Uncertainties with respect to active vibration control
Prof. Dr.-Ing Peter F. Pelz\textsuperscript{1, a}, Dipl.-Ing. Thomas Bedarff\textsuperscript{2, b}, Dipl.-Ing Johannes Mathias\textsuperscript{3, c}

\textsuperscript{1} Technische Universität Darmstadt, Chair of Fluid Systems Technology, Magdalenenstraße 4, 64289 Darmstadt, Germany
\textsuperscript{2} Technische Universität Darmstadt, Chair of Fluid Systems Technology, Magdalenenstraße 4, 64289 Darmstadt, Germany
\textsuperscript{3} Technische Universität Darmstadt, Fachgebiet Produktentwicklung und Maschinenelemente, Magdalenenstraße 4, 64289 Darmstadt, Germany

\textsuperscript{a} peter.pelz@fst.tu-darmstadt.de, \textsuperscript{b} thomas.bedarff@fst.tu-darmstadt.de, \textsuperscript{c} mathias@pmd.tu-darmstadt.de

Keywords: active suspension system, vibration control, robust design

Abstract. The content of this work is the presentation of the prototype of a new active suspension system with an active air spring. As being part of the Collaborative Research Unit SFB805 “Control of Uncertainties in Load-Carrying Structures in Mechanical Engineering”, founded by the Deutsche Forschungsgemeinschaft DFG, the presented active air suspension strut is the first result of the attempt to implement the following requirements to an active suspension system:

- Harshness and wear: Reduced coulomb friction, i.e. no dynamic seal.
- Plug and drive solution: Connected to the electrical power infrastructure of the vehicle.
- Vehicle and customer application by software and not by hardware adaption.

These requirements were defined at the very beginning of the project to address uncertainties in the life cycle of the product and the market needs.

The basic concept of the active air spring is the dynamic alteration of the so-called effective area. This effective area is the load carrying area $A$ of a roller bellow and defined by $A := \frac{F}{p - p_a}$. $F$ denotes the resulting force of the strut, $p$ the absolute gas pressure and $p_a$ the ambient pressure. The alteration of this effective area is realized by a mechanical power transmission, from a rotational movement to four radial translated piston segments. Due to the radial movement of the piston segments, the effective area $A$ increases and so does finally the axial compression force $F$.

The prototype presented in this paper serves as a demonstrator to proof the concept of the shiftable piston segments. This prototype is designed to gather information about the static and dynamic behavior of the roller bellows. Measurements show the feasibility of the concept and the interrelationship between the piston diameter and the resulting spring force.

Introduction

Full Active suspension systems allow controlling heave, roll, and pitch motion of a vehicle body. In the following two different systems are compared. From the strengths and weaknesses analysis of those solutions the demands of innovative active suspensions are discussed.
Fig 1: Left hand side, the Daimler active body control (ABC) [source: Daimler], right hand side, Bose-Suspension-System [source: Bose]

First (i) the up to now most advanced application of such an active suspension system for passenger cars is the active body control (ABC) suspension system invented by Daimler. The body motion is controlled up to a frequency of 5 Hz by means of hydraulically controlled servo cylinders in the four spring struts. The active base displacement of the coil spring together with the passive hydraulic shock controls the body movement [1]. Second (ii) for the electromagnetic Bose system [2], being still in the concept and not production phase, the base displacement of the torsional spring is realized by the electromagnetic force of a linear motor.

Tab. 1: Comparison of the Daimler and the Bose active suspension system. The points 1) to 4) are addressed in the text in more detail.

