Development of an active and integrated suspension system

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Abstract

The traditional solutions of active, semi active or adaptive pneumatic and/or hydraulic spring damping elements are characterized by disadvantages, such as e.g. leakage problems or the large number of components in the case of valve-based damper control or in the case of level control by means of pump or compressor. The complexity of these solutions generates uncertainties in manufacturing and function, in operation the uncertainties are generated by long force response times as well as by wear and ageing of sealing elements. The objective of the concept of this project is the development of a suspension strut and the testing of it in the overall system which does not include the a.m. components. The function of the pump is assumed by an actuator integrated in the strut which adapts the surface acting with pressure to the currently prevailing load request.

This paper presents the developed, constructed, built and tested prototype of an active fluid suspension system (HFDS): The prototype is based upon a standard air spring bellows, used in serial-production for premium car suspension systems.

KEYWORDS: Active Suspension, Airspring

1. Introduction

In vehicle dynamics, the suspension system is object of research since the beginning of modern transportation. It is part of each car, truck, van, train and airplane.

Starting with passive spring damper systems in the early 20th century, advanced by semi-active systems in the late 80’s of the 20th century, the state-of-the-art in automotive vibration control is the active suspension system, established as the Active Body Control (ABC) in 1998 by Daimler /1, 2/. Since its introduction, no real improvement occurs. BOSE for example tries to introduce an active suspension system based on electrical linear actuators for almost 30 years /3/. Recently, Tenneco came out with an active damper system, based on a super-cap-driven hydraulic pump /4/.

All active systems have in common that they are complex, expensive (especially in service) and not applicable for different customer demands. For a common vehicle platform different suspension systems are needed depending on the specifications demanded by the customer. The mentioned ABC-System for example consists of valves, pipes and hoses, filter, cooler etc.

An analysis of these vehicle suspension systems – passive, semi-active or active – in more detail results in the following uncertainties in the system's life cycle or the market whose control (or the control of their impact) is the object of research /5/:

1. **Uncertainties in operating load:** Due to uncertainties in road excitation, vehicle payload, area of application or possible misuse, the operating load can only be estimated in a wide range. Therefore, it is difficult to find an appropriate suspension setup. **Solution to control these uncertainties:** Use of an active suspension system which can adapt to different boundary conditions.
2. **Uncertainties in driving comfort and abrasive wear:** The use of dynamic sealing in today's fluid suspension systems leads to coulomb friction and abrasive wear. The coulomb friction results in a harsh suspension especially at small excitation amplitudes; the abrasive wear increases the risk of component failure and mutates the comfort relevant setup parameter too. Furthermore, fabrication tolerances have a strong influence on the quality of the seal. **Solution:** Omit dynamic sealing!
3. **Uncertainties in operational reliability:** Due to the complexity of contemporary active suspension systems, there is a higher risk of malfunction or component failure. **Solution:** Use a robust system with reduced complexity and high functional integration.
4. **Uncertainties in market needs / customer needs and Original Equipment Manufacturer (OEM) needs:** Today the customer expects a wide choice in configuration details. If the customer asks for an active suspension system, it should be provided with as little effort as possible, consuming conversion work on the car reduces the profitability. Also for different customer demands (sportive or comfortable driving style…) different characteristics for the spring – damper setup are necessary. **Solution:** Plug & Drive capability of the system for a vanishing implementation effort and characteristics setup by software, based on a common hardware.

To control the listed uncertainties in the illustrated manner, the challenge is to

develop an ***active, robust*** *and* ***highly integrated*** *suspension system with* ***no dynamic sealing****,* ***no external actuator*** *or infrastructure,* ***Plug & Drive capability*** *and* ***high flexibility.***

2. Solution: The integrated, robust and active suspension system

The best and approved way to omit dynamic sealing and increase the comfort for suspension systems is the use of a bellows /6/ as shown in **Figure 1**a. The bellows is an elastomer fibre composite with a high bending compliance /7, 8/. It is connected with the piston as well as with the cylinder and seals the included fluid volume (gas or liquid). The load carrying area *A* of a pressurized bellows is given by that diameter, where the bellows loop has a radial tangent /8, 9/ (Detailed Information: /10/). The resulting axial force is $F = (p - p_a)\,A$.

