

Air Spring Air Damper: Modelling and Dynamic Performance in Case of Small Excitations

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ABSTRACT

Air spring systems gain more and more popularity in the automotive industry and with the ever growing demand for comfort nowadays they are almost inevitable. Some significant advantages over conventional steel springs are appealing for commercial vehicles as well as for the modern passenger vehicles in the luxury class. Current production air spring systems exist in combination with hydraulic shock absorbers (integrated or resolved). An alternative is to use the medium air not only as a spring but also as a damper: a so-called *air spring air damper*.

Air spring air dampers are force elements which could be a great step for the chassis technology due to their functionality (*frequency selectivity, load levelling, load independent vibration behaviour, load dependent damping*). Some of their design which avoid dynamic seals by the using of rubber bellows contribute to a better ride comfort.

This paper is about a simulation model for the air spring air damper (LFD from its German name *Lüfffederdämpfer*) combined with an rubber bellow model. First of all it presents the working properties of the LFD, summarises the state of the art of simulation models for air spring air damper and gives some insight into the physics of such systems and their sensitivity to some parameters. Then it deals with rubber bellow and its behaviour under dynamic excitation. For this purpose, a phenomenological rubber bellow model is presented and validated with measurement. The combined model is calibrated based on an existing air spring air damper hardware, validated with measurements and integrated into a quarter car model for ride comfort simulation. At the end the simulation results are discussed to decide whether or not this should be taken in account for ride comfort simulation.

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INTRODUCTION

In recent years there has been an increased demand for ride comfort. These increased ride comfort requests cannot always be fulfilled by a classic chassis setup with steel springs and hydraulic dampers. These high requirements for ride comfort are achieved by conventional suspension and damping systems only at the cost of ride safety. A solution to this conflict lies in the use of active suspension systems, see [Figure. 1](#).

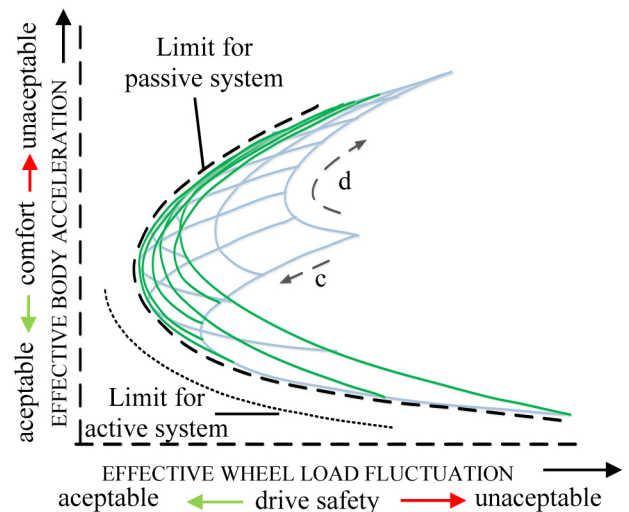


Figure 1. conflict between ride comfort and ride safety[2]

To these active suspension systems belong air springs with active integrated hydraulic shock absorber, see [Figure 2a](#). The use of air as a spring medium enables a load independent adjustment of the body floor height of the vehicle at a desired level (load levelling). The air suspension allows through its air supply and pneumatic control equipment a variable stiffness and a decoupling of the body vibration behaviour and the vehicle load: This is called load independent vibration behaviour. In conventional suspensions, the spring must be stiff enough to avoid the body from sinking too much, even at full load. This is disadvantageous in normal operation. The air suspension can be design softer than conventional suspension and provides therefore a better ride comfort. However, the friction coming from the piston seal of their integrated shock absorber causes rough handling. A solution is the use of air spring air damping systems, whose design avoid dynamic seals ([Figure 2b](#)).

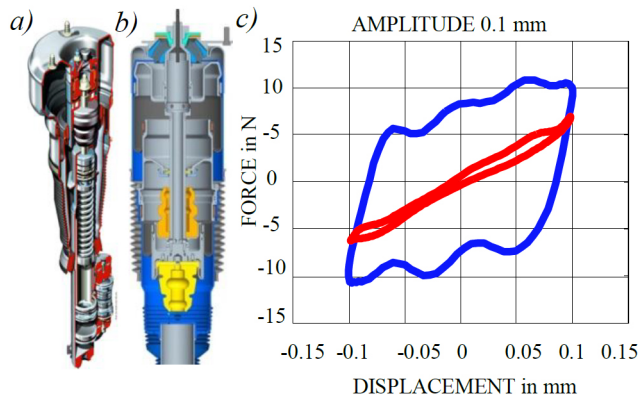


Figure 2. a) Air spring with integrated hydraulic shock absorber of the current Mercedes S class, b) air spring air damper prototype of the same class und c) Hysteresis curve of the both: blue for a) and red for b) by an excitation frequency of 1 Hz

Through the use of rubber bellows, there is no more mechanical connection between the body and the axle. This contributes to reduce the subjectively perceived rough handling. Despite this advantage by the using of rubber bellows as sealing element, the complex stress and strain state by some of their design inhibit their movement on the rolling piston, therefore a direct transmission of the perturbation by small excitation amplitude (harshness). For this reason it should be reduced, to allow the best ride comfort by large scale production of the air spring air damper.

