



Integrated Fluid Dynamic Vibration Absorber for Mobile Applications

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The setup of a suspension always leads to a compromise between comfort and safety. In order to counteract this in a passive approach, one could attach a structural extension in the form of a dynamic vibration absorber to the axle. Thus, energy of the wheel vibrations is diverted into the vibration absorber instead of the body. In comparison to a classic dynamic vibration absorber, which is not in the sense of lightweight construction due to the additional mass, our Fluid Dynamic Vibration Absorber (FDVA) reduces the dynamic mass by using a hydrostatic transmission.

Keywords: dynamic vibration absorber, driving safety, suspension strut, hydraulic transmission

Target audience: mobile hydraulics, automotive, hardware-in-the-loop

1 Introduction and Motivation

The essential requirements for a suspension strut of modern vehicles are high driving safety and high driving comfort at the same time. The control of the driver over the vehicle can only be ensured if there is a contact between the wheel and the ground at any time. Because only the tire has contact to the road, the vertical force of the suspension strut has to be sufficient to ensure the best transmission behavior of horizontal maneuvers /1/. According to Mitschke /2/, the dynamic wheel load fluctuation is a measure of driving safety. The driving comfort is influenced decisively by acceleration acting on the occupants. In reality, the evaluation of driving comfort is very complex and can be evaluated as better or worse by the personal feeling of the occupants. The driving comfort is expressed in simplified terms by the effective value of the amplitude of the vertical acceleration of the body. The suspension strut of a vehicle is the connection between wheel and body, thus affects both driving safety and driving comfort. The setup of a classical suspension system with spring and shock absorber leads to a compromise between driving safety and driving comfort. This compromise becomes obvious in the so-called conflict diagram, where the fluctuation of the body acceleration is plotted over the relative wheel-load fluctuation, Figure 1. The solid lines in this figure represent a constant damping, the dashed lines constant spring stiffness.

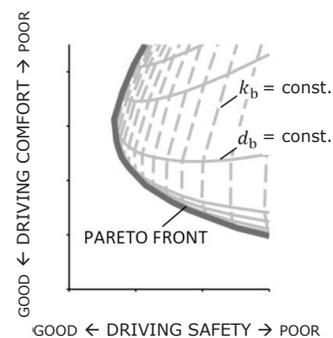


Figure 1: Conflict Diagram for the setup of passive suspension system /3/.

Moving to a higher solid line in the conflict diagram, the damping rate increases. A higher damping rate leads to a higher driving safety but on the other hand, it reduces the driving comfort because shocks are transmitted more strongly from tire to body. Therefore, the Pareto line, which represents the absolute optimum, cannot be undercut by any combination of damping coefficient and spring stiffness for this system.

There are several approaches to shift the Pareto line. On the one hand, there are active systems that adapt the force transmitted by the wheel, such as the active air spring /4/ or Daimler's ABC system /5/. On the other hand, there is also the possibility of using passive systems with different structures.

One well-known structural extension of a passive system is a vibration absorber, which is capable of reducing vibrations of the wheel in a defined frequency band without affecting the body acceleration. This is realized by redirecting the energy of the accelerated wheel into the structure extension.

A further aim is to make the dynamic vibration absorber as light as possible in order to follow the trend of lightweight construction and not to attach unnecessarily high additional masses to the vehicle.

In this paper, we examine how a dynamic vibration absorber, which reduces the heavy mass of the absorber by means of a hydraulic transmission at the same absorber inertia, can be integrated into the suspension strut /6, 7, 8/. Furthermore, we show a mitigation of oscillations by using the fluid dynamic vibration absorber.

For this purpose, firstly the theory of the Fluid Dynamic Vibration Absorber (FDVA), a vibration absorber with a hydraulic transmission, is shown. Secondly, we describe the structure of the functional demonstrator and the experimental set-up (component and HiL tests), which we use to validate the theoretical model. Thirdly, we discuss our results and give a short outlook.

2 Theory

A vibration absorber consists of a capacitance and an inertia and is used in many systems to reduce vibrations. In the case of a mechanical vibration absorber, a comparatively small mass is attached with a spring to the oscillation system. This structural extension takes the energy out of the oscillating basic system, in our case the wheel. Due to the extended Den-Hartog principle, the natural frequency of the dynamic vibration absorber in a suspension strut has to be close to the natural frequency of the wheel /9/. An additional damping of the vibration absorber enables its usage over a wide frequency range but also reduces the vibration isolation in the natural frequency. The vibration energy is dissipated in the damper. Damping is therefore desirable, as long as the system is not only operating stationary in the natural frequency of the vibration absorber. The transfer functions for a vibration absorber with and without damping are shown in Figure 2.

