - \frac{p_H A_n}{m_x g}\right). \quad (15)$$

For $h = 0$, the general static equation has the following form:

$$mg = p_H A_n\,. \quad (16)$$

4.2 Simple dynamic model of a self-damped air bellow

The simple dynamic model of a self-damped air bellow under an excitation force F_e can be described as follows:

$$m\ddot{h} + c\dot{h} + kh = F_e(t) \Rightarrow \ddot{h} + 2\zeta_s\omega_n\dot{h} +$$
$$+\omega_n^2 h = \frac{F_e(t)}{m}, \quad (17)$$

where F_e is the force excited by the wheel acting on the air bellows as an excitation sinusoidal force, expressed as follows:

$$F_e(t) = F_{e0}\sin(\omega_e t) = F_{e0}\sin(2\pi f_e t), \tag{18}$$

where F_{e0} is the amplitude of the excitation force, ω_e is the excitation angular frequency, and f_e is the excitation frequency.

$$f_e = \frac{\omega_e}{2\pi}, \tag{19}$$

c is the damping coefficient of the self-damped air bellow resulting from the pneumatic and inherent damping properties,
ζ_s is the self-damping ratio,

$$\zeta_s = \frac{c}{2m\omega_n} = \frac{c}{2\sqrt{km}}. \tag{20}$$

The damping coefficient c and the damping ratio ζ_s are determined from the equations of forced air flow through the damping orifice between the air bellow and the auxiliary reservoir. The damping coefficient c depends on the excitation frequency. At low excitation frequencies, the forced air flows slowly through the damping orifice, resulting in low levels of damping. At high excitation frequencies, airflow through the damping orifice is rapid, resulting in high damping.

Considering the excitation force in Equation (18), the dynamic Equation (17) of the air bellow with self-damping is expressed as.

$$\ddot{h} + 2\zeta_s\omega_n\dot{h} + \omega_n^2 h = a_e\sin(2\pi f_e t), \tag{21}$$

where a_e is the excitation amplitude:

$$a_e = \frac{F_{e0}}{m}. \tag{22}$$

The analytical solution of the differential Equation (21), for the forced damped vibrations of the pneumatic bellows with self-damping, is as follows.

$$\begin{aligned} h(t) &= A_h\sin(\omega_e t + \varphi) = \\ &= F_{e0}\frac{1}{\sqrt{m^2(\omega_n^2 - \omega_e^2)^2 + 4\zeta_s^2 m^2\omega_n^2\omega_{2e}}} \times, \\ &\times \sin(\omega_e t + \varphi). \end{aligned} \tag{23}$$

where φ is the phase shift response of the air bellows.

The vibrations amplitude A_h of the air bellow stroke for the forced damped vibrations transmitted from the wheel to the vehicle body is expressed as follows:

$$A_h = \frac{h_{st}}{\sqrt{\left(1 - \left(\frac{f_e}{f_n}\right)^2\right)^2 + 4\zeta_s^2\left(\frac{f_e}{f_n}\right)^2}}, \tag{24}$$

where h_{st} is the static stroke (deflection) of air bellow,

$$h_{st} = \frac{F_{e0}}{k} = \frac{a_e}{\omega_n^2}. \tag{25}$$

After considering the vertical movement of the wheels without an unsprung mass on the road surface, the dynamic equation for the simple dynamic model of a self-damped air bellow with kinematic excitation is expressed as follows:

$$\begin{aligned} &m\ddot{h} + c(\dot{h} - \dot{w}) + k(h - w) = 0 \Rightarrow \\ &\Rightarrow m\ddot{h} + c\dot{h} + kh = c\dot{w} + kw, \end{aligned} \tag{26}$$

where w and $\dot{w}$ are the vertical displacement and speed of the road excitation, respectively, determined according to the profile of the road surface.

Kinematic excitation refers to a type of dynamic input in which a dynamic model is forced into motion by a prescribed vertical displacement and velocity of the wheel (base excitation) rather than by an external excitation force applied directly to the mass.