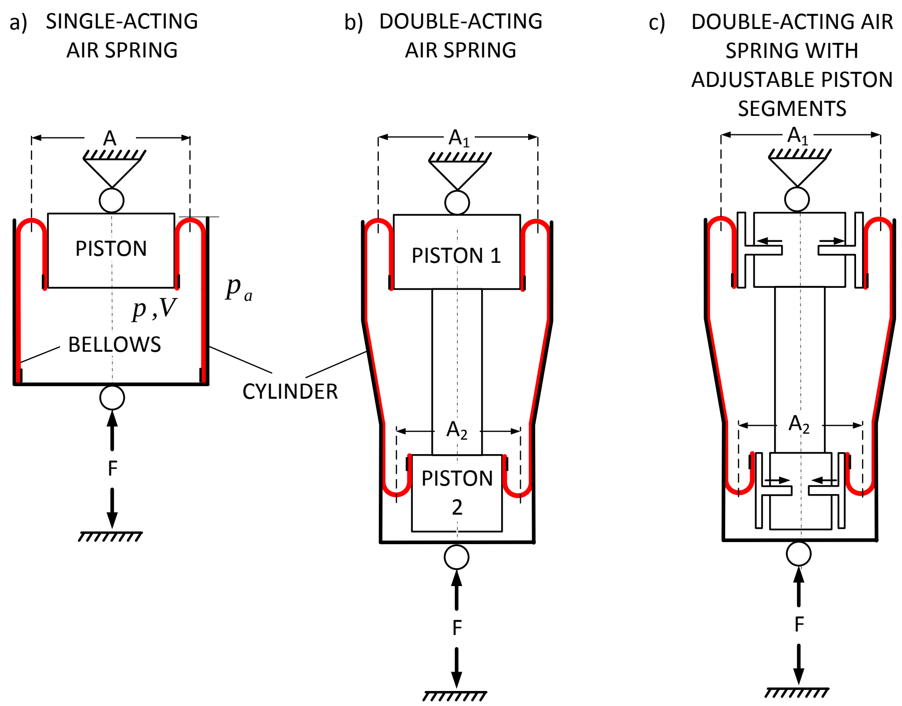

Figure 1: Single-acting air spring (a), double-acting air spring (b), double acting air spring with adjustable piston segments (c)

As shown in /5/, besides other options the best way to change the resulting force *F* is to change the load carrying area *A.* In the case of a single piston system (Figure 1a)), a change of the load carrying area requires relatively large changes in the piston's diameter. Due to the fact, that the bellows can only be stretched in a very limited range, this possibility is ruled out. Otherwise it looks with the concept of two active pistons (Figure 1b). First, the load carrying area $A = A_1 - A_2$ is significantly smaller (therefore a higher pressure is required) and small changes in the pistons' diameters result in large relative changes in the load carrying area /11/. And second, the change of the load carrying area is split; two load carrying areas (A_1, A_2) just have to be changed slightly to achieve the required change of the resulting effective area A.

2.1. Design concept of the new active suspension system

Figure 1c shows the above discussed change of the load carrying area in more detail. The pistons are divided into segments which are forced radially outwards. Due to changes in the roller fold, the load carrying areas change as well. The load carrying area A_1 enlarges its size and the load carrying area A_2 reduces its size. Hence the load carrying area $A = A_1 - A_2$ enlarges as well.

Figure 2 left hand side shows the principle effect of the alteration of the load carrying area as the result of an analytic calculation: on the abscissae the diameter of the upper piston D_{k1} is plotted, on the left ordinate the resulting compression force at constant damper compression travel and on the right ordinate the correlated diameter of the lower piston D_{k2}. The compression travel s of the suspension system is the variable parameter. In this graph, the labeled compression travel (-70 mm to +70 mm) is measured from the design position. Design position defines a specific state of the system: The absolute compression is 70 mm, the gauge pressure is 20 bars and the carried load is 7500 N. The system is designed around the operating parameters of a current Daimler S-Class with air spring suspension.