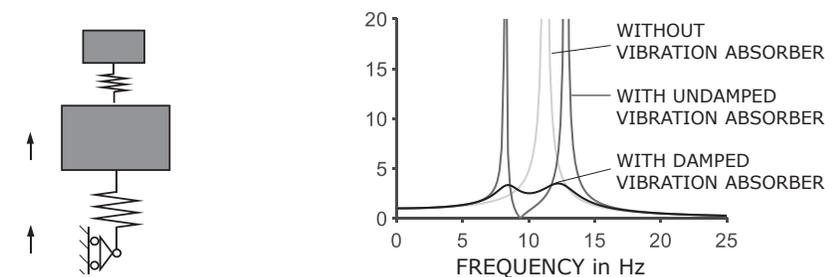


Figure 2: Single mass system with a vibration absorber amplitude and its transfer function.

The closer the absorber mass is to the system mass, the more energy can be absorbed /10/. On the contrary, a higher absorber mass is in conflict with the sense of lightweight construction. This conflict can be overcome by using the

FDVA with a translated absorber inertia /8/. Figure 3 shows in the first column a conventional absorber where the heavy mass is equal to the inertia. In the next column, a fluid mass replaces the majority of the solid mass of the absorber. This does not provide a benefit in comparison to the first column. Heavy mass and inertia $m_K + m_{F,L}$ are still equal. The right-hand column shows a dynamic vibration absorber with a hydraulic transmission. The transmission ratio between the piston and the channel surface is $\alpha = A/a$. Only the translation of the liquid leads to an increase in inertia, which, with low piston mass, is in the order of magnitude of α^{-2} /8/. Additionally, damping is directly integrated in this type of vibration absorber. The oscillation energy is dissipated due to hydraulic losses. The casing of the FDVA needs to be mounted on an inertial system with high inertia. In our case, only the body mass of the vehicle is suitable.

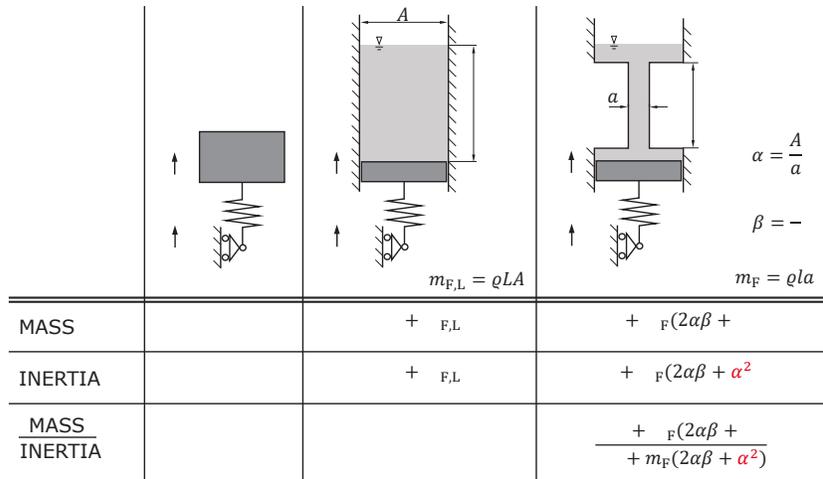


Figure 3: Principle of hydraulic transmission of the absorber inertia.

Summarized, the FDVA is able to absorb oscillation energy of the wheel by adding only small extra masses to the system. The FDVA is used in the suspension strut parallel to the spring and damper. Hydromounts have a similar operating principle /11/. However, one has to mention, that oscillations are transferred from the FDVA to the car body and the driving comfort could be decreased.

3 Experimental Set-Up

The experimental set-up is divided into two aspects, component test with the functional prototype of the FDVA and Hardware-in-the-Loop (HiL) test to examine its performance in a virtual oscillating system. The functional demonstrator is designed for validating our axiomatic model.

