An active suspension with reduced complexity

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ABSTRACT

This paper introduces a new active hydro pneumatic suspension system (HFD) and examines the dynamic behavior of the system. The HFD is developed at the Technische Universität Darmstadt within the Collaborative Research Centre (SFB) 805, supported by Deutsche Forschungsgemeinschaft (DFG). Unlike other active suspension systems, this system is characterized by a reduced complexity. This reduced complexity is succeeded by the integration of the actuator inside the system. Hence, pumps, hoses, filters or tanks can be omitted. Thereby a control of uncertainties is intended by the reduction of components fraught with uncertainty. In addition to this research focus the design of the HFD leads to new functions like active vibration control, stiffness control and the separation of hardware and function. The latter one means that it is possible to adapt the HFD to varying customer demands (such as sport or comfort set up) without any modifications of the hardware.

NOMENCLATURE

A	Area	m²
ΔA_t	pressure exposed area	m²
ΔA_v	Fluid displacing area	m²
A_w	Surface for heat exchange	m²
b_f	Diameter of the bellow convolution	m
c, c_d	Stiffness, stiffness at design point	N/mm

D	Diameter	m
f, f_γ	Frequency, cut-off frequency	1/s
$F, \hat{F}, \hat{F}_d$	Force, Force amplitude, Force amplitude at design point	N
g	gravity acceleration	m/s²
k	Heat transfer coefficient	W/m²K
l	Length of the bellow in contact with the piston	m
n, γ	Polytrophic, isentropic exponent	-
p, p_0	Pressure, pressure at initial state	Pa
P	Power	W
q_v	Volume flow rate due to the deflection	m³/s
q_f	Volume flow rate due to the piston widening	m³/s
q_a	Volume flow rate into the hydraulic accumulator	m³/s
$r, \hat{r}$	Radius of the piston, radius amplitude	m
r_m	Piston radius plus half of the bellow convolution diameter	m
r_t	Bearing radius	m
r_a	Radius of the surrounding tube	m
R	Specific gas constant	J/(kg K)
$s, \hat{s}$	Deflection, deflection amplitude	m
T	Temperature	K
u	Velocity	m/s
V, V_0	Volume, volume at initial state	m³
α	ratio of radii	-
κ	compressibility	m/Pa
ρ	Density of the fluid	kg/m³

1 INTRODUCTION

Every technical system is affected by uncertainties, during the processes of construction, manufacturing and usage. To handle these uncertainties the Deutsche Forschungsgemeinschaft (DFG) is supporting a special research project (Sonderforschungsbereich SFB805, www.sfb805.tu-darmstadt.de) at the Technische Universität Darmstadt. Within this project an active suspension system will be developed, being a hydro pneumatic spring-damper-system with reduced complexity. On the one hand, the reduced complexity leads to fewer uncertainties, on the other hand the innovative design gives a chance to better functions compared with today's suspension components.

To achieve the reduced complexity, hydraulic infrastructure systems like pumps, hoses, accumulators and tanks are being replaced by an integrated actuator. This actuator implements the functions of leveling as well as the active reaction on dynamic loads by widening the piston radius.

To examine the fundamental behavior and causal relations of such a system, first studies were done with analytic calculations, 0-D dynamic simulations (Modelica/Dymola), and 3-D Finite Element Method (ABAQUS).

With the analytic calculations, the correlation between the piston radius, the pressure and the related force is investigated. The 0-D dynamic model deals with the dynamic behavior of the system; the 3-D FEM model is used to determine the forces during the piston widening.

2 THE HYDROPNEUMATIC SPRING-DAMPER SYSTEM – OVERVIEW

The HFD (**Figure 1**) is made up of the four main parts: bellow, piston rod with two pistons, restrictor and hydraulic accumulator. Either one or both pistons have an integrated actuator to increase or decrease the piston radius. The bellow is fixed at both pistons and is in contact with the surrounding tube which significantly increases the burst pressure (/Pel07/). With deflection, the upper piston displaces fluid, the lower one opens space for fluid. The displacement surface for each piston is the piston area minus the piston rod cross sectional area plus half of the surface projected by the bellow (membrane) convolution. The total displacement surface is the difference between