SUSPENSION AND DAMPING BY AIR

The air spring air damper is a force element that consists in its simplest version of two chambers filled with air at a desired pressure p_0 and separated by a piston. The chambers are connected to each other through a valve, see [Figure 3a](#)). The LFD works with pure air as spring and damper medium. The suspension is carried out by the change of the total air

volume V and the damping by the throttle flow from one chamber to another through the valve. According to [9] the desired dissipated energy is not caused by the air internal friction in the valve but only down to the valve. Here splits the free jet into turbulent swirl in which the kinetic energy of gas particles is dissipated into heat. The LFD requires, like every air spring systems, an air supply and pneumatic control equipment, hence its need for more available space in comparison to the conventional suspension systems.

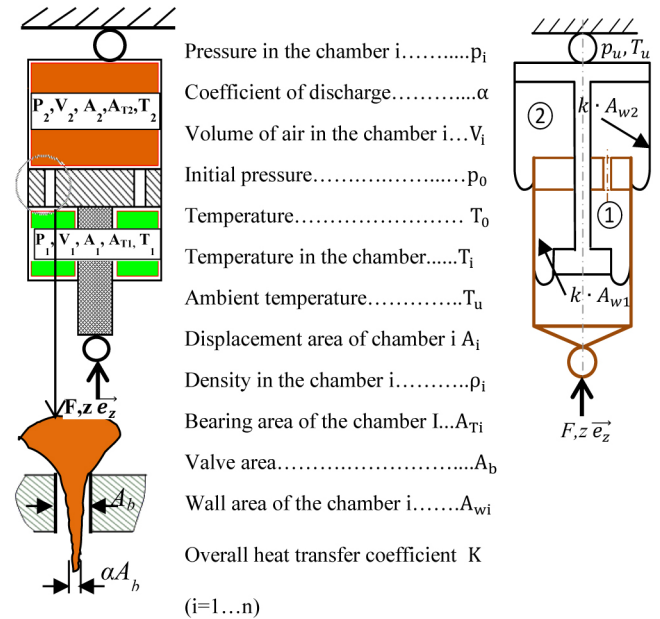


Figure 3. Physical parameters to describe a LFD. a) Principle Scheme and valve description and b) Scheme of a 2-chamber LFD

[Figure 4](#) shows in the column to the left ((a), b) and c)) the behaviour of the LFD in different frequency ranges. In the column to the right (d) e) and f)) there is a linear model of the LFD and its associated dynamic stiffness and dissipated energy in comparison to a hydraulic shock absorber.

Frequency Range $f \ll f_0$ (blue)

For frequencies less than f_0 which is the tuning frequency, the pressure is the same any time in both chambers. The piston has no effect and the LFD works in this case as a soft air spring with the total volume V and the adiabatic stiffness c_0 , see [Figure 4a](#)). For small excitation amplitudes about the initial position stiffness c_0 is given by

$$c_0 := \left. \frac{dF}{dz} \right|_{z=0} = \frac{\gamma p_0 A A_T}{V} + (p_0 - p_u) \left. \frac{dA_T}{dz} \right|_{z=0} \quad (1)$$

with the bearing area

$$A_T := \frac{F_0}{(p_0 - p_u)} = \sum_{i=1}^n A_{Ti} \vec{n}_i \cdot \vec{e}_z,$$

the displacement area

$$A := \sum_{i=1}^n -\frac{dV_i}{dz} = \sum_{i=1}^n -\text{sign}(dV_i/dz) \cdot A_i \quad (3)$$

and the total volume

$$V := \sum_{i=1}^n V_i. \quad (4)$$

Where $\gamma = 1.4$ is the adiabatic exponent, F_0 the resulting force and p_0 the initial pressure, both at the initial position.