3.1 Functional Demonstrator

The functional demonstrator is adapted to the dimensions of the installation in the SFB demonstrator, a load-bearing structure for the examination of uncertainty /3/, and is no optimized solution for a vehicle installation. The rod end of the FDVA on the left-hand side in Figure 4 is connected to the wheel axle. At this point, the vibrations are introduced into the absorber. The force is transmitted to the piston rod via two compression springs that are tensioned against each other. The mutual tension allows oscillation in the z-direction. Springs with four stiffnesses from 302 N/mm to 63 N/mm can be used for the tests.

The piston rod protrudes on both side of the hydraulic volume to keep it constant indecently of the strut deflection. A piston, which is linked to the rod and driven by the spring force, presses the oil of one chamber through ducts into the other chamber. Thus, the hydraulic transmission consists of a double-acting hydraulic cylinder whereby the chambers are connected via several ducts on the outer side.

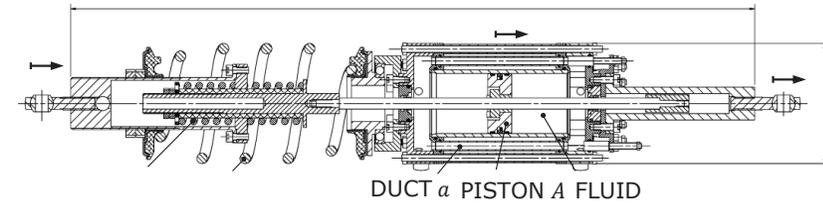


Figure 4: The TU Darmstadt Fluid Dynamic Vibration Absorber with parallel-connected body spring.

The ducts can be closed with mechanical valves in order to change the ratio $\alpha = A/a$ between the surface of the piston A and the ducts a. All possible set-ups of transmission ratio and inertia are shown in Table 1. The moving oil mass in the two chambers, as well as the hydraulically translated oil mass in the ducts, create the absorber inertia.

Opened ducts	1	2	3	4	5	6	7	8	9	10	11	12
α	67.0	33.5	22.6	16.9	13.5	11.3	9.7	8.5	7.5	6.8	6.5	5.6
θ in kg	44.3	23.0	15.8	12.3	10.1	8.7	7.7	6.9	6.3	5.9	5.5	5.1

Table 1: Parameter of the hydraulic transmission.

The body spring is connected in parallel to the FDVA and links the wheel axle to the housing of the FDVA, which in turn is attached to the chassis.

3.2 Test Framework

To validate the axiomatic model, the simulation results are compared with the measurements of the functional demonstrator. All measurements are performed on a servo-hydraulic testing machine at harmonic and stochastic excitations at varying frequencies. The procedure of the measurements starts with the determination of the absorber damping depending on the number of open channels. For this measurement, the hydraulic transmission is examined separately, as shown in Figure 5 on the left. The piston rod is driven directly and not with the absorber springs. It is excited at a constant amplitude of 25 mm with frequencies from 0.06 to 3.2 Hz. The frequencies are adapted to the maximum speed, which occurs in the following tests. This measurement is performed systematically with all setups for the ducts. The duct surface $a = na_0$, increases with the number n of opened ducts a_0 . The damping constant is used for a simplified single-mass oscillator simulation, which in turn is compared with measurement results of the FDVA in order to describe the natural frequency of the FDVA. For this measurement, we examine the complete FDVA (Figure 5 right) at harmonic excitation in the frequency range from 0.1 Hz to 25 Hz at an amplitude of 4 mm.

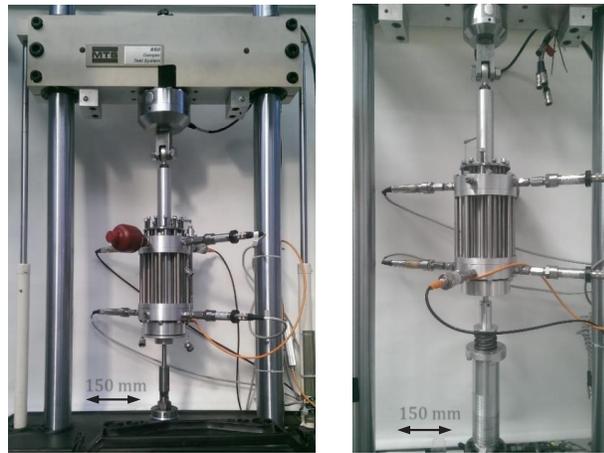


Figure 5: The FDVA on the test damper machine. Damper measuring (left), natural frequency and HiL Measurement (right).

parameter	value
piston surface A	4300 mm ²
duct surface a	63.6 mm ²
chamber Length	77 mm
duct length	170 mm
oil density ρ	880 kg/m ³
piston mass	1 kg
fluid mass m_F	0.0095 kg
absorber spring stiffness k	301 N/mm

Table 2: Parameters of the functional demonstrator.