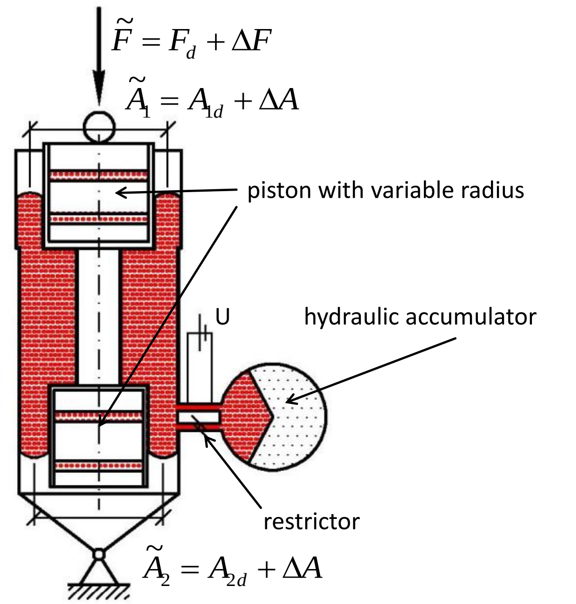

Figure 1: Schematic diagram of the HFD.

these two displacement surfaces. Since the diameter of the lower piston is smaller than the upper one, a volume is displaced. This displaced volume results in a pressure increase inside the hydraulic accumulator. The pressure acts on the effective bearing surface and results in a force. The damping force is either the result of a restriction of the cross sectional area or, in further project phases, the result of an applied potential or current when using electrorheological or magnetorheological Fluids (ERF or MRF).

3 STIFFNESS

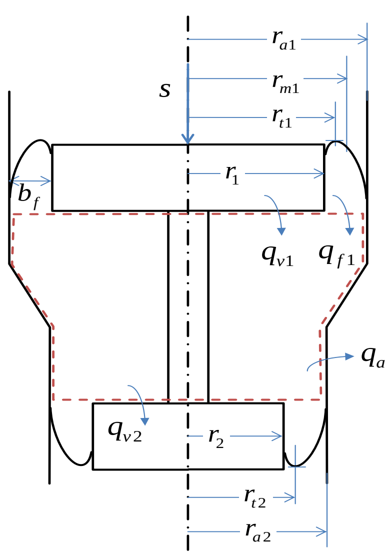

Figure 2: Notations.

Each passive or semi active suspension system has three main functions: Load carrying, energy conservation and energy dissipation. This chapter deals with the dynamic stiffness (0.01Hz…20Hz) and the eigen frequency considering the piston widening and varying loads. All calculations assume one piston with fixed radius (piston 2) and one with variable radius (piston 1).

Figure 2 shows the used parameters for the following calculations. The variable parameter is the radius of piston 1, r_1. In the initial state, all parameter are set in order to obtain a pressure-resulted force of 7500N at a deflection of 76mm. The radii r_{a1}, r_{a2} of the supporting tube remain constant, in the initial state r_{a1}, r_{a2} are 12mm (= b_f) larger than r_1, r_2.

As shown in /Pel01/, the fluid displacing area ΔA_v is given by the piston area plus half of the projected area of the bellow convolution

$$\Delta A_v = \frac{\pi}{2}\left[r_{a1}^2 - r_{a2}^2 + r_1^2 - r_2^2\right]. \tag{1}$$

For cross layer bellows, the radius of the pressure exposed area ΔA_t is approximately the piston radius plus 1/3 of the bellow convolution diameter b_f

$$\Delta A_t = \frac{\pi}{9}\left[(r_{a1} + 2r_1)^2 - (r_{a2} + 2r_2)^2\right], \tag{2}$$

with $r_{a1}, r_{a2}, r_2 = const$.

