Design and prove of concept of an innovative active fluid suspension system
Entwicklung und Untersuchung eines innovativen, aktiven Federungssystems

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Kurzfassung

Inhalt dieser Arbeit ist die Vorstellung des Prototyps eines konzeptionell neuen Linearaktors. Im Rahmen des Sonderforschungsbereiches 805 (SFB805) „Beherrschung von Unsicherheit in lasttragenden Systemen des Maschinenbaus“ ist der vorgestellte Prototyp das Ergebnis, die folgenden - aus der Forschung im SFB805 erkannten - Anforderungen an ein aktives Fahrwerk, d.h. der PKW Anwendung, umzusetzen:

- Reduzierte Coulombsche Reibung, keine dynamischen Dichtungen
- Reduzierte Teileanzahl und reduzierte Komplexität
- „Plug and drive“ Lösung: Anschluss an das elektrische Bordnetz im Fahrzeug.
- Trennung von Hardware und Software: Die Anforderungen des Fahrzeugherstellers oder der Kunden werden nicht über Anpassungen der Hardware sondern über Anpassungen der Software vorgenommen.

Das Grundkonzept der aktiven Luftfeder ist die Änderung der effektiven druckwirksamen Fläche A. Diese ist bei einer Luftfeder mit Rollbalg definiert über $\mathrm{A} := \mathrm{F}/(\mathrm{p} - \mathrm{p_a})$. Hierbei ist F die resultierende Axialkraft, p der Absolutdruck des Gases innerhalb des Bauteils und p der Umgebungsdruck. Die Veränderung dieser effektiven Fläche erfolgt über das radiale Verschieben von einzelnen Kolbensegmenten. Angetrieben von einem hydraulischen Schwenkmotor, werden die Kolbensegmente über Nockenwellen verschoben. Dadurch ändert sich der Durchmesser des Kolbens, was über die damit verbundene Änderung der effektiven Fläche zu einer Änderung der resultierenden axialen Kraft führt.

Der in dieser Arbeit vorgestellte Prototyp dient dem Beweis, dass ein aktives Federsystem mit dem Prinzip der veränderlichen effektiven Fläche möglich ist. Dabei dient der Prototyp der Untersuchung des Balgverhaltes sowie des Gesamtsystems.

Die vorgestellten Messungen zeigen die Umsetzbarkeit des Konzeptes sowie den Zusammenhang zwischen der Veränderung des Kolbendurchmessers und der resultierenden Axialkraft.

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”, funded 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.
- Reduced complexity
- 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 suspension strut is the dynamic alteration of the load carrying area. This load carrying area is the area A of a roller bellows and defined by $\mathrm{A} := \mathrm{F}/(\mathrm{p} - \mathrm{p_a})$. F denotes the resulting force of the strut, p the absolute gas pressure and $\mathrm{p_a}$ the ambient pressure. The alteration of this load carrying 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 load carrying area A increases and so does finally the axial compression force F.

The prototype presented in this paper serves as a demonstrator to prove 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 compression force.

1. Introduction

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) roller bellows, 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. This ratio is superior for air spring systems in comparison to other suspension systems [1].

Fig. 1 shows two principle air spring designs in a schematic drawing.

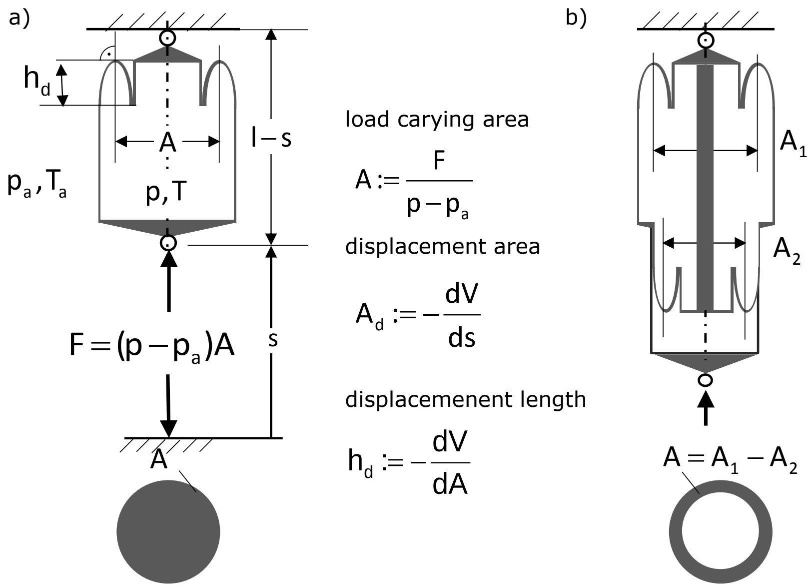

Fig. 1: Suspension strut including roller bellows: a) standard arrangement with one roller bellows and a rolling piston; b) two coaxial arranged roller bellows with the associated rolling pistons.

