

Design of an Active Air Spring Damper

Thomas Bedarff, Philipp Hedrich and Peter Pelz

Technische Universität Darmstadt, Chair of Fluid Systems, Magdalenenstr. 4,
D-64289 Darmstadt, Germany
E-Mail: peter.pelz@fst.tu-darmstadt.de

Since 2009 an active suspension system is under development at the Chair of Fluid Systems at TU Darmstadt. Aim of the project is to control uncertainties of load-bearing systems by adjusting the axial force via altering the effective area of the air spring bellows. This project is part of the Collaborative Research Center (SFB) 805, founded by the German Research Foundation DFG.

The working principle is realised by radially moveable piston segments. A prototype has already shown the potential of this concept. In the next phase of this project the prototype will be scaled and experimentally investigated in a Daimler W221 S-Class test car. The infrastructure of the installed Active Body Control System by Daimler will be used to power the new active suspension system hydraulically.

Topic of this paper is the investigation of the bellows concerning their durability. Furthermore a design method for active air spring damper suspension systems, which can be used to derive the necessary design parameters, is presented.

Keywords: active suspension, active air spring damper

Target audience: Simulation and Validation, Automotive, new applications

1 Introduction

Active suspension systems, like the well-known Active Body Control (ABC) by Daimler /1/ or the BOSE Suspension System /2/, which is still not available on the market, are the results of decades of research on this topic. Current developments are the Active Suspension via Control Arm (ASCA) by ZF Lemförder /2/ and the suspension system of the armoured military infantry vehicle Dingo II, which is used by the German Bundeswehr /3/. The Magic Body Control by Daimler, that is the successor of the ABC-System, was released in 2013. It is based on the ABC-System and additionally a preview controller, the so-called Road Surface Scan, is used to control the system depending on the condition of the road /4/.

All of these systems have in common that an actuator, which is hydraulically or electro mechanically driven, operates on the suspension or on a conventional spring. The systems are capable of influencing the vehicle's body movement up to excitation frequencies of five Hertz, which is approximately two times the body's eigenfrequency. By using active systems it is possible to overcome the conflict between comfort and driving safety.

The best passive suspension systems concerning comfort are air springs with adaptive dampers /5/. Therefore a combination of a passive air spring damper with an active system seems to be a promising idea to combine both comfort and dynamic behaviour.

This paper gives an overview of the major challenges during the development process of the active air spring damper.

First the basics of passive and active air springs are briefly explained. The main parts of this paper are the experimental and numerical examination of the air spring bellows and the design of an active air spring damper (aLFD), which fits in the Daimler W221-S-Class.

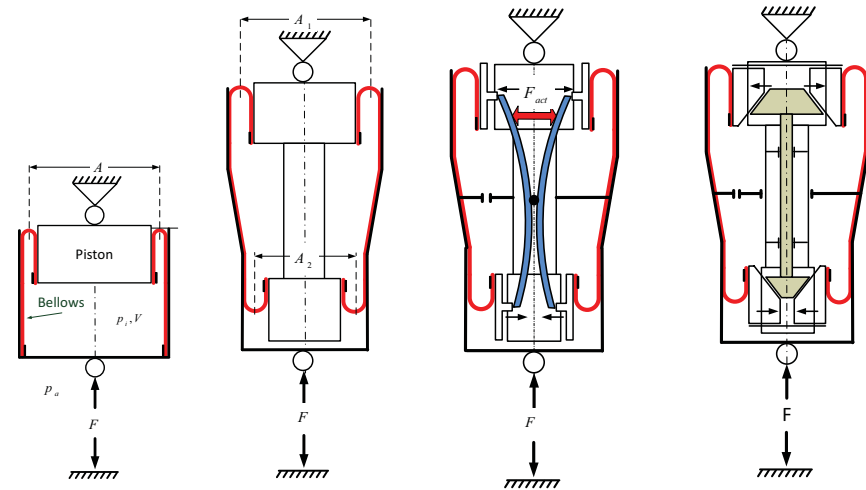


Figure 1: Single piston air spring (1), double piston air spring (2), active air spring damper with two active pistons (3 and 4) with different types of actuation and mechanical coupling

Figure 1 shows a conventional air spring (1), consisting of a piston and an air spring bellows. The volume of the enclosed gas is denoted by V , its pressure by p_i and the ambient pressure by p_a . The resulting axial force is given by $F = A(p_i - p_a)$ with the so-called effective area A . In general it is defined by $A := F / (p_i - p_a)$ and can only be identified experimentally. It depends on the gas pressure, the deflection of the air-spring and the bellows (structure, material, etc.).