<table>
<thead>
<tr>
<th></th>
<th>hydraulic base displacement of a coil spring (Daimler)</th>
<th>electromagnetic base displacement of a torsion bar (Bose)</th>
</tr>
</thead>
<tbody>
<tr>
<td>electrical power supply at each suspension strut</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td>mass specific power</td>
<td>+</td>
<td>- 1)</td>
</tr>
<tr>
<td>comfort at high frequency, small amplitude (Coulomb friction)</td>
<td>- 2)</td>
<td>+</td>
</tr>
<tr>
<td>fail save (drivable without energy supply)</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>logistic and assembly costs for different suspension system on one vehicle platform</td>
<td>- 3)</td>
<td>-</td>
</tr>
<tr>
<td>maintenance costs</td>
<td>- 3)</td>
<td>-</td>
</tr>
<tr>
<td>one hardware solution meets different OEM specifications by software adaption</td>
<td>- 4)</td>
<td>-</td>
</tr>
</tbody>
</table>
Table 1 compares the hydraulic and the electromagnetic active suspension systems of Daimler, shown in Fig 1 left hand side, and Bose, shown in Fig. 1 right hand side. Even though such a comparison is always subjective and always critical, it should be done at the start of a new development. The four points especially marked in table 1 are discussed in more detail in the following:

1) **Power specific weight:** Comparing electromagnetic with hydrostatic motors, the mass specific power of the hydrostatic side is always favorable due to the high fluid pressure. This is the case for active suspension systems as well, where the power is determined by the body mass, the velocity and demand on control. Roughly the weight of electromagnetic devices is by one order of magnitude above the weight of the comparable hydrostatic device.

2) **Harshness due to coulomb friction:** As pointed out, the Daimler ABC system consists of a conventional hydraulic shock absorber and a (stiff) coil spring. Thus the suspension is harsh first due to the high eigenfrequency and second due to the Coulomb friction within the hydraulic shock absorber. For amplitudes smaller than the friction force divided by a typical suspension stiffness the harsh appearance increases significantly.

3) **Uncertainties due to Manufacturing and maintenance costs:** The logistic and assembly costs are significant in the case where for example the customer has the choice between an air suspension and an active body control. The complete chassis infrastructure depends on the customer choice.

4) **Uncertainties due to market needs:** Related to that point is the demand to change the vehicle dynamics not by changing the hardware, i.e. the suspension strut, but by changing the controller, i.e. by a software solution. Thus one suspension strut would meet the expectation of a BMW driver, who is looking for a sportive chassis, or a Mercedes driver, enjoying the driving comfort, and uncertainties in the market needs can be addressed.

Following the above realized analysis, three tasks were defined at the very beginning of the project to address uncertainties in the life cycle of the product and the market needs:

(i) **Harshness and wear:** Reduced coulomb friction, i.e. no dynamic seal.

(ii) **Plug and drive solution:** Connected to the electrical power infrastructure of the vehicle.

(iii) **Vehicle and customer application by software and not by hardware adaption.**

As being part of the Collaborative Research Unit SFB805 “Control of Uncertainties in Load-Carrying Structures in Mechanical Engineering”, founded by the Deutsche Forschungsgemeinschaft (German research foundation), the main focus of our research was the question “how can we manage uncertainties in active suspension systems?”.
**Table 2: How to meet uncertainties in active suspension system**

Table 2 gives a first reflection on that question. There is a clear difference between the overall market needs, the individual car manufacturer (OEM), and the later vehicle owner. Reviewing table 1 and 2 the design should be smart, it should be a plug and drive solution and the dynamic characteristics as well as the power management including recuperation - or as we call it, phase controlled energy feedback -, should be controllable by software.

**Basic Concept**

Air suspension is the state of the art suspension for luxury vehicles, sedan or SUV. The main reason for the success of air suspension systems is the invention of a thin (1.6 mm wall thickness) rolling bellow, which came to production in the Daimler S-Class in 1998. Essential for the success of an invention is the relation between customer values (which can also be emotional, i.e. subjective values) to costs for customer [3]. This ratio is superior for air spring systems in comparison to other suspension systems.

Fig. 2 shows two principle air spring designs in a schematic drawing.

The double roller bellows principle (Fig. 2b) allows a significant reduction of the load carrying area and is a promising concept to reduce coulomb friction, i.e. to reduce harshness [4].