The double roller bellows principle (Fig. 1b)) allows a significant reduction of the load carrying area and is a promising concept for air damping solutions [2].

The load carrying area A of an air spring bellows, sketched in Fig. 1 a), is given by that diameter, were 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. 1 b). For this concept the load carrying area is given by $\mathrm{A} = \mathrm{A}_1 - \mathrm{A}_2$, where the index 1 denotes the upper bellows and the index 2 the lower one.

For all the fluid suspension systems the static equilibrium yields $\mathrm{F} = (\mathrm{p} - \mathrm{p_a})\mathrm{A}$, where p denotes the absolute pressure (gas or liquid) within the device and $\mathrm{p_a}$ the ambient pressure. This very simple relation, which in turn serves to define the load carrying area, $\mathrm{A} \coloneqq \mathrm{F}/(\mathrm{p} - \mathrm{p_a})$ allows us to discuss all fluid suspension systems in a unified manner:

With the definition of the displacement area $A_d := -dV/ds$ and the displacement length $h_d := -dV/dA$ (Index 'd' stands for displacement) and the assumption that the absolute pressure p is at least by one order of magnitude greater than the ambient pressure p_a, the relative change in the compression force becomes

$$\frac{dF}{F} \approx \frac{dA}{A} + \frac{dp}{p}. \tag{1}$$

To determine whether the dynamic change is isentropic or isothermal, the condition for isentropic change is used (see also [3])

$$\sqrt{\frac{a^2}{l^3 g}} \ll 1. \tag{2}$$

$l := V/A$ indicates the typical length of the suspension system, with the gas volume V, $a = \lambda/\rho c_p$ denotes the thermal diffusivity and g the special mass specific gravity constant.

This relation is fulfilled for most technical fluid suspension systems; hence the isentropic relation $p(\rho) = \mathrm{const}\, \rho^\gamma$ is valid for the dynamic change of state. With this, and the assumption that the change of the gas mass within the suspension strut is quasi-static in most cases (isothermal), the relative force change of the most general quasi-static fluid suspension system becomes

$$\frac{dF}{F} \approx \gamma \underbrace{\frac{ds}{V}}_{[0]} A_d + \underbrace{\frac{dA}{A}}_{[1]}\left(1 + \gamma \frac{h_d A}{V}\right) + \gamma \underbrace{\frac{dz}{V}}_{[2]} A_d + \gamma \underbrace{\frac{dx}{V}}_{[3]} A_d + \underbrace{\frac{dm}{m}}_{[4]} \tag{3}$$

In the fundamental and very use full relation (3), the relative change in the compression force is given by the five terms [0] to [4]. They represent the relative change in the compression force due to

[0] compression,
[1] the relative change in the load carrying area (topic of this work),
[2] a relative base displacement (displacement z),

[3] a relative change in the fluid volume of a hydro-pneumatic suspension system ($dV = dx\,A_d$), and

[4] the relative change of the gas mass of a gas spring (used for quasi-static leveling).

Even though the terms [2] to [4] appear harmless, the infrastructure and hardware (pump, valves, hoses etc.) required within a suspension system to gain these effects are essential.
The advantages concentrating on the change in the load 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 in future prototypes.
- A small change in the cross sectional area of the piston results in a large change of the cross sectional area and therefore in the load carrying area, especially for the double bellows solution shown in Fig. 1 (right hand side).
- The package space within the two pistons can be used to integrate the actuators.

2. Design concept of the new active fluid suspension system

Fig. 2 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. 2 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. 3.

Fig. 2 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.