The concept of the active air spring is based on the idea, that the resulting axial force can be altered quickly by changing the effective area A .

A further development of the single bellows air spring is the double bellows air spring shown in figure 1 (2). The normal vectors of area A_1 and A_2 point in opposite directions which results in the axial force $F = (A_1 - A_2)(p_i - p_a)$. The effective area is now the annulus area $A_1 - A_2$. Due to the fact, that A_1 is slightly larger than A_2 , minor changes in the diameters of A_1 and A_2 lead to relatively large changes in the resulting area $A_1 - A_2$ causing major changes in the resulting axial force F .

The diameter change is constructively realized by radially moveable piston segments. To demonstrate and examine this principle, a test rig with hydraulic driven piston segments is set up in the laboratory of the Chair of Fluid Systems at TU Darmstadt (/6/, /7/, /8/).

An active air spring damper with two active pistons, radially moveable segments and a mechanical coupling of the upper and the lower segments is displayed in figure 1 (3) and (4). Due to the coupling of the segments the static load acting on the upper and lower segments, which is caused by the pressure force on the segments, will be compensated. The simplest realization of this working principle would be a lever as shown on part (3) of picture 1.

However, research results show that this solution would result in giant levers, which do not fit into the available cross section. A feasible solution is shown on part (4) figure 1. The coupling is realized by a wedge gear. The two pistons are coupled with a central piston rod with wedges on both sides. The segments slide on the surfaces

of the wedges and an axial displacement of the central piston rod causes a radial movement of the segments. Both actuation and mechanical coupling are implemented in one component only.

2 Air spring bellows – an experimental and numerical investigation of the most crucial component

The most crucial component of the entire suspension system is the air spring bellows. For this reason its behaviour was investigated both numerically and experimentally to ensure that the active air spring is ready for usage in a test car.

In accordance with the project partner TrelleborgVibracoustic the goal is set that the bellows needs to withstand at least 300,000 sinusoidal compression and rebound cycles at 2 Hertz with 15 mm amplitude. This is of course still far away from 10,000,000 cycles requested for the series production but it is sufficient to test active air spring damper a couple of hours in the test car.

After pilot tests and minor improvement a passive piston, representing the exact geometry of the piston segments, was used to run the endurance tests (see figure 2). The boundary conditions for this test are shown in table 1.

Variable	Value
Initial pressure at design point	20 bar
Ambient temperature	~22°C
Excitation	2 Hz sinusoidal
Amplitude	15 mm

Table 1: boundary conditions for the bellows endurance test

After 500,000 compression and rebound cycles, the bellows was still hermetic, nevertheless a thorough examination showed significant abrasive wear of the bellows' outer surface in the transition area of the bellows convolution.

Conclusion: The current design of radially moveable segments, with a gap in between, in combination with a standard bellows is sufficient for at least 500,000 compression-rebound cycles. To reach cycle at the magnitude of 10,000,000, the geometry of the segments and even more important the design of the bellows have to be improved.



Figure 2: Passive piston with the active pistons geometry (left), abrasive damage to the bellows (right)

To improve the geometry of the active piston, a finite element model of the bellows was used. The model itself is provided by the industrial cooperation partner TrelleborgVibracoustic.

The 1.6 mm thick roller bellows is modelled with solid continuum elements arranged in three layers in total, two elastomer layers with a fiber reinforcement layer in between. The two fiber layer are laid to form a cross ply with a given angle between the fiber strings.