Only measurements with one and two open ducts are carried out here, since the natural frequency of the vibration absorber with the used spring with three open channels is already 25 Hz and outside of the validity range of the quarter car model. A higher number of open channels makes sense only when using the softer vibration absorber springs. The parameters of the functional demonstrator are listed in Table 2.

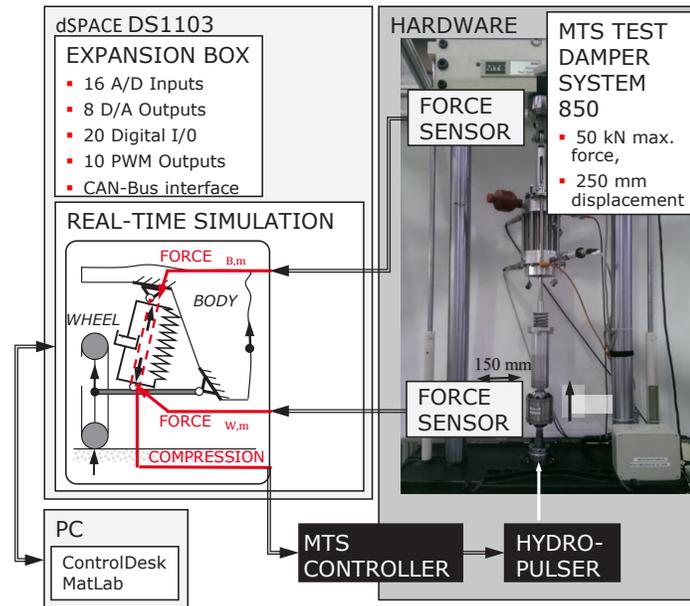


Figure 6: Basic structure of the HiL tests with the FDVA (with reference to /4/).

Subsequently, the FDVA is examined in a HiL simulation. Figure 6 shows the basic structure of the HiL test. The measurements are carried out with a dSPACE-System, on which the real-time simulation of a two-mass oscillator (Figure 7) is running. The parameters of the dual-mass oscillator are listed in Table 3.

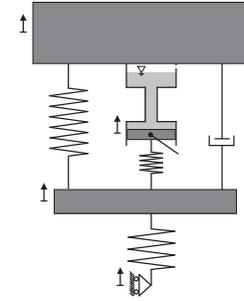


Figure 7: Two-mass oscillator model with FDVA.

Parameter	Value
body mass	290 kg
wheel mass	40 kg
body spring stiffness	19.7 N/mm
tire stiffness	200 N/mm
body damping	1170 Ns/m

Table 3: Parameter of the virtual two-mass oscillator.

The excitation z_0 with frequency shares up to 25 Hz corresponds to a drive over a federal highway (Bundesstraße) at a driving velocity of 100 km/h /4/. The term “wheel mass” describes the unsprung mass, thus parts of the suspension, brake and the wheel /3/. With this excitation and the measured forces at the FDVA, the simulation model calculates the compression of the strut. This signal is transmitted to the MTS test damper system, which applies the actual compression. The simulation model is based on the equation

$$\begin{bmatrix} m_B + \alpha^2 m_F & -\alpha(1 + \alpha)m_F \\ -\alpha(1 + \alpha)m_F & [m_p + \alpha + \alpha m_F] \end{bmatrix} \begin{bmatrix} \ddot{z} \\ \dot{z} \end{bmatrix} + \begin{bmatrix} -B & -d_B \\ -B & B \end{bmatrix} \begin{bmatrix} \dot{z} \\ z \end{bmatrix} + \begin{bmatrix} k_B & -B \\ - & +k \end{bmatrix} \begin{bmatrix} z_W \\ z \end{bmatrix} = \begin{bmatrix} -p_v \alpha \alpha \\ k \\ p_v \alpha \alpha \end{bmatrix} \quad (1)$$

where p_v is the pressure losses.

