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Keywords: vehicle, chassis, active air spring damper, fluid dynamic absorber, quarter car, vibration reduction, control of uncertainty

Abstract. This paper presents two new technologies in order to optimize the operation of a conventional spring-damper-system. Therefore, the function structure, such as the energy flow of a conventional system, is investigated and optimized. The first resulting technology is the fluid dynamic absorber (FDA), which is still a passive solution and improves the energy flow of the conventional spring-damper-system with the help of an absorber with a hydraulic transmission. The second technology is the active air spring damper (AASD), which is an active variant of a spring-damper-system and optimizes the energy flow by using electrical energy. We use a quarter car model to examine the performance of our technologies and compare them in the conflict diagram where driving comfort vs. driving safety is shown within the scope of uncertainty. The FDA improves the driving safety at almost the same comfort. The driving comfort is improved by using the AASD. We also examine the system behavior at uncertain loads. The results show that they are capable of handling this uncertainty.

Motivation

The requirements of a vehicle suspension system are to carry the load, to stabilize the body and to lead the wheel safely to reach optimal driving comfort and driving safety at minimal effort. The additional demands like minimal weight or constructed size need to be fulfilled. Generally, a vehicle consists of a body mass \( m_b \), wheel mass \( m_w \) including the axle mass, wheel stiffness \( k_w \) and a passive spring-damper-system, i.e. stiffness \( k_b \) and damping constant \( d_b \), connecting the body mass and the wheel mass. The limitation of this spring-damper-system becomes obvious when investigating the vertical dynamics with a quarter car vehicle; see Fig. 1(a). This two degree-of-freedom-system (DOF) with base excitation is a sufficient approach to examine spring-damper-systems. The operating point of this conventional system is predefined by the spring stiffness and the damper constant. It is fixed if the damper is not adjustable. To fulfill the requirements, a trade-off between driving comfort and driving safety has to be made when tuning the spring and the damper. For simplification we use the standard deviation of the body acceleration \( \sigma(\ddot{z}_b) \) to rate the driving comfort instead of the weighted vibration severity as defined in VDI 2057 [1]. The related wheel load fluctuation \( \sigma(F_w)/F_{w0} \), with the static wheel load \( F_{w0} \) and the wheel load \( F_w = k_w(z_0 - z_w) \), is equivalent to the driving safety. This is illustrated in the conflict diagram in Fig. 1(b). The Pareto front shows the limitation of the conventional topology of a spring-damper-system [2].

Our aim is to develop new and more flexible systems to connect the body mass with the wheel mass which is able to overstep the limitations of the conventional topology. The new topologies should control uncertainty, like different drivers or unknown loads, during the usage of a vehicle. That is why we develop and compare new topologies of spring-damper systems like Vergé et al. did with a hydrostatic transmission [18]. We show the capability of our new topologies to control uncertainty by simulations. By doing so, our new topologies are evaluated within the stress field of uncertain use,
availability and equipment expenditure. Methods developed by the Collaborative Research Center (SFB) 805 are used to describe and analyze the uncertainty within use. After that, the prototypes need to be designed, constructed and investigated within the real SFB Demonstrator [3].

**Introduction**

Product development aims to identify an appropriate technical method to achieve a desired function. It is necessary to verify that the solution corresponds to the customer requirements considering cost aspects, quality, reliability and so on. The overall product development process used to identify an appropriate solution is provided in single working steps to reduce complexity. Starting with an abstract product idea, possible versions of a solution are identified on varying levels of abstraction. From that point a promising solution is chosen and all attributes of the product are gradually organized right up to an entire solution [4, 5]. All essential working steps are combined in a standardized procedure model. Therefore, a well-known and accepted formulation to structure the development process is guideline 2221 [6] elaborated by Verein Deutscher Ingenieure (VDI). It can be roughly assigned to a project definition (phase 1), a variation of abstract partial solutions and their combination in an overall concept (phase 2), an elaboration of layout (phase 3) and to a specification of the product documentation (phase 4). Chiefly the procedure model of VDI 2221 constitutes a first sector-independent guideline for a development project.