In general, the stiffness is given by $c = dF/ds$ with $dF = dp\Delta A_t$. Using the total derivative of the polytrophic relation $d(pV^n) = V^n dp + npV^{n-1}dV = 0$, the differential deflection $ds = -dV/\Delta A_v$ and the pressure p_d at the design point (deflection $s = s_d$) with the vehicle load F, the stiffness becomes a function of the initial states p_0 and V_0 and the cross sectional areas ΔA_v and ΔA_t which both are functions of r_1:

$$c = \frac{dF}{ds} = \frac{np}{V}\Delta A_t \Delta A_v = np_0 \frac{V_0 \Delta A_t \Delta A_v}{(V_0 - s_d \Delta A_v)^2} = \frac{nF^2}{V_0 p_0}\frac{\Delta A_v}{\Delta A_t}, \tag{3}$$

for a cylindrical l piston. By doing so, we assume that the vibration's frequency is higher than the cut-off frequency f_γ which marks the change between isothermal and adiabatic behavior. At f_γ, the loss angle has its first maximum, as shown in chapter 4 $f_\gamma \approx 0.1Hz$.

Assuming the load F is due to a heavy mass with $F = m\,g$ the system's eigen frequency ($f = \sqrt{gc/F}/(2\pi)$) is shown in **Figure 3** for three different cases as a function of the stiffness and the eigen frequency ($f = \sqrt{gc/F}/(2\pi)$) of F after leveling: leveling with constant gas mass (m_{gas} = *const*, typical hydro pneumatic), constant gas volume (V_{gas} = *const,* typical air spring) (/Eul03/) and with variable piston radii (HFD). Obviously, the behavior of the HFD is similar to an air spring, with tendency to a hydro pneumatic system with preferred constant remaining eigen frequency. It has to point out, that the piston radius remains constant during deflection at these studies. A variable piston radius during deflection is investigated in chapter 4.

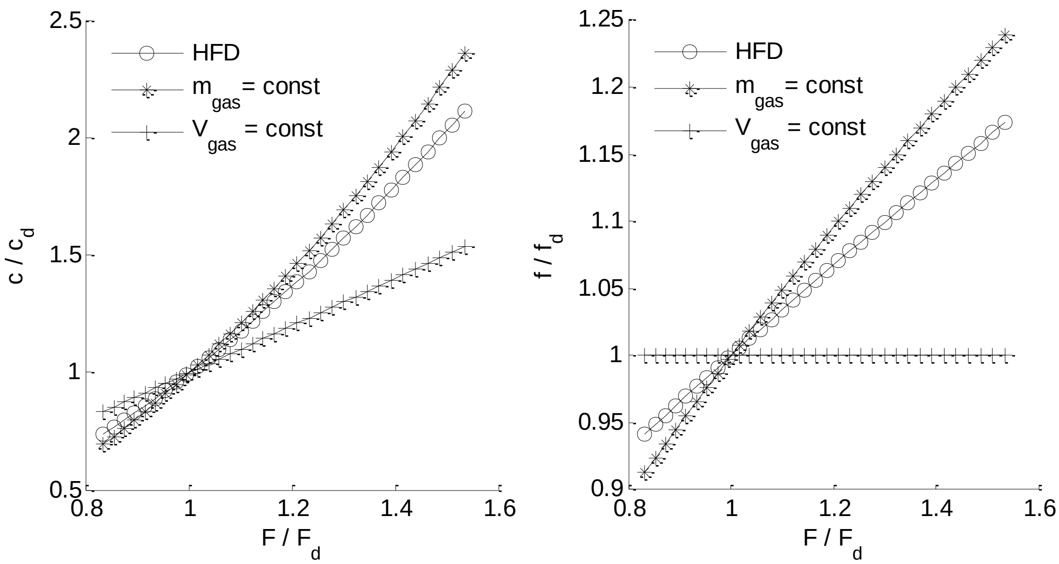

Figure 3: Stiffness ratio (left) and eigen frequency ratio for the HFD compared with common leveling systems.

Figure 4 shows the general effect of the piston widening. For a given fixed radii ratio $\alpha = r_1/r_2$ of the two piston radii and a given force F, a change of the ratio to $\tilde{\alpha} = \tilde{r}_1/\tilde{r}_2$ results in a force $\tilde{F}$.

If for example the ratio α is 1.1 a widening about 5% of this ratio ($(\tilde{\alpha}/\alpha) - 1 = 5E^{-2}$) causes a force $\tilde{F}$ that is 25% larger than F, i.e. it is possible to vary the force in a wide range with slight changes of the piston radius.