The load carrying area $A$ of an air spring bellows, sketched in Fig. 2a, is given by that diameter, where the bellows loop has a radial tangent. For the purpose of our research a very interesting concept is the double roller bellows concept shown in Fig. 2b. For this concept the load carrying area is given by $A=A_1-A_2$, where the index 1 denotes the upper bellows and the index 2 the lower one.
For all the fluid suspension systems shown in Fig. 3 the static equilibrium yields

$$F = (p - p_a)A,$$

where $p$ denotes the absolute pressure (gas or liquid) within the device and $p_a$ the ambient pressure.

The very simple relation (1), which in turn serves to define the load carrying area, $A := F/(p - p_a)$, allows us to discuss all fluid suspension systems shown in Fig. 3 in a unified manner. Before doing so we have to define the displacement area $A_d$ of the fluid suspension system as

$$A_d := -\frac{dV}{ds},$$

where $V$ is the gas volume and $s$ the compression displacement of the suspension strut as shown in Fig. 2. The index “d” stands for displacement. For the special case of a plunger piston there is no difference in the area, $A_d = A$, whereas for a rolling lope or bellows, $A$ is slightly greater than $A_d$ due to kinematic reasons. Similar we define a displacement length as

$$h_d := -\frac{dV}{dA}.$$

From the static equilibrium (1) the relative change in force follows as

$$\frac{dF}{F} = \frac{dA}{A} + \frac{dp}{p - p_a}.$$

Typical for a technical fluid suspension system the absolute pressure $p$ is at least by one order of magnitude greater than the ambient pressure $p_a$. Hence in most cases it is justified to write

$$\frac{dF}{F} \approx \frac{dA}{A} + \frac{dp}{p}.$$

With the typical length of the suspension system defined by the gas volume divided by the typical area, which is the inverse of the volume specific area of the device,

$$l := V / A_d \approx V / A.$$
Fig. 3: Overview of fluid suspension systems: (4) air suspension; (3) active hydro-pneumatic; (2) active base displacement (hydraulic or magnetoelectric) and the thermal diffusivity of the gas $\alpha = \lambda / \rho c_p$, the cut-off-frequency for the transition from isothermal to isentropic change is given by

$$\omega_\gamma = \frac{a}{l^2}.$$ (7)
The heave eigenmode of an air suspended mass has the eigenfrequency

$$\omega_0 = \sqrt{\frac{g}{l}}. \tag{8}$$

Hence with $\omega_0 \gg \omega_f$ the condition for isotropic change of state is

$$\sqrt{\frac{a^2}{\gamma g l}} \ll 1. \tag{9}$$

It is fulfilled for most technical fluid suspension systems for dynamic applications. Hence the isentropic relation

$$p(\rho) = \text{const} \ \rho^\gamma \tag{10}$$

is valid for the dynamic change of state, with the homogeneous gas density $\rho = m/V$. $\gamma$ is the isentropic exponent which assumes the value of $7/5$ for air.

With Eq. 10 the relative force change (Eq. 5) becomes

$$\frac{dF}{F} \approx \frac{dA}{A_1} \gamma \frac{dV}{V_0} - \frac{dm}{m_4}. \tag{11}$$

The change of the gas mass $dm$ within the suspension strut is in most cases quasi-static and hence isothermal, i.e. non isentropic. Thus the equation of state is isothermal and hence $p(\rho) = \text{const} \ \rho$.

The volume change in Eq. 11 can be considered to be

$$dV = -A_d \frac{ds}{s_0} - h_d \frac{dA}{A_1} - A_d \frac{dz}{z_2} - A_d \frac{dx}{x_3}. \tag{12}$$

The different terms in Eq. 11 and Eq. 12 are labeled by the very same labels used in Fig. 3. Thus the appearance of the different effects in different technical solutions for fluid suspension systems ranging from the classic air suspension system (term (0), (4)) hydro-pneumatic system (term (3)), base displacement system (term (2)), to the one we consider in our work, where we change the rolling piston area (term (1)), becomes obvious.