**Function structure.** In the context of product development, the function structure has a great importance. It ends up with a solution-neutral description of the development task. This avoids prefixing in order to get a wide range of possible solutions. The creation of a function structure can be applied for new and adaptive designs [4]. For new designs the function structure is created for further concretion based on a requirements list and advanced further into reality during the development process. For an adaptive design a known solution is constructed as a base for creating a function structure. It is selectively varied in order to generate additional potential solutions. A function structure itself depicts the conversion of energy, material and signals within a previously determined system boundary. The description of a function structure varies in literature, because it always gives a subjective point of view on a problem [7], so it is important that the practicality is constantly in focus [4]. For this approach the generally applicable sub-functions “channeling”, “connecting”, “changing”, “storing” and “varying” by Pahl/Beitz are used [4].
Product models represent the product on a defined abstract level in the virtual product life cycle. All relevant properties and information which are necessary for the different working steps of product development are represented here [6]. In the context of the procedure model of VDI 2221 [6], product models are used to predict the product behavior or as a basis for decision-making in the development process, for example to define or verify product properties. On the one hand the creation of a product is based on ideas, assumptions, schemes or concepts on different levels of abstraction [8]. On the other hand there is incomplete information about the product and its product life-cycle, so there is a lack of information during the entire product development process. If relevant influences are ignored, only considered insufficiently or irrelevant influences are depicted during product modeling because of the lack of information, there might be a deviation between the realized and planned product behavior, so product modeling is subjected to uncertainty. According to Heinrich Hertz uncertainty in general is a lack of information [9], so relevant information for product models is partially incomplete or does not exist at all. This phenomenon can be illustrated by a map using set theory, see Fig. 2. At each step of concretion, from the relevant reality to the model and from the model to its parameters, uncertainty due to simplifications and assumptions occurs [10], so uncertainty is illustrated by a gap between the reality and the relevant reality.

SFB-805-uncertainty-model. The SFB 805 investigates uncertainty in order to be able to handle it [11]. In the context of SFB 805 uncertainty occurs in processes, so the SFB 805-process model has been developed in order to be capable of depicting and analyzing uncertainty [12]. It is based on different models like the SADT-model or the process model of Heidemann and contains a system boundary, in which a process with its initial and final state such as its influencing factors (resources, disturbances, user or information) is examined. By using semi-active or active systems there is a possibility to intervene in a process in order to react to occurring uncertainty. Based on this potential the process model has been extended in order to depict active systems and to differentiate them from semi-active and passive systems. For this purpose the appliance has been detached from the process which provides the working factor to realize them, so a fundamental aspect of the extension is the differentiation between the process itself and the product that the company produces for this purpose. Thus, the interaction between the appliance and the process can be examined. Active systems are differentiated from passive systems by the fact that they are able to provide an additional part to the existing working factor. Semi-active systems are only able to influence the appliance which has an indirect influence on the working factor.

Evaluation of three Solution Scenarios

This section is divided into two parts. The first part describes the development of two new technologies to overstep the limitation of driving safety vs. driving comfort. In the second part we investigate all three topologies and analyze their characteristics by simulation.
**Applied Product Development.** In order to uphold the driving comfort and driving safety of a spring-damper-system for an uncertain usage, the operating point has to be created flexibly. According to an adaptive design, a function structure is developed by using an existing spring-damper-system. The system boundary is drawn around the spring-damper-system and the adjusting energy flow is considered, whereby the sub-function “storing” represents the inertia and spring energy storage (Fig. 3). The mechanical energy exiting the system boundary should be preferably small and should approach zero quickly. An optimization of the existing spring-damper-system can be achieved by influencing the energy flow within the system boundary. On one hand, it is possible to extend the storing of energy in the system by adding another storage with a transmission, see Fig. 3(b). On the other hand, energy can be transferred to the system to influence the energy flow, see Fig. 3(c). Both possibilities are discussed hereafter.

![diagram](image_url)

**Fig. 3: Function structures of the suspension.**
**Fluid Dynamic Absorber.** This topology optimizes the energy flow within the system by displacing and partially dissipating the energy. The energy is displaced into a further degree of freedom and it is returned phased delayed to the main system [13]. The dissipation is realized by hydraulic losses and the tuning of the main system is not changed, i.e. \( m_b \) and \( k_b \). Hence, the dynamic vibration absorber reduces the vibrations of the main degrees of freedom [13]. The inertia of the dynamic vibration absorber could increase by a transmission. Thus, the same performance is achieved by a smaller mass. For example, the transmission ratio can be realized hydraulically, electrically and mechanically. Advantages of the hydraulic transmission are the large power density and the simple implementation. Hence, the hydraulic transmission has better availability and lower equipment expenditure.

The result of the optimization is a fluid dynamic absorber with transmission (FDA) shown on Fig. 4(a) [14]. The FDA is connected with a spring of stiffness \( k_a \) to the wheel. The piston displaces the fluid in the reservoir when it is moving. The transmission is realized by alteration of the flow cross section, thus, the flow is accelerated in the channel. The transmission increases the inertia of the fluid and dissipates energy according to the inertia and friction pressure losses. The housing of the FDA is connected with the body and is sealed. The usage process of the FDA is shown in the process diagram; see Fig. 5(a).