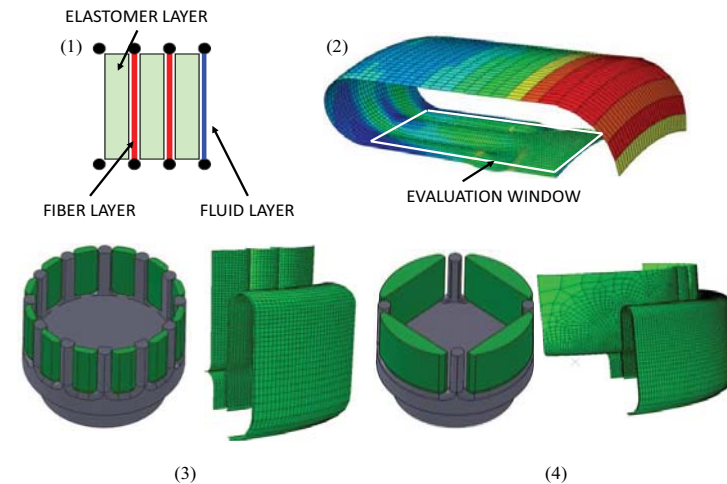


Figure 3: structure of the FE-model (1), convoluted bellows with evaluation window (2), depiction of a piston with 12 moveable segments, a piston with 4 moveable segments and the related bellows (3) and (4)

The aim of the numerical investigation described in this section is to find an optimal number of piston segments. The larger the number of piston segments, the smaller the gap in between them and the smaller the rate of change in the width of the gaps during segment movement. On the other hand, more piston segments lead to more deviation from a cylindrical piston surface.

The FE-simulation results in a tremendous amount of output data. For each element, the following output variables are calculated:

- RBFOR: Force in the rebar, calculated for the inner and the outer rebar layer
- RBANG: Angle in degrees between the rebar and the user-specified isoparametric direction

As reference for the analysis of the results serves an air spring with a cylindrical piston whose durability is proved for more than 10,000,000 compression-rebound cycles. To reduce the large amount of numerical results to a single comparable number, the root mean square (RMS) of all element data inside the evaluation window (figure 3 (2)) is calculated. For the analysis, the RMS is calculated for:

- The absolute maximum and minimum value of the rebar forces for the inner and outer rebar layer,
- The absolute maximum and minimum value of the rebar cord angles for the inner and outer rebar layer,
- The differences between the rebar forces and the rebar cord angles of the reference and the system with radially moveable piston segments, for both the inner and outer rebar layer,
- The maximum and maximum value of the difference between the rebar forces and the rebar cord angles of the reference and the system with moveable piston segments, for both the inner and outer rebar layer,
- The difference of the rebar forces between the inner and the outer rebar layer

The numerical simulations were done for systems with 4, 6, 8 and twelve moveable segments, each with a bellows rebar cord angle of 18°, 25° and 30°. The reference system is simulated with a rebar cord angle of 18°, 25° and 30° too. The gas pressure is set to 20 bar, gas temperature and ambient temperature are set to 20°C.

Figure 4 shows exemplary the results for the RMS of the deviation of the rebar forces from the reference. “Reference cord angle = cord angle” means, that the cord angle of the reference system is exactly the same as of the system with the moveable segments. The upper two graphs show the results for the inner rebar layer, the lower ones the results for the outer rebar layer.

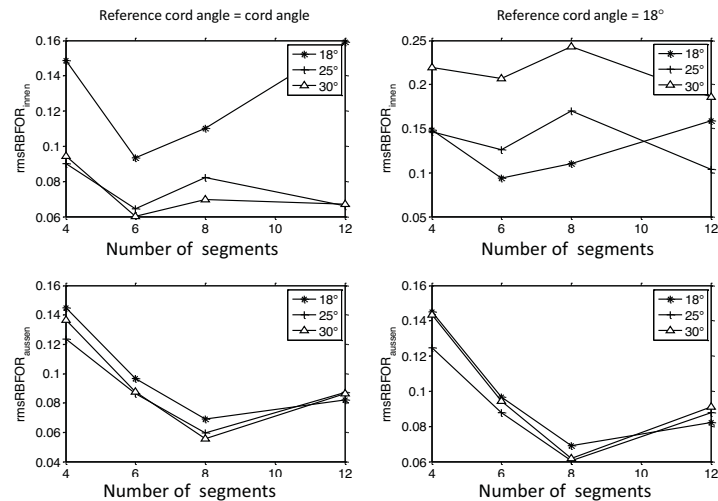


Figure 4: Results of the FE-simulation: RMS of the deviation of the rebar forces from the reference

It is obvious, that the smallest deviations from the reference for the outer rebar layer occur at 8 moveable segments, almost independent of the rebar cord angle. The results for the inner rebar layer are not that obvious. The system with 12 moveable segments is not much worse than the 6-segment system.