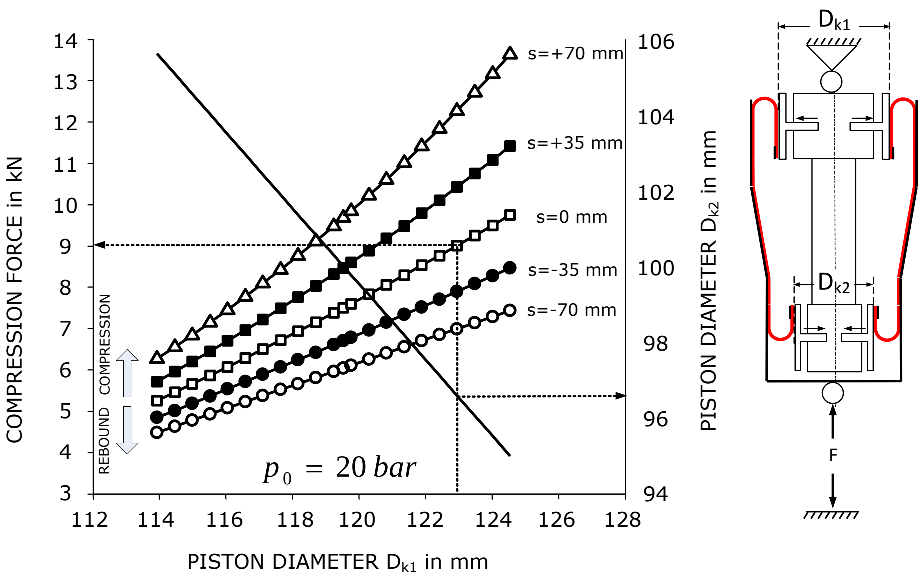

Figure 2: Calculated compression force versus piston diameter for different suspension compression travel values s.

The white square markers in Figure 2 show the change of the load carrying area in the design state. 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 Figure 2 shows: For an aspired load of 9 kN with no deflection (s = 0) the diameter D_{k1} is widened to 123 mm whereas D_{k2} is reduced to 96 mm. This spread in diameter increases with increasing compression travel *s* due to the related increase of the gas pressure.

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 /12, 13/

Figure 3 shows the radial shifting of the piston segments. The thick line symbolizes the roller bellows.

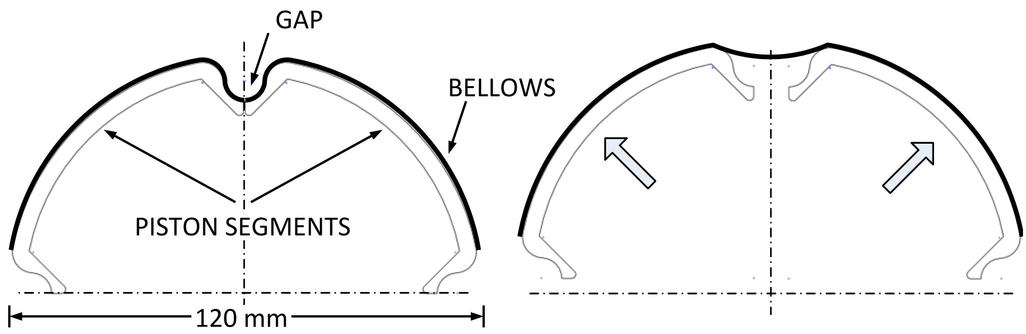

Figure 3: Principle of the piston widening by shifting piston segments and by using a gap between the segments.

The robust solution is based on the special geometry of the piston surface. In Figure 3 two piston segments are shown. A gap was put in between the two parts. 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 bellow. No gilding of the bellows over the pistons' surfaces appears and the bellows is bended but not stretched. This bending has no problematic impact on the bellows as it is the main working principle of the rolling bellow.

2.2. Feasibility study of the concept by a finite element simulation

To predict the bellows behavior, especially during segment shifting, an advanced FE-Model of the HFDS was developed and consequently enhanced for this research. With the help of the numerical model the entire assembling and usage process, starting from the installation of the roller bellows (assembling) 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. **Figure 4** shows the upper part of the implemented model at three times during the simulation of the assembling procedure. It consists of two piston segments, the piston, a symmetric quarter circle model of the bellows and a supporting cylinder.