The main advantage of the FDA is the very low ratio of weight to inertia and therefore it is attractive for mobile applications. The disadvantage of the FDA is that it needs to be supported by an initial system, here the body, to make the transmission work. Thus, the FDA stimulates the initial system. It is not suitable to absorb the body oscillation by the FDA because in this case it has to be supported by the wheel and the wheel mass is much smaller than the body mass. Hence, the wheel cannot be used as initial system.

**Active Air Spring Damper.** Another approach to overstep the conflict between driving comfort and driving safety is to use an active strut to apply forces during the usage. By using an active system one is able to influence the process and can thus respond to uncertainty in usage, such as unknown load or excitation. The system is adaptable and its operation range is more flexible.

The only fully active suspension system available on the market is the Magic Body Control by Daimler, the successor of the Active Body Control which was launched in 1999. A hydraulic piston in series with the steel spring applies forces with a frequency up to 5 Hz [15]. Due to the greater spring
stiffness needed, the driving comfort at higher excitation frequencies is poor. Therefore we take a
different approach and develop an active air spring damper (AASD). Motivation is to combine the
advantages of an air spring damper, such as very good driving comfort and automatic level control,
with those of an active system.

By altering the load-carrying area of the air spring during usage, axial tension and pressure forces
are applied as shown in Fig. 4(b). The load-carrying area of an air spring with outside guiding of the
bellows is a circle with the diameter \( d_A = \frac{1}{2}(d_p + d_o) \), where \( d_p \) denotes the piston diameter and \( d_o \)
the outside guide diameter. The piston diameter is changed with four radially positionable segments
which are evenly distributed along the circumference of the piston. We use a double piston air spring
(damper) shown in Fig. 4(b) with a ring circular load-carrying area. The advantage is that only small
changes in the diameters \( d_{p1} \) and \( d_{p2} \), always in opposite directions, are needed for large relative area
changes. We already proved the operational capability of the active air spring damper experimentally
[17]. An air spring damper combines the functions of a spring and a damper in one component. The
oscillation energy is dissipated when compressed air flows through an orifice from one chamber into
another. Due to its working principle the air damper is damping frequency specific. Normally the
orifice is adjusted so that the damping energy reaches its maximum at the wheel natural frequency.

The usage process of the air spring damper is shown in the process diagram, see Fig. 5(b). The
entire system consists of the air spring damper itself, the hydraulic pump, the hydrostatic transmission,
the controller unit and displacement and acceleration sensors. Electric power enters the system and
is transformed into hydraulic power in the pump. The hydrostatic transmission transmits the power
to the air spring damper where it is used to alter the load-carrying area. Its wear can be minimized
by an algorithmic structure synthesis as shown by Vergé et al. [18]. The controlled variables are the
body acceleration, the wheel load fluctuation and the compression of the active strut \( \Delta z \). The available
electric power \( P \) is another important value. The increased complexity and effort of this active system
compared to the passive solutions becomes obvious based on the numbers of components, but we will
not address this issue in great detail.

Fig. 5: Process diagram of the topologies according to [16].
Simulation. To identify the potential and performance of our new topologies, we use a quarter car model (two-mass system). Although this model is restricted due to the neglected effects of coupled masses [2], it is sufficient for our purpose. We simulate a ride on a very poor road (class E according to ISO 8608 [19]) at a velocity of 54 km/h. The parameters of the quarter car correspond to those of a typical middle class car with a conventional suspension consisting of a linear spring and damper (body mass $m_b = 290$ kg, wheel mass $m_w = 40$ kg, body damping rate $d_b \approx 1100$ Ns/m, body stiffness $k_b \approx 20$ kN/m and wheel stiffness $k_w = 200$ kN/m). In a second step we examine the robustness of the suspension system with regard to uncertain loads (body masses of 100 and 290 kg). We use zero-d-models (lumped parameters) to model the FDA and the AASD. The model of the FDA consists of conservation equations of mass and vertical momentum. This set of equations is solved numerically in MatLab. The FDA is added in parallel to the conventional suspension system and therefore the damping rate is not changed. The object-oriented language Modelica is used to model the air spring damper [20]. The conservation equations of mass, vertical momentum and energy are solved. This model is integrated as a system function (s-function) in MatLab Simulink where the hydraulic actuator and the controller are modeled. The hydraulic actuator and transmission is modeled as a first-order lag element with a cutoff frequency of 5 Hz. To control the active system a simple PID-controller is used. The desired value is the body acceleration $\ddot{z}_b$.