Conclusion: The analysis of the other mentioned criteria almost always leads to the same result that the optimal number of piston segments ranges from 6 to 8. The best results can be obtained with a rebar cord angle of 18°. Nevertheless, it needs to be mentioned, that the transfer of the numerical results to the real world is critical. Therefore, the result that 6 to 8 segments and a 18° rebar cord angle are the best configuration for the described active air spring damper are just reasonable assumptions.

3 Dimensioning of the active air spring damper

Under the assumption that the road excitation can be treated as a stochastic excitation, the standard deviation of the occurring quantities can be used to describe the system behaviour using a quarter-car-model with a linearized active air spring damper. According to Mitschke and Wallentowitz /9/ the standard deviation is defined as

$$\sigma_q^2 = \int_0^{50\pi} H^2 \Phi_h(\omega) d\omega \tag{1}$$

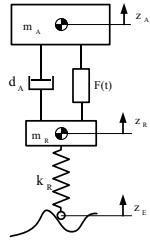
with the quantity q (e.g. the dynamic wheel travel, the wheel load fluctuation or the unsprung mass acceleration). The transfer function \hat{q}/\hat{z}_E is denoted by H and the power spectral density (psd) of the road profile $\Phi_h(\Omega) = v \Phi_h(\omega)$, which can be approximated by

$$\Phi_h(\Omega) = \Phi_h(\Omega_0) \left[\frac{\Omega}{\Omega_0} \right]^{-w} \tag{2}$$

where $\Omega = 2\pi/L$ denotes the wavenumber, $\Phi_0 = \Phi(\Omega_0)$ describes the value of the psd at the reference wavenumber $\Omega_0 = 10^{-6}$ m/rad and $\omega = v\Omega$ the waviness of the street. By setting the waviness to $w = 2$, the road profiles can be characterized by its reference value Φ_0 . In table 2 the classification of various road types is given.

	Classification	Min	Mean	max
$\Phi_h(\Omega_0)$	Autobahn	0.3	1.0	7.8
	Federal Highway	0.4	2.1	9.5
	Highway	0.4	5.3	29
	District road	0.7	12.2	62
w	Autobahn	1.5	2.0	2.6
	Federal Highway	1.7	2.3	2.9
	Highway	1.9	2.4	2.9

Table 2: Classification of various road types (source: /9/)



m_A	Sprung mass	485 kg
m_R	Unsprung mass	50 kg
d_A	Damper coefficient	1750 Ns/m
k_R	Wheel stiffness	150000 N/m

Figure 5: Quarter-car-model

The required dynamic spring deflection is defined as $z_{dyn} := 6\sigma_{z_A - z_R}$. In the same manner the required segment deflection is set to $r_{seg,max} = 6\sigma_{r_{seg}}$. To determine these characteristic values for a given road and car, a quarter-car-model with sky-hook-control is used (figure 5). Assuming a linear model for the air spring and the damper, the system behaviour can be described by the following set of differential equations:

$$\begin{pmatrix} \ddot{z}_A \\ \dot{z}_R \\ \dot{z}_A \\ \dot{z}_R \end{pmatrix} = \begin{bmatrix} \alpha C_2 - d_A & d_A & -C_3 & C_3 \\ m_A & m_A & m_A & m_A \\ d_A - \alpha C_2 & -d_A & C_3 & -C_3 + k_R \\ m_R & m_R & m_R & -m_R \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{pmatrix} \dot{z}_A \\ \dot{z}_R \\ z_A \\ z_R \end{pmatrix} + \begin{pmatrix} 0 \\ k_R \\ 0 \\ 0 \end{pmatrix} z_E \quad (3)$$