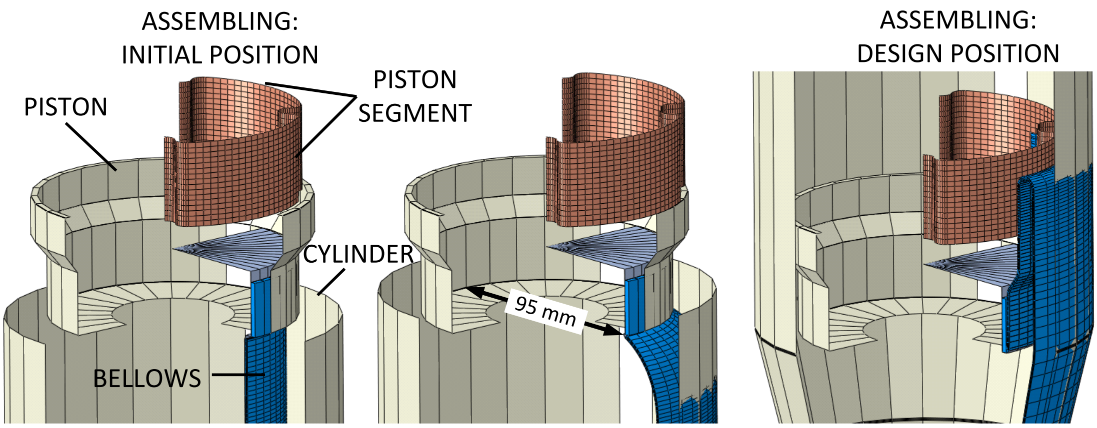

Figure 4: nonlinear finite element model of the HFDS

Several geometries of the piston segments and their interaction with the bellows were analyzed. Even though the bellows gets stretched while expanded, due to the gap between the piston segments slight compression stress occurs within the fiber. This is a very important (and unexpected) result, because of the compression stress only bellows with nylon fiber can be used. A bellows with aramid fiber would be destroyed.

After confirming the feasibility of the concept the FE-Model was enhanced to represent the entire prototype (see next section). The final model comprehends all relevant parts of the prototype: two pistons, two bellows (symmetric quarter model) two segments and two supporting cylinder.

The enhanced model was used to predict and validate the measurement results of the prototype and to refine the analytic calculations. Due to the gap between the piston segments, the load carrying area differs from the assumed area based on a circle.

2.3. Prototype and proof of concept

The technical realization of the shifting piston segments is shown in **Figure 5**. An axle, similar to a camshaft is powered by a gear wheel (Figure 5, 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 (Figure 5 left). 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 Newtons 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. Solutions for this challenge have to be developed in future work (see section 'Outlook'). The current concept mainly deals with the bellows expansion and helps to gather data and experience about the systems behavior.

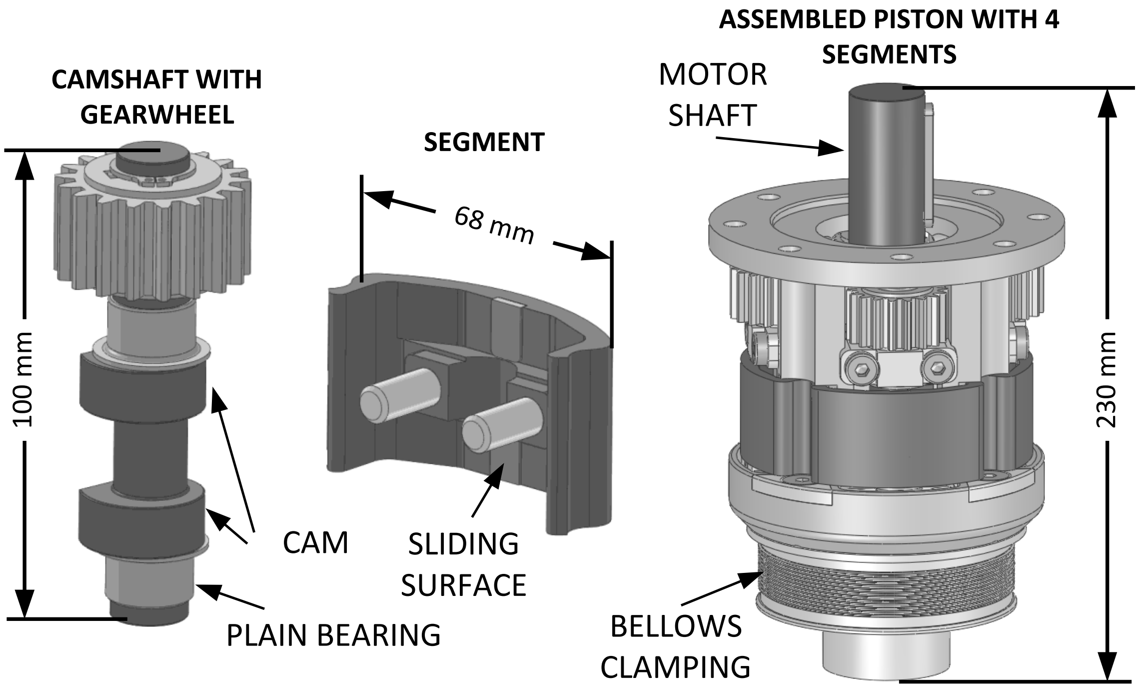

Figure 5: Gearwheel driven camshaft (left), piston segment (middle) and complete assembled piston with four implemented piston segments.

In section 'Solution: The integrated, robust and active suspension system' 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 active and equipped with shiftable segments. 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.