4 Results

The simulation results of the FDVA show a significant decrease of the wheel oscillation at almost constant body oscillation, Figure 8. The transfer function is defined by $V(x) = \hat{x}/\hat{z}_0$ in the following section.

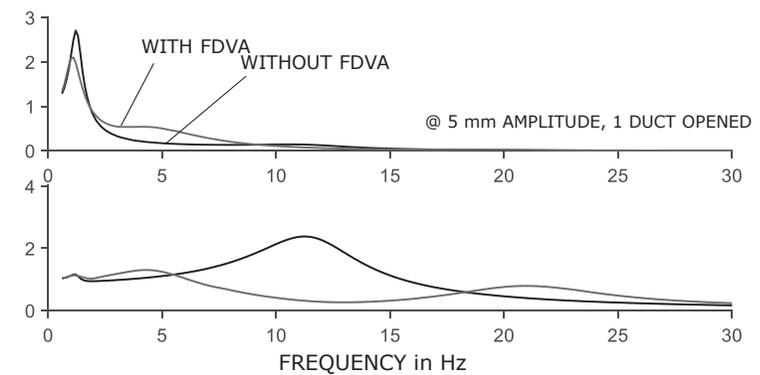


Figure 8: Simulation results of a quarter vehicle model at a harmonic excitation of 5 mm amplitude.

The measurements are used to validate the virtual model of the FDVA. The results are divided into damping, natural frequency and HiL-Simulation.

4.1 Damping of the Hydraulic Transmission of the FDVA

An important component of the FDVA is the hydraulic transmission. This acts almost like a damper. In contrast to a damper, the FDVA has a significantly larger inertial mass and therefore an additional force acts at the reversal point of a harmonic excitation, as shown exemplary in Figure 9. For the calculation, the body motion is set to zero and the force on the body is calculated using the motion equation of the body. It follows the force on the body is

$$= -\alpha(1 + \alpha)m_F\ddot{z}_A + p_v\alpha a, \quad (2)$$

where \ddot{z}_A is the acceleration of the excitation z_0 . The measurement results fit qualitatively and quantitatively well to the simulation results, whereby Coulomb's friction was not taken into account in the simulation, which is why the curve of the simulation is not widened at the turning points. The Coulomb's friction, which is identified experimentally, is approximately 30 N.

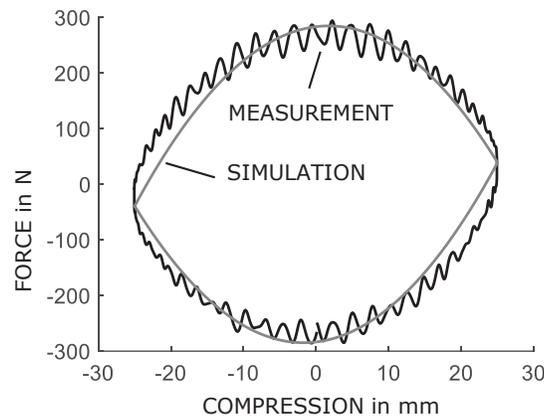


Figure 9: Body force of the FDVA at a harmonic amplitude of 25 mm at 1 Hz.

The harmonic excitation at 1 Hz and 25 mm amplitude corresponds to the maximum velocity and thus forces occurring in the HiL test. Consequently, it can be shown that the attenuation in this range can be described with the simulation model. The oscillations of the measurement result from a measuring noise, as well as the control of the testing machine.

4.2 Natural Frequency of the FDVA

To ensure that the hydraulic transmission is working properly, the FDVA is regarded as a single-mass oscillator with excitation via the spring, shown in Figure 10 /12/.

The damping constant $b = 2900$ Ns/m for one open duct is determined from the measurements in section 4.1 by a first-order polynomial. The transfer function for this case is

$$(\eta, D): = 1/\sqrt{(1 - \eta^2)^2 + 4D^2\eta^2}, \text{ with } D: = b/2\omega_0 m, \quad \eta: = \Omega/\omega_0 \text{ and } \omega_0 = \sqrt{k/m} \quad (3)$$

according to Magnus /11/. The phase is calculated with

$$\tan\psi = 2D\eta/(1 - \eta^2). \quad (4)$$

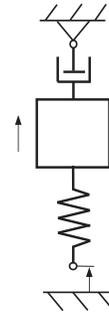


Figure 10: Single-mass oscillator with base excitation.