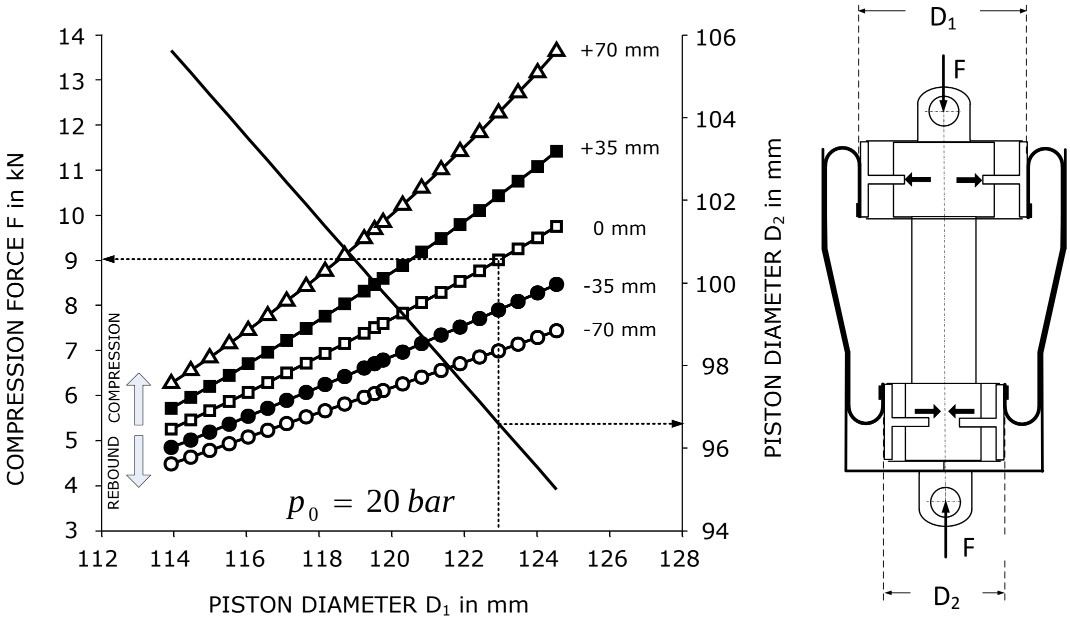

Fig. 1: Simulated compression force versus piston diameter for different suspension compression travel values s.

The white square markers in Fig. 2 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. 2 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.

3. Solving the design challenge "Change in the load carrying area of a roller bellows"

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. 3 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. Due to the pressure inside, the bellows adheres to the piston and does under no circumstances glide along the piston surface. An outward shifting of the piston segments, leads to a stretching of the bellows only between the segments. As a consequence, the bellows would be destroyed.

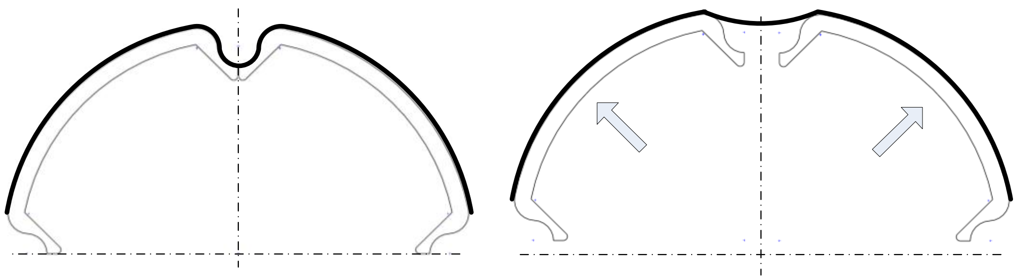

Fig. 2: Principle of the piston widening by shifting piston segments and by using a gap between the segments.

The solution to this problem is based on the special geometry of the piston surface. In Fig. 3 two piston segments are shown. A gap was put in between these two segments. The bellows 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 bellows.

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.

4. Prototype and prove of concept

The technical realization of the shifting piston segments is shown in Fig. 4. An axle, similar to a camshaft is powered by a gear wheel (Fig. 4, 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).

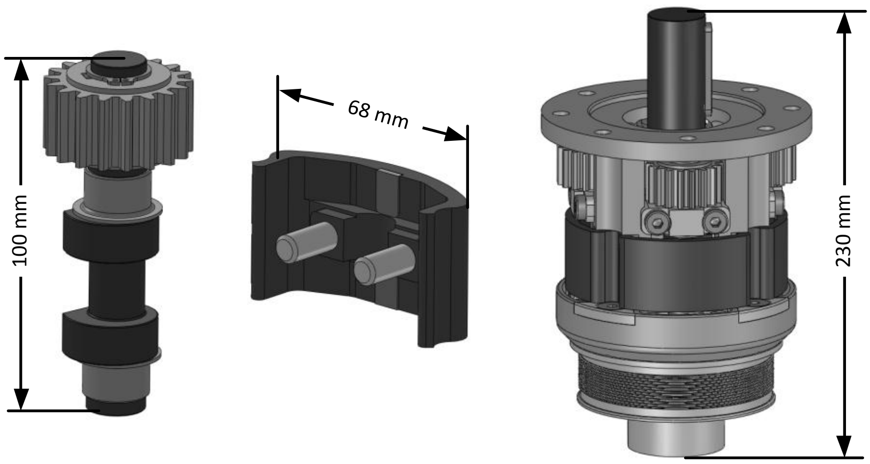

Fig. 4: Gearwheel driven camshaft (left), piston segment (middle) and complete assembled piston with four implemented piston segments.

Four of these camshafts are mounted inside the piston and are powered by one gear wheel (Fig. 4 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.