The most important task of research is the design of the adjustable piston. Since only slight changes of the piston radius are necessary, it might

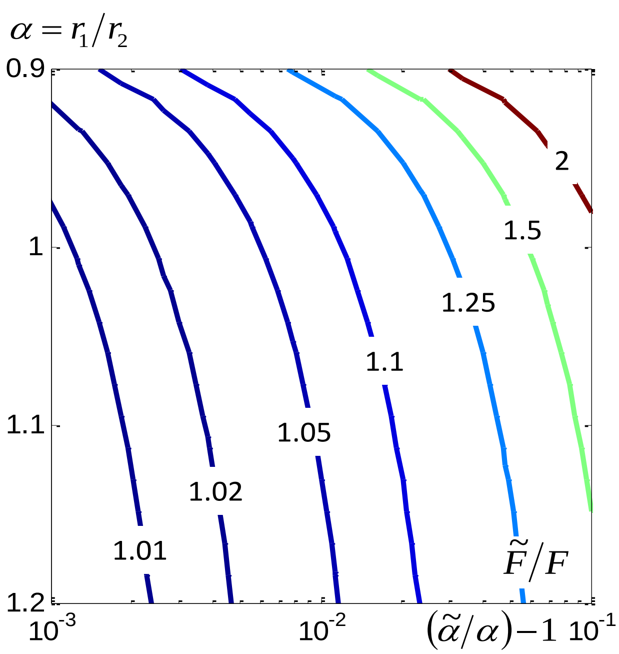

Figure 4: Correlation between piston widening and Force

be possible to use piezo-actuators for the widening. **Figure 5** (left) shows an early design study for such a system (see also /Pel08/): the piston consists of an elastomer ring with inlaid piezo-actuators. If the actuators increase in length, they extrude the elastomer radially and the piston radius increases. In Figure 5, the elastomer ring affects on radially ordered lamellae. These lamellae form a flexible surface on which the bellow lies on.

To examine the resulting forces during the piston widening, a complex multi-layer FE-model of the bellow was created, see Figure 5 (right).

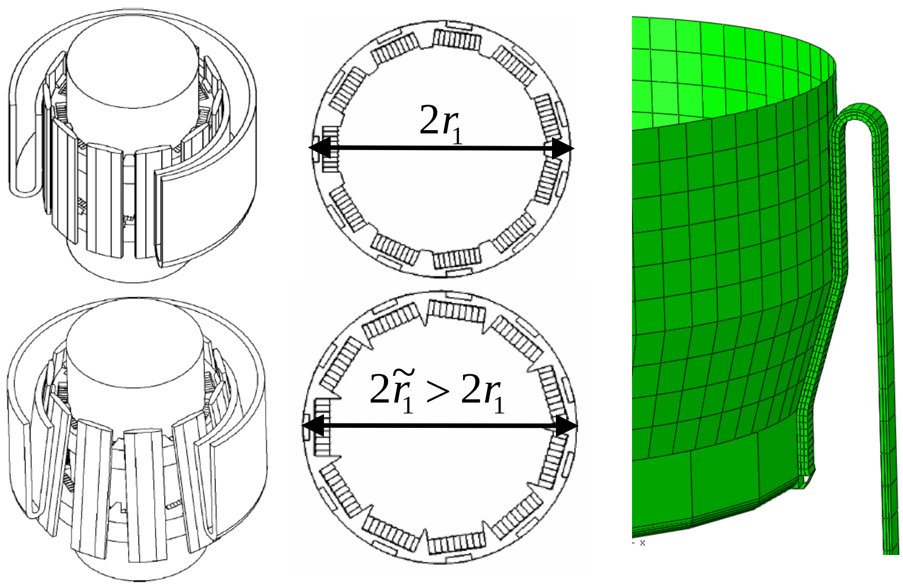

Figure 5: Piston widening with piezo actuator (left) and corresponding FE model (right).

4 DYNAMIC SIMULATION

By using the equation of mass conservation with consideration to the yielding of the bellow, we get

$$\frac{dp}{dt}\kappa V_b + \Delta q_v + q_a + q_{f1} = 0, \qquad (4)$$

whereby V_b is the volume inside the bellow in its deflected position, κ is the yieldingness of the bellow (determined by using the FE-model) and

$\Delta q_v = q_{v1} + q_{v2} = \dot{s}\Delta A_v$. By definition, the volume flow rate for inflowing fluid is negative. The volume V_b is calculated by integrating $dV_b/dt = -q_a$. To estimate the volume flow rate q_f due to the widening of the piston, we examine the change of the area A and the corresponding volume V covered by the bellow convolution.