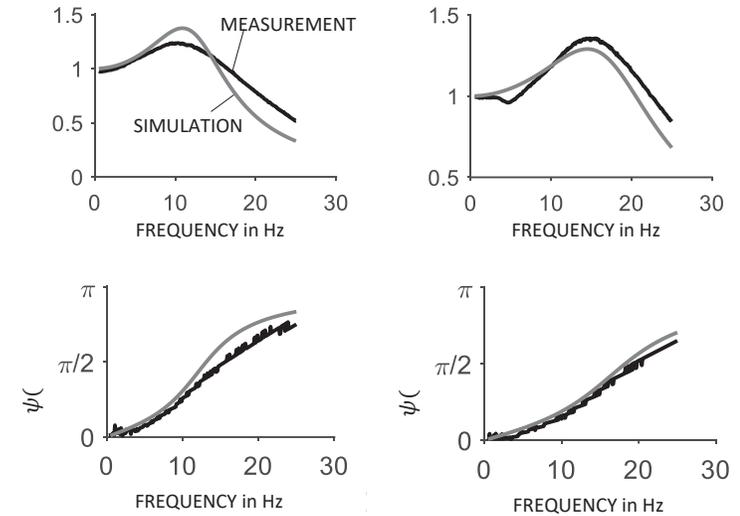


Figure 11: Transfer function of the FDVA with one opened duct.

Figure 12: Transfer function of the FDVA with two opened ducts.

The natural frequency of the analytical solution is shown at the intersection with the phase shift of $\pi/2$ and also at maximum of amplification function. Figure 11 shows the measured values in comparison to the analytical model for one opened duct. With a parameter identification, the natural frequency of the FDVA can be estimated for the phase shift of $\pi/2$. At $f_{0,real} = 15$ Hz, the measured natural frequency is slightly higher than that of the analytical model, which is designed for a natural frequency of $f_{0,sim} = 13.1$ Hz. The higher frequency $f_{0,real}$ can be explained by the fact that Coulomb's friction is not taken into account. The difference in shape of the curve, especially for the amplification, can also be explained by the non-linear friction, but has no influence on the determination of the natural frequency by means of the phase shift.

Figure 12 shows the same curves as Figure 11 for 2 opened ducts. By opening a second duct, the inertia changes as shown in Table 1. With a lower inertia, the natural frequency increases. This behavior can also be well reproduced.

The equation for the natural frequency can be used to calculate the existing inertia, which leads to an inertia of $\theta_{real,1 \text{ duct}} = 33.1$ kg for one open channel and $\theta_{real,2 \text{ ducts}} = 19.1$ kg for two open channels. In this case, the simulation value is higher. Further investigations of Coulomb's friction and leakage currents at the piston seal must be carried out. The real functional demonstrator does not quite reach the simulated value, but is much larger than the heavy mass of $m = 1.6$ kg. The inertia thus is translated hydraulically.

4.3 HiL Simulation Results of the Functional Demonstrator

The vehicle is modeled virtually in the real-time simulation environment and coupled with the real FDVA on the test damper system. This makes it possible to examine a component independently of other components in the vehicle. The measurement of the FDVA in the HiL simulation not only provides results for the actual component but also for the quarter vehicle. This makes it possible to show the structure vibrations, wheel vibrations, as well as body acceleration and wheel load. The results for a drive over a federal highway described by Hedrich /4/ can be seen in Figure 13. The considered quantities are related to the excitation.

The two graphs show a particular agreement in the natural frequency range of the FDVA. Outside this range, deviations can be detected, but no model adjustment has yet been carried out. Remarkable are the better results of

the function demonstrator in comparison to the simulation, which is especially visible in the wheel vibrations. The component measurements, as well as the comparison of the HiL results validate the axiomatic model.

However, it is not only the validation that is of interest, but also the effect of the FDVA on the driving behavior. In addition to the graphs with FDVA, a graph for a simulation without FDVA is also shown. The properties of an absorber are as expected. With excitations in the natural frequency of the absorber, a clearly better behavior is evident in all four transfer functions. However, outside of a certain range, which is in case of an open channel between 10 Hz and 19 Hz, a worse behavior is to be found, determined by a larger gain. The maximum of the acceleration amplification of the body is reduced due to the additional damping of the FDVA. With the built-in FDVA, however, the accelerations increase again compared to the configuration without FDVA as soon as the excitation exceeds a frequency of more than 1.6 Hz. Looking at the behavior of all four transfer functions, an improvement can only be seen so far in a small frequency band.