Thus the relative force change of the most general quasi-static fluid suspension system becomes

$$\frac{dF}{F} \approx \gamma \frac{ds}{V_0} A_d + \frac{dA}{A_1} \left(1 + \gamma \frac{h_d A_d}{V_2}\right) + \gamma \frac{dz}{V_2} A_d + \gamma \frac{dx}{V_3} A_d + \frac{dm}{m_4}. \tag{13}$$

As it becomes clear, Eq. 13 is valid for the conservative suspension strut (no damping force) in its equilibrium. Term (0) represents the spring force of the suspension system. Term (1) represents the relative force related to the change of the cross section area, which is addressed in our research. Term (3) represents an active base point displacement. As it becomes obvious this is from a physical point of view equivalent to an active change of the gas volume, which is done by active hydro-pneumatic solutions. The last term represents the change of gas mass which is achieved by a pneumatic infrastructure within the chassis (compressor, valves and air dryer).

Even though the terms (2) to (4) appear harmless, the infrastructure and hardware required within a suspension system to gain these effects are essential.

Due to the slowness of the gas supply system, the effect (4) can be used for quasi-static leveling purposes, but not for an active suspension system.
The advantages concentrating on the change in the carrying area are:

- There is no need for an external pump or compressor, i.e. the solution is a pump-less system.
- It is a “plug and drive” solution, since the internal actuator should be electrically driven.
- A small change in the cross section of the piston would result in a large change in the cross section area especially for the double bellow solution shown in Fig. 2 on the right.
- The package within the two pistons can be used to integrate the actuator.

Hence, for the mentioned reasons, it is worthwhile to consider only term (1) in Eq. 13 and to find a design solution for the principle given by

\[
\frac{dF}{F} \approx \gamma \frac{ds}{V} A_d + \frac{dA}{A} \left(1 + \gamma \frac{h_d}{l}\right),
\]

for the one bellows solution and

\[
\frac{dF}{F} \approx \gamma \frac{ds}{V} A_d + \frac{dA}{A} \left(C_1 - \frac{dA_2}{dA_1} C_2\right),
\]

with the two constants \( C_{1,2} = 1 \pm \gamma \frac{h_{n,2}}{l} \).

for the double bellows solution. Preferably \( dA_2/dA_1 < 0 \) to enhance the actuator. The linearization of Eq. 15 yields

\[
\frac{\Delta F}{mg} \approx \gamma \frac{s}{l} + \frac{\Delta A_1}{A} \left(1 - \frac{dA_2}{dA_1}\right)
\]

for \( h_d << l = V/A_d \).

**Transfer function of the active suspension system**

The equation of motion for the supported mass reads (see Fig. 4 left hand side)

\[
m \ddot{z} = \Delta F.
\]

With Eq. 16 this results in

\[
\ddot{z} \approx g \gamma \frac{s}{l} + g \frac{\Delta A_1}{A} \left(1 - \frac{dA_2}{dA_1}\right)
\]

with \( s = z_0 - z \) or

\[
\ddot{z} + \omega_0^2(l) z = \omega_0^2(l) z_0 + g \frac{\Delta A_1}{A} \left(1 - \frac{dA_2}{dA_1}\right).
\]

with the eigenfrequency \( \omega_0 = \sqrt{\gamma \frac{g}{l}} \).

Assuming a harmonic base displacement \( z_0 = \hat{z}_0 \sin \Omega t \), the transmission function becomes

\[
\frac{\hat{z}}{\hat{z}_0} = \frac{1 + l \frac{\Delta A_1}{\hat{z}_0 \gamma} \left(1 - \frac{dA_2}{dA_1}\right)}{1 - (\omega_0 / \Omega)^2}.
\]
The factor \((1/dA_2/dA_1)\) in Eq. 16 serves as an amplification factor. With the double bellows concept, the two pistons are coupled cinematically. Whenever the upper piston gets expanded \((dA_1>0)\), the lower one gets contracted \((dA_2<0)\) and vice versa.

**Closed Loop Control**

Fig. 4 on the right hand side shows the two degrees of freedom vehicle model, consisting of a cassis mass \(m_C\), a wheel mass \(m_W\) and the active fluid suspension system as an actuator. For the passive system, the actuator is replaced by suspension with the stiffness \(k_C\) and a damper coefficient \(d_C\). The wheel excitation is given by a measured road profile, taken from a country road for a speed of approximately 70 km/h.