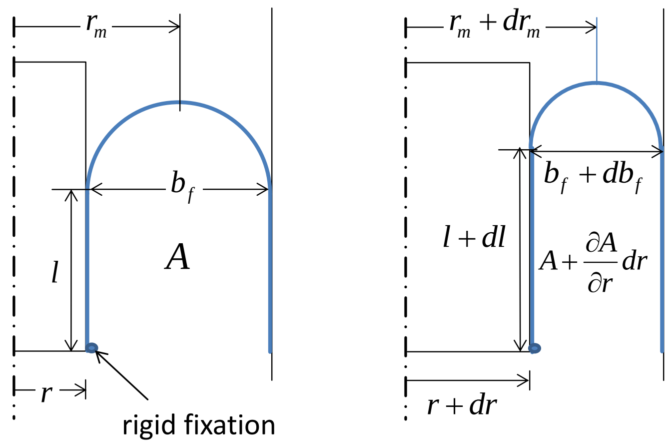

Figure 6: Enlargement of the piston radius

Because the volume increases due to the widening of the piston radius (r) and the increase of the spring deflection (s), we have to develop an equation for the flow rate at constant spring deflections. To obtain this equation, we make some assumptions to simplify the model. The radius r shall be constant along the piston; the bellow convolution shall be a semicircle. Because the bellow is fixed with a rigid ring (see **Figure 6**) the first assumption can hardly be achieved. In fact, this assumption represents the upper limit for the volume flow rate due to the piston widening.

With the definitions in Figure 6, the change of the cross sectional area A becomes by neglecting terms of higher order

$$dA = \frac{\partial A}{\partial r} dr = -l dr \, . \qquad (5)$$

Hence the volume $V = 2\pi r_m A$ and the volume flow rate q_f can be calculated by

$$\left.\frac{dV}{dt}\right|_{s=const} = \left.\frac{dV}{dr}\right|_{s=const} \frac{dr}{dt} = 2\pi \dot{r} \left(A \frac{dr_m}{dr} + r_m \frac{dA}{dr} \right) = 2\pi \dot{r} \left(\frac{1}{2} A - l r_m \right) = q_f \, . \qquad (6)$$

The required values for r, l, r_m and b_f are only functions of the parameters spring deflection s , initial piston radius r_d and Δr, determined by geometry:

$$r = r_d + \Delta r, \qquad l = \frac{s}{2} + \frac{\pi}{4}(r - r_d), \qquad r_m \frac{1}{2}(r + r_a), \qquad b_f = r_a - r. \tag{7}$$

The hydraulic accumulator is described with the laws of mass conservation and energy conservation (/Pel01/). This leads to a system of algebraic differential equations for the density ρ, the pressure p and the temperature T:

$$p = \rho RT, \qquad V\dot{\rho} + \rho q = 0, \qquad V\dot{p} + \gamma pq + (\gamma - 1)A_w k\,(T - T_u) = 0\,, \tag{8}$$

(with the volume flow rate q, the volume of the accumulator V, the ideal gas constant R, the isentropic exponent γ, and the heat transfer coefficient k). The advantage of this model compared to the polytrophic relation used in Eq. (3) is that no assumption about the polytrophic exponent is needed.

The restrictor can be described with Bernoulli's law for unsteady flow. Assuming a rigid pipe with constant cross sectional area, incompressible fluid and negligible potential difference, Bernoulli's law results in

$$p_1 - p_2 = \rho L\dot{u} + \Delta p_v \tag{9}$$

with the length of the pipe L, the flow velocity u and the pressure loss Δp_v.

The velocity u is given by the inflowing volume rate q_a and the pipes cross sectional area; for laminar flow, the pressure losses are given by $\Delta p_v = 8 q_a \eta L / \pi R^4$ with the dynamic viscosity η and the pipe's radius R *(/Spu07/).* With the help of the harmonic balance nonlinear losses due to inertia can also be modeled by this approach at least for harmonic signals.