$$\begin{pmatrix} \ddot{z}_A \\ z_A \\ F_R \\ s \\ z_R \\ r_{seg} \end{pmatrix} = \begin{bmatrix} \alpha C_2 - d_A & d_A & -C_3 & C_3 \\ m_A & m_A & m_A & m_A \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & -k_R \\ 0 & 0 & -1 & 1 \\ 0 & 0 & 0 & 1 \\ \alpha & 0 & 0 & 0 \end{bmatrix} \begin{pmatrix} \dot{z}_A \\ z_A \\ z_R \\ z_R \end{pmatrix} + \begin{pmatrix} 0 \\ k_R \\ 0 \\ 0 \\ 0 \\ 0 \end{pmatrix} z_E,$$

with

$$\begin{aligned} C_1 &= \frac{\pi}{2} [(r_{k10} + r_{a1}) + (r_{k20} + r_{a2})k_s] \\ C_2 &= i \left(\frac{kp_0}{V_0} A_{T_{ax0}} A_{V_{seg0}} + (p_0 - p_a)C_1 \right) \\ C_3 &= i^2 \frac{kp_0}{V_0} A_{T_{ax0}} A_{V_{ax0}} \\ \alpha &= -\frac{r_{seg}}{\dot{z}_A} \end{aligned} \quad (4)$$

The transfer functions are now numerically calculated with the state-space function of the MATLAB control system toolbox, which converts a dynamic system to a state-space-system.

The following plots show some exemplary results. In figure 6 the required segment deflection is displayed depending on the type of road and the velocity. At a speed of 130 km/h on a federal highway a segment deflection of 6mm is needed to prevent actuator displacement saturation.

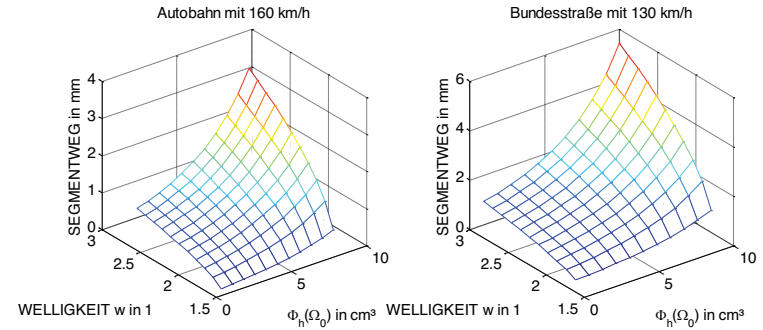


Figure 6: Actuator displacement depending on different road profiles and velocity

Figure 7 shows the results for the peak power consumption for each of the four active air spring dampers, figure 8 the required spring deflection.

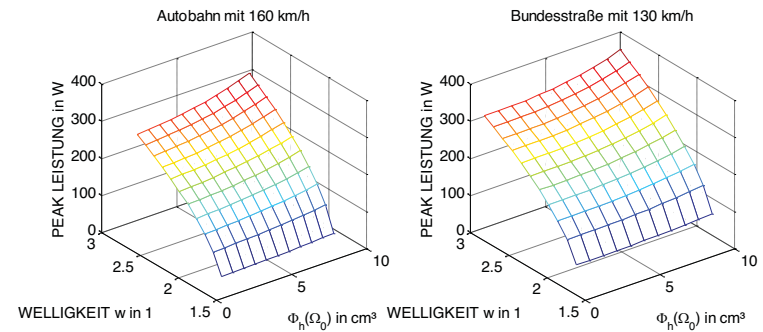


Figure 7: Peak power consumption depending on different road profiles and velocity

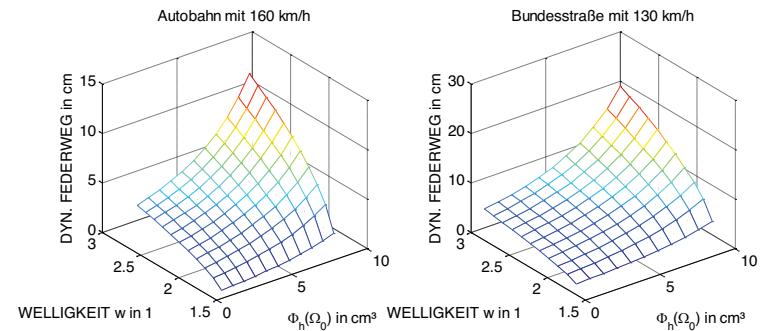


Figure 8: Spring deflection depending on different road profiles and velocity

4 Summary and Conclusion

The bellows is the most crucial component in an active air spring damper system. In this paper a short overview of the numerical and experimental investigation of such bellows is given and the results are presented.