With respect to vertical vehicle dynamics, the closed loop control is intended to meet two requirements:

1) Comfort control: to have the ride as comfortable as possible the acceleration of the chassis should vanish, i.e. \(\ddot{z}_C = 0\).

2) Driving safety: to transmit transfer forces to the street, a minimal wheel load fluctuation is required.

These two goals are in conflict with each other, more driving comfort for example leads to less driving safety. This conflict is illustrated in Fig. 6: The driving comfort is given by the vibration intensity (in this work: not weighted), the driving safety by the normalized wheel load fluctuation. In Fig. 6, the wheel load fluctuation is normalized with the wheel load fluctuation given for an ideal actuator, i.e. the chassis acceleration vanishes.

For a passive system a variation of the chassis suspension spring stiffness and damper coefficient leads to the conflict diagram depicted in Fig. 6. A higher damper coefficient \(d_C\) of the chassis damper results in a reduced wheel load fluctuation (increasing driving safety), a lower chassis suspension spring stiffness \(k_C\) results in a reduced chassis acceleration (more driving comfort). With the passive system, is not possible to decrease both the chassis acceleration and the wheel load.
fluctuation beyond the cross curve. A lower wheel load fluctuation will always result in higher chassis acceleration. The white points in this conflict diagram indicate that the maximum relative travel of the suspension is exceeded.

With an active suspension system, it is possible to overcome these restrictions. The goal of each active suspension system is to reduce the wheel load fluctuation as well as the chassis acceleration, i.e., reach the point of origin in the depicted conflict diagram. To do so, a sophisticated control algorithm is needed which deals with the vertical dynamics as well as with the longitudinal and lateral dynamics. In this work, only the vertical dynamics is considered. As a first approach, the active fluid suspension system is equipped with a combined sky-hook ground-hook controller. The sky-hook controller (damper coefficient $d_{\text{sky}}$) applies a force against the chassis velocity; the ground-hook controller applies a force against the wheel velocity. As expected, the chassis acceleration can be reduced significantly without violating the deflection limits. However, the wheel load fluctuation increases due to the concept of the controller.

**Design concept of the new active fluid suspension system**

Fig. 7 right-hand side shows the above discussed change of the load carrying area in more detail. The piston is divided in segments which are forced radially outwards. Due to changes in the roller fold, the load carrying area changes as well. In Fig. 7 the load carrying area $A_1$ is enlarged and the load carrying area $A_2$ is reduced. Hence the difference cross sectional area $A=A_1-A_2$ is enlarged as well. Due to the fact that two pistons are used and $A$ is significant smaller than $A_1$ or $A_2$, a small change of the single load carrying areas $(A_1,A_2)$ will have a relatively large impact on the change of the difference cross sectional area $A$ and hence a large impact on the relative force change $\Delta F/F$ given in Eq. 4.

Fig. 7 left-hand side shows the principle effect of the alteration of the load carrying area as the result of a simple calculation: on the abscissae the diameter of the upper piston is plotted, on the left ordinate the resulting compression force at constant damper compression travel and on the right ordinate the associated diameter of the lower piston. The compression travel $s$ of the suspension system is the parameter. In this graph, the labeled compression travel (-70 mm to +70 mm) is measured from the design position.
The white square markers in Fig. 7 show the change of the load carrying area in the design state. Within the project the design state is defined for a static compression force of 7.5 kN. The specification meets roughly the requirements of a luxury passenger vehicle.

Within the project for the final design solution of the active system, the lower piston always changes its diameter when the upper one does but in an opposite sense, to enhance the sensibility of the system. The dashed line in Fig. 7 shows: For a load of 9 kN the diameter $D_1$ is widened to 123 mm whereas $D_2$ is reduced to 96 mm. This spread in diameter increases with increasing compression travel $s$ due to the increase in gas pressure.

**Robust Design.** For the technical realization of the alternating load carrying area a solution had to be found. The solution has to solve the following conflict: On the one hand there are large forces due to the pressure inside the bellows that must be overcome. And on the other hand the package space inside the piston is very limited. A feasible solution for this conflict is the radial shifting of the piston segments, described in the next section.