The sinusoidal vertical displacement and velocity of the wheel on a road profile can be approximated using the following equations:

$$w = A_r\sin(2\pi f_e t),\ \dot{w} = A_r f_e\cos(2\pi f_e t), \tag{27}$$

where A_r is the amplitude of the road profile (road unevenness) for the vertical movement of the wheel.

Taking into account Equation (27) in Equation (26) results in:

$$\begin{aligned} &m\ddot{h} + c\dot{h} + kh = c\dot{w} + kw = cA_r f_e\cos(2\pi f_{et}) + \\ &+ kA_r\sin(2\pi f_e t) = \sqrt{(kA_r)^2 + (c2\pi f_e Ar)^2} \times \\ &\times \sin(2\pi f_e t + \varphi), \end{aligned} \tag{28}$$

where φ is the phase shift of the wheel displacement.

The static component of the excitation force acting on the air bellow, which was generated by the wheel from the surface of the road, was determined using Equation (28).

$$F_{e0} = a_r k\sqrt{1 + \frac{c^2}{k^2}4\pi^2 f_e^2} = A_r k\sqrt{1 + 4\zeta_s^2\left(\frac{f_e}{f_n}\right)^2}. \tag{29}$$

The static stroke of air bellow is as follows:

$$h_{st} = \frac{F_{e0}}{k} = A_r\sqrt{1 + 4\zeta_s^2\left(\frac{f_e}{f_n}\right)^2}. \tag{30}$$

Substituting Equation (29) into Equation (24), the vibrations transmissibility T between the amplitude A_h of the air bellow (identical to the amplitude of the vehicle body) and the amplitude A_r of the wheel (identical to the amplitude of the road surface) is determined as a function of the frequency ratio f_e/f_n and self-damping ratio ζ_s:

$$T = \frac{A_h}{A_r} = \frac{\sqrt{1 + 4\zeta_s^2\left(\frac{f_e}{f_n}\right)^2}}{\sqrt{\left(1 - \left(\frac{f_e}{f_n}\right)^2\right)^2 + 4\zeta_s^2\left(\frac{f_e}{f_n}\right)^2}}. \tag{31}$$

The damping ratio of the self-damping air bellows is not constant and varies with the vibrations frequency. At high-frequency vibrations, the self-damping air bellow typically has a higher dynamic stiffness but a lower damping ratio. Increasing the air pressure in the self-damping air bellow increases both its stiffness and the damping ratio.

Vibrations transmissibility T depends only on the self-damping ratio ζ_s when the resonance frequency ratio $f_e/f_n = 1$:

$$T = \sqrt{1 + \frac{1}{4\zeta_s^2}}. \tag{32}$$

Vibrations transmissibility T depends only on the frequency ratio f_e/f_n when the self-damping ratio ζ_s is negligibly small:

$$T = \frac{1}{\left|1 - \left(\frac{f_e}{f_n}\right)^2\right|}. \tag{33}$$

The contribution of individual damping to the damping ratio was analyzed.

The contribution of the inherent damping component of the air bellow to the damping ratio ζ_s is in the range of 0.03 to 0.05 [22]. The exact value of this damping ratio component is highly dependent on the internal material configuration and the hysteresis of the air bellow.

The contribution of the pneumatic damping component of the air bellow in the damping ratio ζ_s is in the range of 0.15 to 0.25 [23].

For comparison, the typical damping ratio ζ_h in the under-damped range of a hydraulic absorber in an air suspension is approximately 0.65, which is 65 % of the critical damping ratio $\zeta_{cr} = 1$ [24]. The actual value of the damping ratio ζ_h varies depending on the damper design and control system. The damping ratio ζ_h of ASS shock absorbers is selected to provide a compromise between the comfort and stability, approximately 0.3-0.5 for a comfortable ride or 0.6-0.7 for better handling [25].