By using the finite element method, it has been shown that the number of radially moveable piston segments, which are needed to alter the axial force of the active system, has an impact on the bellows. To compare and analyse the results, a reference passive air spring is used.

With 6 to 8 segments, the deviation to the reference reaches its minimum for almost any kind of evaluation criteria. Taking also economic aspects into account, it seems to be reasonable to use 6 segments.

To obtain the necessary values for system parameters like spring deviation, power consumption etc. a design method, based on stochastic street excitation, is presented.

For future work it is planned to compare these results with time domain simulations of a nonlinear model of the active air spring damper. It is expected, that due to the nonlinearity of the air damping, the results will differ from the results of the linear model.

5 Acknowledgement

The authors like to thank the German Research Foundation DFG for funding this research within the Collaborative Research Center (SFB) 805 “Control of Uncertainties in Load-Carrying Structures in Mechanical Engineering”.

Furthermore, the authors especially like to thank the project cooperation partner TrelleborgVibracoustic, namely Mr. Erhard Moog, Mr. Jürgen Berg and Mr. Michael Metzger for the excellent support.

Nomenclature

<i>Variable</i>	<i>Description</i>	<i>Unit</i>
p_i, p_a, p_0	Gas pressure, ambient pressure, initial gas pressure	[bar]
V, V_0	Gas volume, initial gas volume	[m ³]
F	Force	[N]
Φ_h	power spectral density (psd) of the road profile	[m ³]
Ω	Wavenumber	[1/m]
w	waviness of the street	[1]
q	quantity	[-]
σ	Standard deviation	[-]
κ	Isentropic exponent	[1]
ω	Angular frequency	[rad/s]
i	Axle drive ratio	[1]
r_{seg}	Segment displacement	[m]
m_A, m_R	Sprung mass, unsprung mass of the quarter car	[kg]
d_A	Damper coefficient	[Ns/m]

k_R	Tire stiffness	[N/m]
\ddot{z}_A, \ddot{z}_R	Acceleration of the sprung mass and the unsprung mass	[m/s ²]
\dot{z}_A, \dot{z}_R	Velocity of the sprung mass and the unsprung mass	[m/s]
z_A, z_R, z_E	Coordinate of the sprung mass, the unsprung mass and the excitation	[m]

References

- /1/ Pyper, M. Schiffer, W., Schneider, W., *ABC - Active Body Contro: von der Blattfeder zum aktiven System für mehr Fahrsicherheit und Komfort*, Verlag Moderne Industrie, 2003
- /2/ Heißing, B., Ersoy, M., *Fahrwerkhandbuch Grundlagen, Fahrdynamik, Komponenten, Systeme, Mechatronik, Perspektive*, Vieweg, 2011
- /3/ Schäfer, E., *Modular-hierarchische modellbasierte Entwicklung und Optimierung einer Regelung für ein aktives Federungssystem*, Dissertation Universität Paderborn, 2009
- /4/ Daimler AG, *Chassis technology: From independent wheel to active suspension (from 1930)* Internet <http://www.daimler.com/dccom/0-5-1317350-1-1323804-1-0-0-1317351-0-0-135-0-0-0-0-0-0-0-0-0-0.html>, January 2014
- /5/ Pelz, P. F., *PKW Federung und Dämpfung – Übersicht, Trends, Zukunft* Lecture, TU Darmstadt, 2008
- /6/ Bedarff, T., Pelz, P. F., *Development of an active and integrated suspension system.*, 8th. IFK, Dresden, 2012
- /7/ Bedarff, T., Pelz, P. F., *An active suspension with reduced complexity*, 7th. IFK, Aachen, 2010
- /8/ Bedarff, T., Pepter, P. F., *Design and Prove of Concept of an Innovative Active Fluid Suspension System*, VDI-Berichte 2183, pp 283-294, 2011
- /9/ Mitschke, M., Wallentowitz, H., *Dynamik der Kraftfahrzeuge*, Springer, 2004