Figure 6 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 system is mounted into a 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 Figure 6) 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.

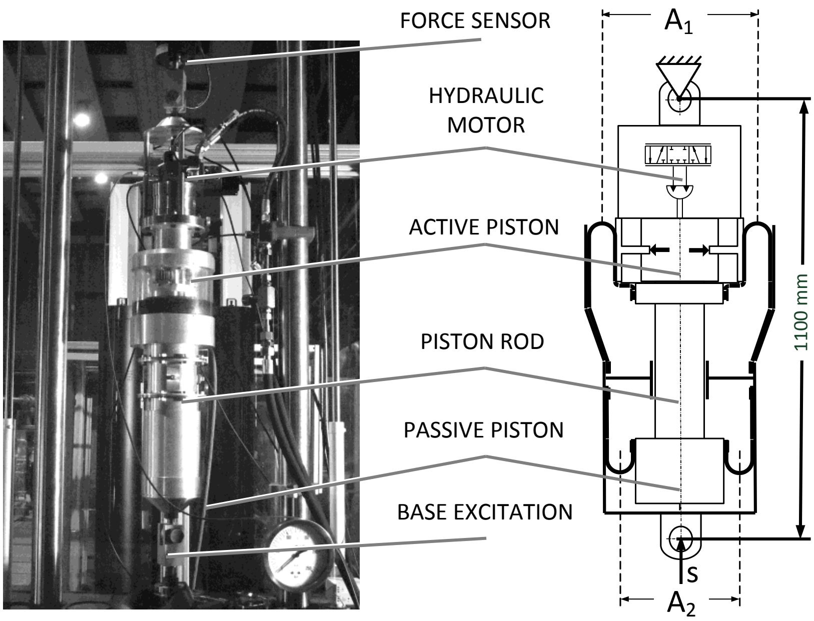

Figure 6: Test-bench (left hand side) and schematic diagram of the built active air spring (right hand side).

The right side of Figure 6 shows the principle structure of the prototype (/14,15/). 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.

3. Measurements

Figure 7 depicts results from a measurement, compared with the calculations used for the estimation in Figure 3 and with the results of a FEM analysis. 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. Due to the aforementioned assumption of a circular load carrying are in the analytical calculation, they don't fit the measurements quite well. In contrast, the results of the FEM analysis fit the measurements quite well; the difference is always less than 4%

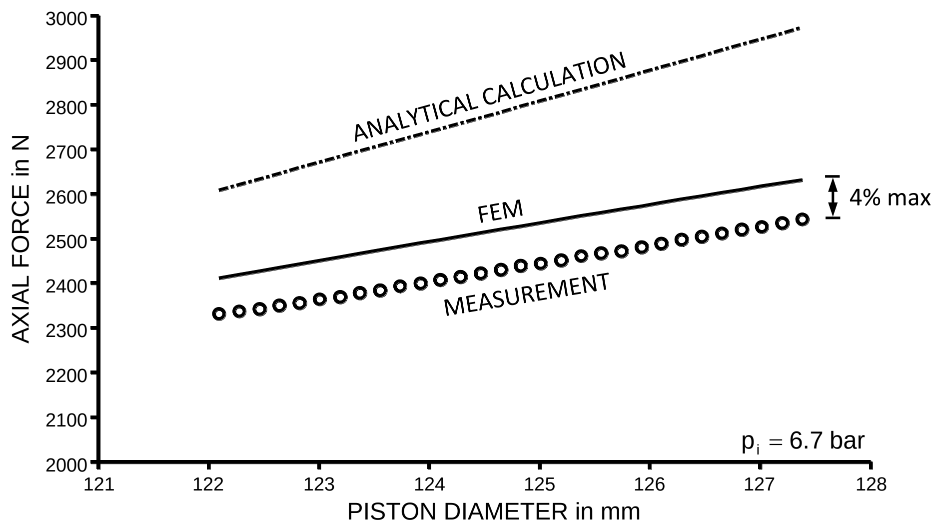

Figure 7: Comparison of the measurements with analytic calculations and the results of a FEM analysis

Ongoing experimental research analyses the capability of the HFDS for active vibration damping. Therefore, a closed loop control has to be designed, the above discussed measurement data were gathered with a simple open-loop control.

4. Acknowledgement

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5. Nomenclature

F	force	N
p	pressure	N/m²
p_a	ambient pressure	N/m²
A, A_1, A_2	load carrying areas	m²
V	volume	m³
D_{k1}, D_{k2}	piston diameter	m

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