Figure 5 shows a comparison of the transmissibility T charts for different damping ratios: ζ_0 no-damping, ζ_i inherent damping, ζ_s pneumatic self-damping, ζ_h hydraulic absorber damping, and ζ_{cr} critical damping.

In the case of the damping ratio $\zeta_0 = 0$, for the resonance vibrations frequency $f_e = f_n$, the vibrations transmissibility T is infinite, $T = \pm\infty$. Furthermore, the value of vibrations transmission is $T = 1$ for $f_e/f_n > \sqrt{2} \approx 1.4142$. This implies that the selection of the ASS should be based on the assumption that $f_e/f_n >$ 1.4142, for which the vibrations transmissibility is $T = 1$.

The determination of the vibrations transmissibility T is crucial to understanding how excitation forces are transferred through the ASS in the body of the vehicle. The vibrations transmissibility T can be used to determine the vibrations amplitude A_h of the air bellow:

$$A_h \approx A_r T. \tag{34}$$

From Equation (34) it follows that for $T = 1$, the amplitude $A_h = A_r$. From the charts in Figure 4, it follows that for $f_e = 1.4142\ f_n$, the vibrations transmissibility $T = 1$ regardless of the damping ratio ζ.

Based on the dynamic model, the air bellow vibrations amplitudes A_h were numerically determined and compared for different damping coefficients z when $T = 1$. The calculations assumed a trailer air bellow with the following parameters: $D_{ef} = 0.100$ m, $H = 0.100$ m, $p = 6$ bar, $m = 2600$ kg.

The amplitude of the excitation force F_{e0} was estimated based on A_r and the stiffness k_r of the tire as follows:

$$F_{e0} \approx A_r k_r. \tag{35}$$

Based on the ISO 8608 standard, the typical irregularity amplitude A_r was set to 2 mm [22]. The stiffness k_r of the tire is 400 kN/m for the trailers. It is