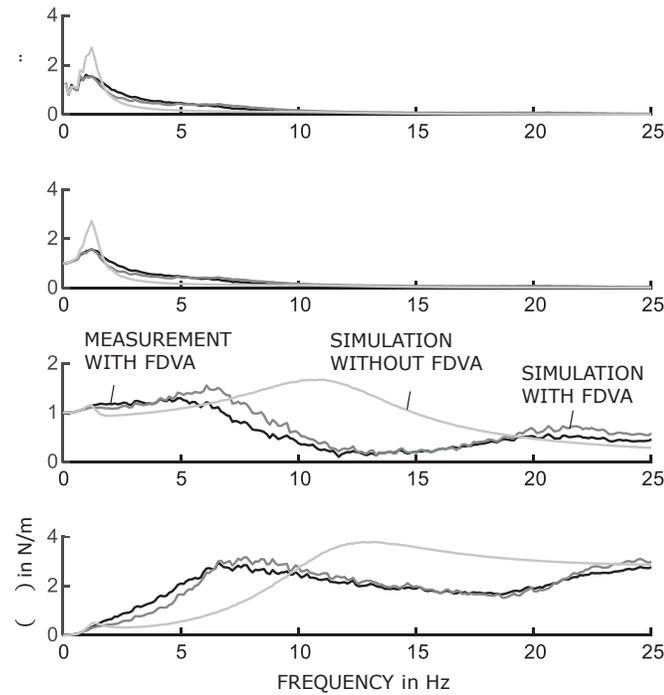


Figure 13: Comparison of a HiL measurement of the FDVA with the corresponding simulation results and a simulation without FDVA.

5 Summary and Conclusion

In this article, a new concept of a hydraulically transmitted vibration absorber for installation in a suspension strut was presented. The functionality of the Fluid Dynamic Vibration Absorber was shown and first component measurements as well as measurements in the Hardware-in-the-Loop environment with a virtual quarter car were used to validate the axiomatic model. The simulation model has already shown that it is possible to improve driving safety without, or only slightly, affecting driving comfort and thus shifting the Pareto line. The measurements presented show that the vibration absorber fulfils its function and that the hydraulic transmission of the vibration absorber inertia is working quite well.

In the next step, we plan to optimize the design of the vibration absorber, which can be achieved by simply adjusting the vibration absorber spring. An adjustment of the natural frequency of the vibration absorber is necessary, because a vibration absorber that is not optimally designed for the system natural frequency could even worsen the vibration behaviour.

Furthermore, it is conceivable to realize a semi-active version of the Fluid Dynamic Vibration Absorber by designing the closing of the channels with electromagnetic valves. This means that the natural frequency of the absorber can be adapted to the driving situation.

The advantage of the Fluid Dynamic Vibration Absorber is the increase of driving safety. This is particularly important in motor racing. Downsizing of the vibration absorber is necessary to integrate it into a racing car. If the vibration absorber springs are located inside of the hydraulic transmission and only the piston moves on the piston rod, it is possible to avoid the extension of the piston rod, which leads to a significant reduction of the length.

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Nomenclature

The first column of the following table shows the symbols utilized for physical and mathematical quantities. The second column shows the meaning of each quantity. The dimension of each physical quantity is denoted in the third column, based on the generic quantities length (L), mass (M) and time (T).

Variable	Description	Dimensions
a	duct surface	L^2
A	piston surface	L^2
D	damping coefficient	1
	body damping	$M T^{-1}$
f	natural frequency	T^{-1}
	body force	$M L T^{-2}$
	duct length	L
	chamber length	L
	absorber spring stiffness	$M T^{-2}$
	body spring stiffness	$M T^{-2}$
	tire stiffness	$M T^{-2}$
	mass	M
F	fluid mass	M
	body mass	M
	vibration absorber mass	M
	wheel mass	M

	piston mass	M
p_v	pressure losses	$M L^{-1} T$
	excitation	L
	body excitation	L
	vibration absorber excitation	L
	wheel excitation	L
α	ration between piston and ducts	1
η	ratio of frequency	1
ρ	oil density	$M L^{-3}$
θ	inertia	M
ω	natural frequency	T^{-1}

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