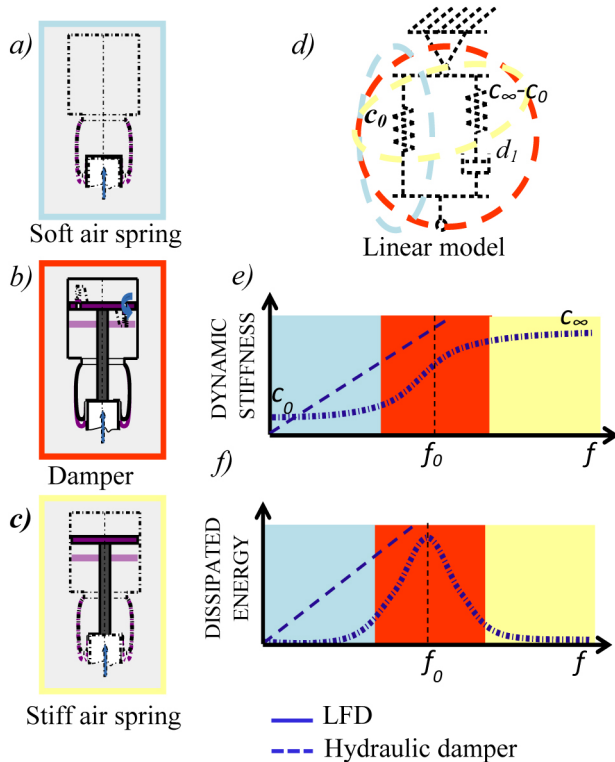


Figure 4. a)-c) LFD behaviour in different frequency domain and d)-f) a linear LFD model and a qualitative dynamic Behaviour of a LFD and a hydraulic damper

Frequency Range $f \gg f_0$ (yellow)

For frequencies f higher than the tuning frequency f_0 . There is no time for the air to achieve pressure compensation in both chambers (stiff air spring). The valve has no effect and the two volumes of the LFD act as two parallel connected air springs. In this case the highest level of stiffness c_∞ is reached, see Figure 4 c). For small excitation amplitude around the initial position the stiffness c_∞ is given by

$$c_\infty = \gamma p_0 \left(\frac{A_1 A_{T1}}{V_1} + \frac{A_2 A_{T2}}{V_2} \right) + (p_0 - p_u) \frac{dA_T}{dz} \Big|_{z=0} \quad (5)$$

Frequency Range $f \approx f_0$ (red)

At this range a transition takes place between the adiabatic stiffness and the upper stiffness c_∞ with increasing

frequency. Energy is dissipated only in this range (damper). This characteristic, which allows a damper to be adjustable and in terms of the desired frequency, is known in the literature as frequency selectivity. For small perturbation around the equilibrium the behaviour of the LFD can be approximate by a linear model, see Figure 4d). For a harmonic excitation with frequency f and small amplitude of excitation $\hat{z}$ the maximum energy dissipation W_{dmax} can be calculated as follows

$$W_{dmax} := \int_0^{1/f} F \dot{z} dt \Big|_{f=f_0} = \frac{\pi}{2} \hat{z}^2 (c_\infty - c_0) \quad (6)$$

Compared to other conventional suspension systems, the LFD which shows promising characteristics presents some disadvantages:

- The Energy dissipation leads to an increase in temperature. The lower heat capacity of air compared to oil leads to stronger warming. This must be accounted for in the design process.
- All parameters must be set properly already in the design phase. Otherwise, new construction requires subsequent changes.
- A hardening of the roll bellows used as seal occurs at small excitation amplitude.
- The coupling of the suspension and damping characteristic makes a vehicle tuning difficult.

MODELLING AND SENSIVITY ANALYSIS

This section presents the state of the art of the thermo-fluid-dynamic LFD model as well as presented some insight into physics of such systems [8]. The model is based on physical fundamentals and plausible assumptions. It allows not only a pure module description, but also dimensioning and tuning options. Therefore it can be used in all popular multi body simulation programs in the time and frequency domains for ride comfort, handling and NVH simulation.

The LFD Thermo-Fluid Dynamic Model

The following equations correspond to a two chamber LFD as illustrated in Figure 3b). For the explanation of the quantities used please also refers to the Figure 3. For convenience the partial differential $\partial/\partial t$ is denoted here by a dot. The model consists of:

Mass Conservation Equation

Moving up the piston in Figure 3 to $z\vec{e}_z$ up, the integral form of the conservation of mass in the both chamber becomes:

$$\dot{\rho}_1 (V_1 + zA_1) + \rho_1 \dot{z}A_1 - \dot{m} = 0$$

$$\dot{\rho}_2(V_2 - zA_2) - \rho_2 \dot{z}A_2 + \dot{m} = 0 \quad (8)$$

The first term on left side in (7), (8) describes the local change of mass, the second term describes the mass flow rate of the moving walls and the third term the mass flow rate as result of a valve flow.