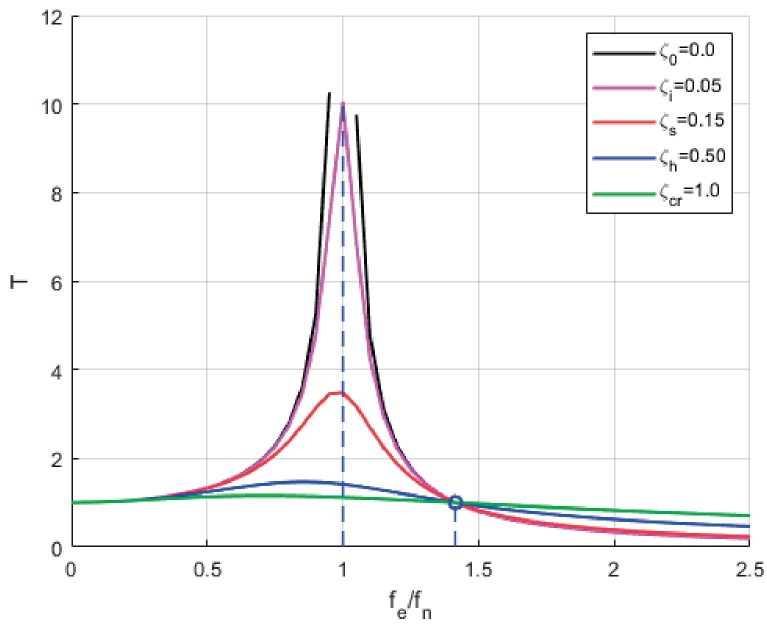

***Figure 5** Comparison of transmissibility T charts for different damping ratios ζ*

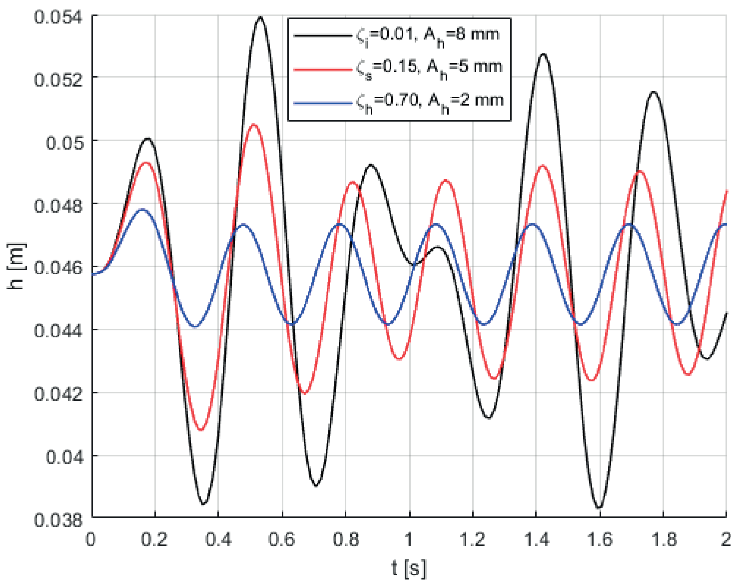

Figure 6 *Comparison of the stroke air bellow vibrations for different damping ratios ζ in the case of vibrations transmissibility, when T = 1*

assumed that at low frequencies (f_e < 5 Hz), the wheel moves almost exactly along the road profile and that the amplitude of the wheel displacement is almost equal to the height of the unevenness of the road.

Figure 6 shows a comparison of the stroke vibrations of the air bellow for different damping ratios z (inherent, pneumatic self-damping, hydraulic absorber) in the case of vibrations transmissibility, when $T = 1$.

A comparison of the stroke air bellow vibrations shows that condition in Equation (34) is satisfied ($A_h = A_r = 2$ mm) only for the damping ratio ζ_h of the external hydraulic damper (shock absorber). A self-damping air bellow with a fixed damping orifice may be insufficient for effective damping in air suspensions; therefore, further work will be conducted on semi-active self-damping air bellows.

5 Discussion

The scientific novelty of this study lies in the systematic application of vibrations transmissibility to the ASS with a self-damping air bellow model. Although the vibrations transmissibility is well known in vibration theory, this study applies it explicitly and comprehensively to ASSs with self-damping air suspension systems, which are less common in the literature. In this study are analytically linked the airflow-induced damping in an orifice and vibrations transmissibility, providing the frequency-dependent damping behavior. The direct comparison of pneumatic and hydraulic damping through transmissibility charts enables a quantitative assessment of the proximity of pneumatic self-damping to hydraulic damping performance. By considering the self-damping air bellows as a potential damper replacement under defined conditions, this study contributes to research on simplified, lighter, and energy-efficient suspension systems, which are particularly relevant for electric and commercial vehicles.

This study demonstrates that vibrations transmissibility T is crucial for understanding the transfer of excitation forces to the vehicle body through the ASS and affects ride comfort and safety. The vibrations transmissibility formula was determined from the solution of analytical dynamic equations for the classical and simple dynamic models of the air bellow. The classical model of the air bellow arises from its basic parameters, such as stiffness, natural frequency, effective area, and static state, as input for dynamic analysis of the simple dynamic model of the air bellow. The simple model of the self-damped air bellow considers its damping coefficient (damping ratio). In ASS, it is possible to use a self-damping air bellow stand-alone or in combination with a shock absorber. The vibrations transmissibility T was determined as the relationship between the amplitude A_h of the air bellow (vehicle body) and the amplitude A_r of the wheel (road surface) as a function of the frequency ratio f_e/f_n and self-damping ratio ζ_s. The vibrations transmissibility T states were considered as $T = 1$ for $f_e/f_n = 1.4142$, $T = \pm\infty$ for $f_e = f_n = 1$, and $T(f_e/f_n)$ for $\zeta_s = 0$. The transmissibility T charts for different damping ratios are compared: ζ_0 no-damping, ζ_i inherent damping, ζ_s pneumatic self-damping, ζ_h hydraulic absorber damping, and ζ_{cr} critical damping. At the characteristic point $T(f_e/f_n = 1.4141) = 1$, the amplitudes of each damping ratio were found to be equal. The stroke vibrations h of the air bellow was compared for the different analyzed damping ratios ζ in the case of vibrations transmissibility when $T = 1$. The analysis of these results indicates that the damping of the hydraulic absorber fully meets the requirements of the ASS. The self-damping air bellows can be used as stand-alone suspensions under appropriate road conditions.