The dynamic model is built up in Modelica, a standard to describe dynamic systems. To examine the dynamic behavior, a mono harmonic deflection is imposed on the system. For each frequency an individual simulation was run. Thereby, the radius of the piston is a function of the deflection: $r = (s - s_d)(r_d + \hat{r})/\hat{s}$ ($\hat{s}$: deflection amplitude). The amplitude of the radius $\hat{r}$ varies from 0% to approx. 20% of the piston radius in the design point r_d. The results of the simulation are the pressure inside the HFD and therewith the spring and damping force. These results are evaluated over one period in the steady state.

The dynamic stiffness c and the loss angle δ are here defined as

$$c := \frac{\hat{F}}{\hat{s}}, \qquad \sin(\delta) := \frac{W_d}{\pi \hat{z}^2 c} \tag{10}$$

The dissipated energy W_d is the area inside the hysteresis curve.

The power P, required to widen the piston, is given by

$$P = q_f \, p \,. \tag{11}$$

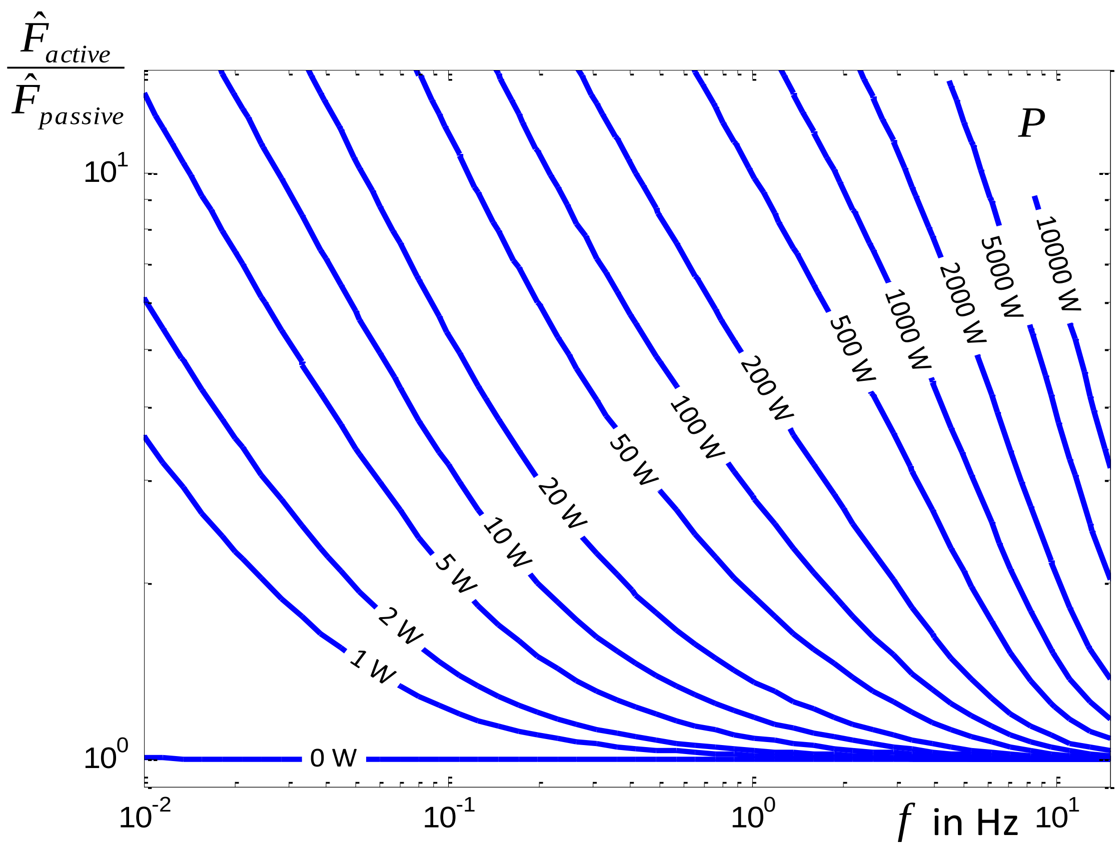

Figure 7: Required power to increase the force amplitude at varying frequency

Figure 7 shows the results of a frequency sweep ranging from 0.01Hz to 15Hz with $\hat{z}$ = 0.01 m. Because the HFD is an active system, it is possible to influence the behavior with energy input. The force amplitude at design point (indicated by index ‘d’) $\hat{F}_d$ can be varied by transferring energy. For example, to increase the force amplitude at 1Hz from $\hat{F}_{active} = \hat{F}_{passive}$ to $\hat{F}_{active} = 2\hat{F}_{passive}$, $P = 100W$ have to be transferred.