Fig. 8 shows the radial shifting of the piston segments. The thick line symbolizes the roller bellows. The problem with this kind of expansion is the interaction of bellows and piston. Throughout the detailed design of the piston segments two main uncertainties occur:

1. The problem seems to be the interaction of the bellows and the piston. Due to the pressure inside, the bellows adheres to the piston. Will the bellows glide along the piston surface as it is necessary to shift the segments?

2. An outward shifting of the piston segments, leads to a stretching of the bellows only between the segments. Will the bellows be destroyed by this stretching?

Based on design reviews with bellows experts the recommendation was to build up a test system to get more information about both uncertainties. In this test system bellows with different properties (rubber, fiber angle etc.) should be stretch inwards and outwards over many periods. Due to the results and the so increasing knowledge design changes for example of the bellows fiber angle should be discussed to design a working solution. It is easy to understand that this is a risk full approach for the designer. Design time, monetary and material effort will increase a lot. Additionally if the results are negative the designer needs a new solution or the whole concept may fail.
At this point it is very important for the designer to think about a design change that leads to a robust solution where the consideration of both uncertainties is not necessary anymore. By this approach no additional knowledge will be necessary saving time and money.

This robust solution should avoid a gliding between the bellows and the piston and reduce the stretching of the bellows to a minimum. A possible solution with both approaches integrated in a small but effective design change is shown in Fig. 8:

![Fig. 8](image)

**Fig. 8:** Principle of the piston widening by shifting piston segments and by using a gap between the segments.

The robust solution is based on the special geometry of the piston surface. In Fig. 8 two piston segments are shown. A gap was put in between the two parts. The bellow lies in this gap when relaxed and gets tensioned when the piston segments shift outwards (the piston expands). With this technique the segments can be moved without putting too much strain on the bellow. No gliding of the bellow along the piston appears and the bellow is bended but not stretched. This bending has no problematic impact on the bellow as it is the main working principle of the rolling bellow.

**Feasibility study of the concept by a finite element simulation.** The practicability of this idea is tested by a numerical simulation. A nonlinear finite element model of the roller bellows was developed together with Vibracoustic GmbH & Co. KG, a company of the Freudenberg group, and consequently enhanced for this research. With the help of the numerical model the assembly process, starting from the installation of the roller bellows to the alteration of the load carrying area could be analyzed. The 1.6 mm thick roller bellows is modeled with solid continuum elements arranged in three layers: 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 directions. This structure is modeled in continuum approach. The simulations, not shown here, show a robust solution not for all but some fiber materials.

**Prototype and prove of concept.** The technical realization of the shifting piston segments is shown in Fig. 9. An axle, similar to a camshaft is powered by a gear wheel (Fig. 9, left). The cam glides on hardened pads and pushes the piston segments outwards. The camshafts and the piston segments are mounted with floating bearings (linear bearings for the segments and radial bearings for the camshaft).

Four of these camshafts are mounted inside the piston and are powered by one gear wheel (Fig. 9 left hand side). The axle driving shaft is powered by a hydraulic swivel motor with a torque of up to 400 Nm at 200 bar hydraulic pressure. This high torque and the connected high power consumption are necessary because a force of up to several Kilo Newton is needed to move the segments. It is easy to explain, where these big forces come from: If for example the absolute pressure inside the bellows is 11 bar, an integration of this pressure over the area that is in contact with the piston (circumference 300 mm, height 35 mm) leads to a pressure related force of approximately 2.5 kN per segment. A solution for this challenge has to be developed in future work. The current concept mainly deals with the bellows expansion and helps to gather data and experience about the systems behavior.
In the Introduction the concept of two varying pistons was presented. The prototype though is a suspension strut with two pistons but only one of them, the top one, is variable. Therefore the influence of the segment shifting on the load carrying area is less, but still sufficient to show the principle feasibility and to collect information about the system behavior.