Strengths of this study:

1. This study specifically focuses on vibrations, comfort and safety, which are critical design criteria for modern vehicles, including electric and heavy-duty vehicles.
2. In this study, both classical and simple dynamic models of the ASS air bellow were developed, providing a transparent theoretical framework for vibrations transmissibility analysis.
3. A comparison between the inherent and pneumatic self-damping of air bellows and hydraulic absorbers provides valuable information regarding the effectiveness of various damping solutions for the ASS.
4. The use of vibrations transmissibility T as a unifying metric enables a clear and physically meaningful evaluation of the transfer of road excitations through the ASS to the vehicle body to be performed.
5. The results lead to practical conclusions regarding the feasibility of self-damping air bellows with semi-active solutions, which is valuable for suspension design.

Weaknesses of this study:

1. This study is primarily analytical and numerical, and no full-scale experimental measurements of the vibrations transmissibility were presented to validate the predictions of the ASS model.
2. The ASS model was simplified since some components, such as the unsprung mass, tire stiffness, real excitation force, and vehicle dynamics, were neglected, which limited its direct applicability to the dynamic analysis of the full suspension system of the vehicle.
3. The self-damping air bellow uses a fixed damping orifice, which can be insufficient under certain road conditions, thereby reducing the effectiveness of its action.
4. Road excitation is primarily represented by sinusoidal inputs that do not fully capture stochastic real road profiles.

6 Conclusions

In this study were investigated the vibrations transmissibility characteristics of an ASS equipped with a self-damping air bellow to evaluate its suitability for improving vehicle comfort and safety. Based on the analytical modelling and comparative analysis, the following conclusions can be drawn:

1. Effectiveness of self-damping air bellows
   A self-damping air bellow exhibits measurable vibrations attenuation capability owing to the pneumatic damping generated by the airflow through an orifice between the air bellow and an auxiliary reservoir. Compared to an undamped air spring, the self-damping configuration significantly reduced the vibrations transmissibility near resonance, thereby improving the ride comfort.
2. Influence of damping mechanism on vibrations transmissibility
   The form and magnitude of the damping strongly affect vibrations transmissibility. Hydraulic shock absorber damping provides the lowest peak transmissibility and the widest effective isolation frequency range. Pneumatic self-damping offers moderate improvements but is less effective than hydraulic damping, particularly at low excitation frequencies.
3. Frequency-dependent performance damping
   The damping effect of a self-damping air bellow with a fixed orifice is highly frequency-dependent. Although the acceptable vibrations isolation can be achieved near the design frequency, the transmissibility increases outside this narrow operating range, thereby limiting the robustness of the performance under varying road conditions.
5. Design and application considerations
   A self-damping air bellow cannot fully replace a conventional shock absorber under all operating conditions. Adaptive or semi-active self-damping mechanisms with variable airflow resistance are required to achieve consistent vibrations control. Self-damping air bellows are best suited as complementary damping elements or primary dampers in applications with a well-defined excitation spectrum.
6. Contribution and future work
   In this study is demonstrated that the vibrations transmissibility is an effective evaluation metric for air suspension systems with pneumatic damping characteristics. The identified limitations of self-damping air bellows highlight the need for future studies focusing on semi-active control, experimental validation, and integration into full-vehicle suspension models. In this control system, the stiffness of the air bellows and pneumatic damping can be adjusted in real time for optimal ride comfort, regardless of road profile.

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Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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