Energy Conservation Equation

With the same consideration as in equations before, the integral form of the conservation of energy in the both chamber becomes:

$$\frac{1}{\gamma-1} \dot{p}_1(V_1 + zA_1) + \frac{1}{\gamma-1} \dot{p}_1 \dot{z}A_1 - \dot{m}c_p T_t + A_{w1}K(T_1 - T_u) = 0 \quad (9)$$

$$\frac{1}{\gamma-1} \dot{p}_2(V_2 - zA_2) - \frac{1}{\gamma-1} \dot{p}_2 \dot{z}A_2 + \dot{m}c_p T_t + A_{w2}K(T_2 - T_u) = 0 \quad (10)$$

The first on the left in (9), (10) describe the local change of internal energy. The second term describes the enthalpy flow of the moving walls and the third term the energy flow rate as a result of a valve flow. With T_t is the upstream temperature. The fourth term is the heat flux over the wall of the LFD neglecting the thermal inertia of the metal.

Ideal Gas Equation

The following equations give the thermal state of the gas in each chamber:

$$p_1 = \rho_1 R T_1 \quad (12)$$

$$p_2 = \rho_2 R T_2 \quad (13)$$

Valve Model

The flow behaviour across the valve is given in (14). The mass flow rate $\dot{m}$ is the only one responsible for the dissipation, as it is shown in [9]. With the same consideration as the mass and energy conservation equations used before the mass flow rate can be described by the Mach number M_e , the speed of sound a_e and ρ_e the density of the air jet at the cross-section αA_b .

$$\begin{aligned} \dot{m} &:= \iint_{\alpha A_b} \rho_e \vec{u} \cdot \vec{n} dS = \alpha A_b M_e \alpha_e \rho_e \text{sign}(p_2 - p_1) \\ &= \text{sign}(p_2 - p_1) \begin{cases} \alpha A_b \sqrt{2p_{up}\rho_{up}} \cdot \sqrt{\frac{\gamma}{\gamma-1} \left[\left(\frac{p_{dn}}{p_{up}} \right)^{2/\gamma} - \left(\frac{p_{dn}}{p_{up}} \right)^{\gamma+1/\gamma} \right]} & \frac{p_{dn}}{p_{up}} \geq 0.528 \\ 0.484 \alpha A_b \sqrt{2p_{up}\rho_{up}} & \frac{p_{dn}}{p_{up}} < 0.528 \end{cases} \end{aligned} \quad (14)$$

with p_{up} and p_{dn} respectively the pressure upstream and downstream.

In the model is assumed that there is no internal fluid friction and no heat exchange through the wall of the valve. Hence the acceleration of the air upstream until to the cross-

section αA_b is isentropic and the state of the gas is determined by stationary compressible Bernoulli equation for ideal gases.

If $\frac{p_{dn}}{p_{up}}$ is equal to or less than 0.528 as it is shown in [8], that means critical or over critical, the air in the chamber upstream is accelerated to the speed of sound. Because the information downstream cannot be transported in the opposite of flow direction with a speed higher than the sound speed, the mass and energy flow rate through the valve is independent of the thermodynamic state of the gas downstream.

If $\frac{p_{dn}}{p_{up}}$ greater than 0.528, that means under critical, the air jet speed at the cross section αA_b is less than the speed of sound. The pressure to the subsonic flow is determined by the surrounding air. Here the mass and energy flow rate through the valve depends on the thermodynamic state of the gas downstream.

Resultant Force

The following equation gives the LFD's resultant force

$$F = p_2 A_{T2} - p_1 A_{T1} - p_u (A_{T2} - A_{T1}). \quad (15)$$

Sensitivity Analysis

Let the resulting force of the LFD defined in (15), is the force response to a harmonic excitation $z = \hat{z} \sin(\omega t)$. The application of the force and the displacement of the spring over time have a phase shift δ , which is known in the literature as loss angle. Plotting the force over the displacement of the spring, a hysteresis curve is formed, see Figure 5.

To determine the transfer behaviour to harmonic excitation for the simulation and also for the experiments, the following quantities can be defined: the dynamic stiffness and the dissipate energy. The dynamic stiffness is defined as follows

$$c_{dyn} := \frac{F_{max} - F_{min}}{2\hat{z}} \quad (16)$$

where F_{max} and F_{min} are the maximum and minimum forces and $\hat{z}$ the excitation amplitude. The dissipate energy W_d corresponds to the dissipation energy per oscillation cycle. It can be calculated by a numerical integration of the force-displacement-hysteresis curve of a vibration cycle.

$$W_d = \oint F dz. \quad (17)$$

With the help of the physical LFD model describe on the previous chapter, a prediction of amplitude- and frequency-dependent behaviour of LFDs can be derived.