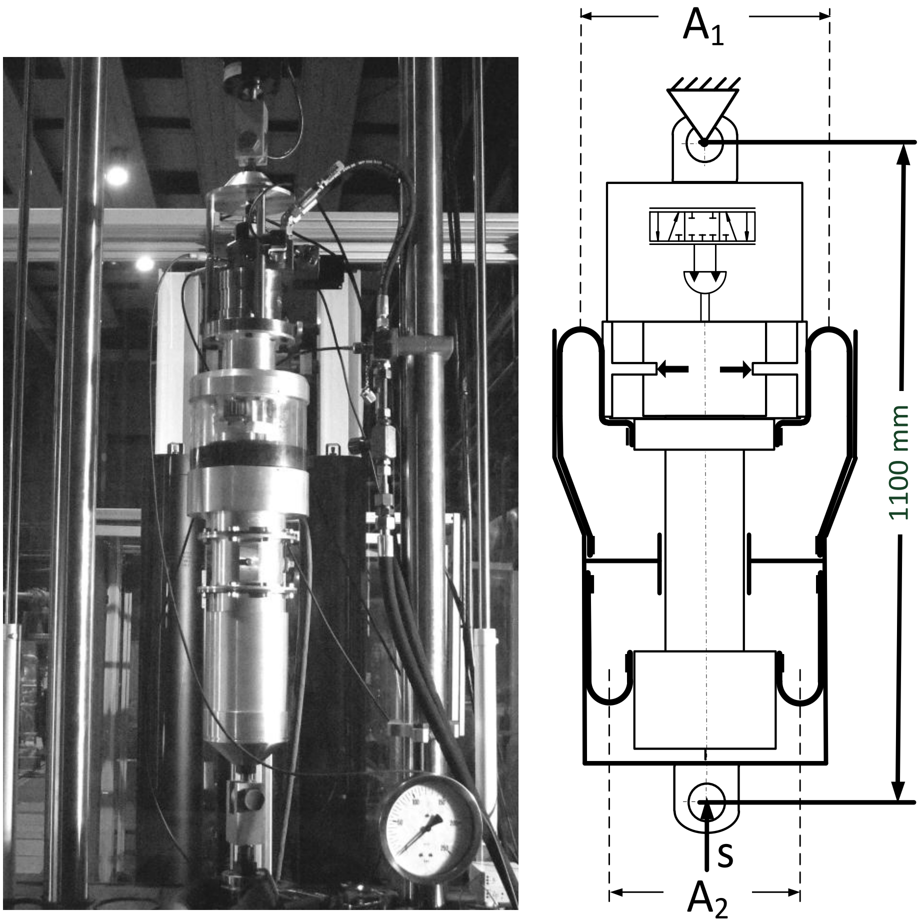

Fig. 5: Real test-bench (left hand side) and schematic diagram of the built active air spring (right hand side).

Fig. 5 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. 5) 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. 5 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.

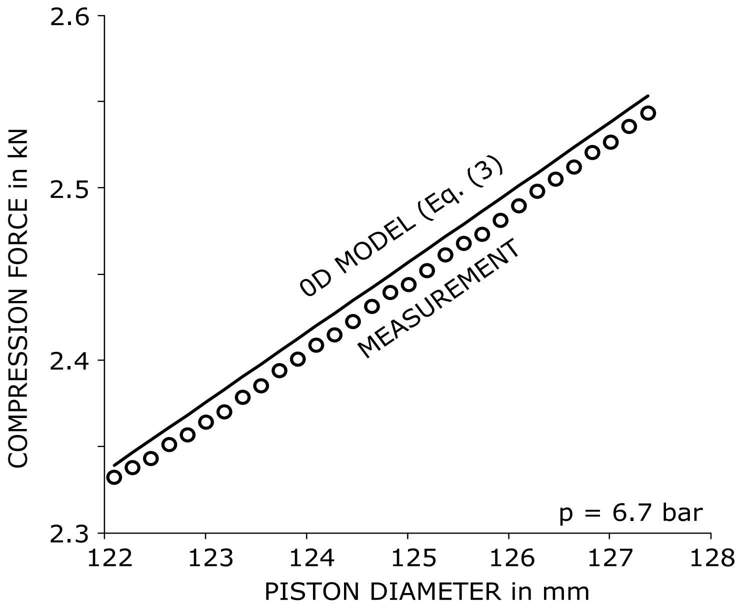

Fig. 6: Comparison of the measurements with the calculations

Fig. 6 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 measured forces almost fit the predicted forces of the very basic model (3). Due to the essential gaps between the segments, the load carrying area is smaller than the ideal axisymmetric one.

5. 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 mail stone 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.

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