![Conflict diagram with all three topologies at the nominal body mass of 290 kg.](image)

Fig. 6: Conflict diagram with all three topologies at the nominal body mass of 290 kg.

Each topology is simulated and analyzed. The results are shown in the conflict diagram, Fig. 6, which was introduced in the motivation. This diagram shows the conflict in finding the optimal setup for a spring damper system. It does not only depend on the spring stiffness and damping constant but also on the road and driving velocity [21]. The conflict diagram shows the body acceleration fluctuation $\sigma(\ddot{z}_b)$ over the wheel load fluctuation $\sigma(F_w)$ with respect to the static wheel load $F_{w0}$. The acceleration presents the intensity of vibrations. Therefore low body accelerations are equal to a good driving comfort. The driving safety is mainly dependent on the wheels remaining in constant contact with the road. This can be determined by the wheel load fluctuation. The driving safety is good if it is small.

The pareto line represents the limit of the convectional system. It can only work above this line. The new topologies meet our expectation and overstep the limitation by the pareto front of the conventional topology.

During the usage of the spring damper system many influences on the systems are uncertain but we only deal with an uncertain load for now. The change in load, which is equivalent to different body
masses, has a large impact on the system’s behavior and performance. The results are illustrated in the two conflict diagrams for a body mass of 100 kg and 350 kg, see Fig. 7.

![Conflict Diagrams](image)

Fig. 7: The conflict diagrams for a body mass of 100 kg (a) and 350 kg (b).

**Discussion**

Analyzing the presented results, there are four facts to mention:

1. The new topologies overstep the limitation of the conventional topology.

2. The FDA increases the driving safety and reduces the comfort only a little bit. The latter is substantiated by the fact that the FDA is supported by the body. This becomes obvious for the lighter body mass. The stimulation of the body is reduced by scaling down the damper constant $b_n$, i.e. to 50%. By doing so, the comfort is similar and the driving safety is 12% better than using the conventional topology.

3. The advantages of the active system become obvious in the conflict diagram. The driving comfort is better than that of the passive ones for all three body masses. Even the driving safety is better for the body masses 290 kg and 350 kg but not for 100 kg. This is due to the great impact of the controller on the system performance. The simple PID-controller provides good results for great body masses, but not for the small one.

4. The equipment expenditure of AASD is higher than that of the passive topologies. This is essentially because of the controller and the complex hydraulic actuator.

**Conclusion**

We presented two new topologies to overstep the limitation in driving comfort and safety of a conventional spring-damper-system by analyzing the function structure of the limited conventional system. By analyzing and improving its function structure we derived our new solutions. One of our topologies is passive, the so-called fluid dynamic absorber (FDA), an absorber with a hydraulic transmission. The other topology is active, which means external energy is used to reduce vibrations. The active solution is an active air spring damper (AASD).

To investigate the potential of our approaches we simulated the vertical dynamics of a car with a quarter car model, a two-mass-system, with parameters of a typical middle class car. We calculated the standard deviation of the body acceleration, i.e. the driving comfort and the wheel force deflection,
i.e. the driving safety for a ride on a very poor measured road (road class E according to ISO8608 [19]) at a velocity of 54 km/h. We showed that we exceeded the limit of the conventional suspension system, which is represented by the pareto front in the conflict diagram where driving comfort vs. driving safety is illustrated. The FDA improves the driving safety at almost the same comfort. The driving comfort is improved by using the AASD. We also examined the system behavior at uncertain loads. The results show that they are capable of handling this uncertainty. Additionally, the air spring damper has the advantage that the full suspension travel is always available due to the adjustment of the initial pressure to the load.

In a next step we will build the FDA and a second prototype of the AASD to characterize them experimentally on our servo-hydraulic damper test system and to perform real time hardware-in-the-loop tests. Hardware-in-the-loop means that we have a real strut on the test rig interacting with the simulation of the virtual quarter car running on our dSpace system. After that we will test our technologies in our SFB-demonstrator. Based on these tests we will compare our two new topologies regarding performance and effort within the scope of “active vs. passive”. Besides this, the assumptions of the product models, on which the comparison between both solutions for a more flexible operating point is based on, has to be considered. So the uncertainty in modeling has to be analyzed. Therefore, a systematization of the modeling process such as the related assumptions has to be formalized in order to make a statement to the causes of modeling uncertainty. With the help of that, modeling uncertainty can be integrated into the comparison of both solutions. Finally, the question of whether the active or the passive solution is best for a suspension system can be answered.

Acknowledgment. We would like to thank Deutsche Forschungsgemeinschaft (DFG) for funding this project within the Collaborative Research Centre (SFB) 805.

References


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