Fig. 10 shows the test-bench and a schematic diagram of it. The swivel motor is mounted on top of the upper piston and controlled with a closed loop circuit. The active suspension is mounted into an servo hydraulic test rig. Therewith it is possible to emboss the system with definite amplitudes and frequencies and to measure the resulting compression forces. The hydraulic swivel motor is powered by an axial piston pump (not shown in Fig. 10) which powers the camshaft. The following signals are measurement categories: The pressure in the two chambers of the swivel motor, the gas pressure, the temperature in the air spring, the amplitude and the speed of the basement excitation, the pivoting angle of the motor and the resulting spring force. Bases on the measured oil-pressure swivel motor the driving torque is calculated roughly. The measured pivoting angle serves to calculate the radial displacements of the piston segments.

The right side of Fig. 10 shows the principle structure of the prototype. For an easier assembly and to provide a linear guiding for the piston rod, the suspension strut is built with two roller bellows. The linear guide, also a plain bearing, prevents the air spring from buckling. The top piston is equipped with the moving segments, powered by the hydraulic swivel motor. The basement excitation is applied at the bottom part of the air spring, at the external guide.

The prototype has a weight of almost 60 kg and an overall height of 1100 mm. It is clear that this prototype will never be implemented in a car or truck. But this is not relevant because the intention for this prototype is, as already mentioned, to prove the presented concept in real life and to gain more information about the behavior of the roller bellows which is successfully done.
Fig. 11 depicts first results from a measurement, compared with a simple calculation based on the equations mentioned before. The abscissa shows the piston diameter, the ordinate the resulting force of the fluid suspension system. As expected, the resulting force increases during the variation of the load carrying area.

The calculation based on Eq. 4 assumes a circular load carrying area. In fact, the real load carrying area is smaller due to the gaps between the segments. The analysis of the nonlinear FE model results in a correction term, \((-0.007D_1+1.83)\), with the diameter of the upper Piston \(D_1\). By multiplying the circular load carrying area with this area correction term (which results in the load carrying area given by the FE model), the measurement fits quite good the calculation.

**Fig. 11:** Comparison of the measurements with the calculations
Conclusion and Outlook

The first prototype of a new active fluid suspension system is introduced. The concept is a change of load carrying diameter of a hydro or pneumatic linear actuator. The presented prototype together with the discussed prove of concept is one major milestone on the way to a final active suspension system. This one should not depend on an external hydraulic infrastructure (including pump, i.e. it is “pump-free solution”). It is free of Coulomb friction and hence robust in the sense of wear. Above that the stiffening at small amplitudes which is associated to Coulomb friction is reduced. By reducing the number of parts, the robustness of the system should be increases.

There were two design challenges to meet. First a concept of a segmented piston was presented. Second a mechanical transmission was designed to drive the piston segments.

Further investigations are needed to prove the capability of the presented concept and prototype for active vibration control.

List of Symbols

\( a \) \quad \text{thermal diffusivity}
\( A, A_d \) \quad \text{load carrying area, displacement area}
\( d_C, d_W, d_{sky} \) \quad \text{damper coefficients of the chassis, the wheel and the sky-hook damper}
\( f_C, f_W \) \quad \text{chassis and wheel eigenfrequency}
\( F \) \quad \text{force}
\( g \) \quad \text{gravity constant}
\( h_d \) \quad \text{displacement length}
\( k, k_C, k_W \) \quad \text{stiffness, stiffness of the chassis suspension spring, stiffness of the wheel.}
\( l \) \quad \text{typical length}
\( m, m_C, m_W \) \quad \text{mass, chassis mass, wheel mass}
\( p, p_a \) \quad \text{pressure, ambient pressure}
\( s \) \quad \text{compression}
\( V \) \quad \text{volume}
\( z \) \quad \text{vertical coordinate}
\( z_0, \dot{z}_0 \) \quad \text{base displacement, amplitude of the base displacement}
\( \gamma \) \quad \text{isentropic exponent}
\( \omega_C, \omega_0 \) \quad \text{cut off frequency, eigenfrequency}
\( \Omega \) \quad \text{excitation frequency}

References