In **Figure 8**, the stiffness and the loss angle are depicted as a function of the frequency. At 0.1Hz, the loss angle has its first local maximum. This is because of the change from isothermal to adiabatic behavior while compressing the gas inside the hydraulic accumulator. This proves the assumption for an adiabic change during vibration stated before.

The stiffness increases with higher amplitudes for the piston radius $\hat{r}$. For higher frequencies, the stiffness increases due to the inertia of the fluid, represented by Eq. (9).

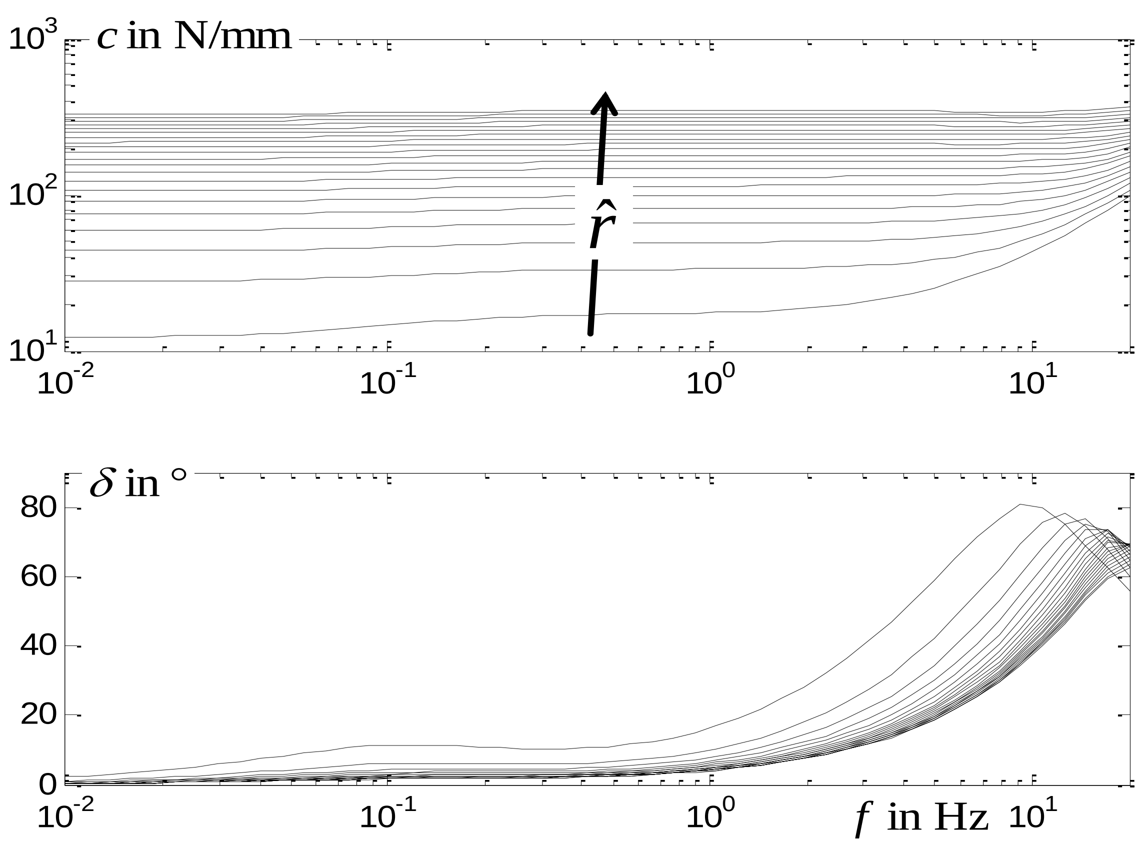

Figure 8: Stiffness (top) and loss angle (bottom) as a function of the frequency

5 FUTURE WORK

On the basis of the above-mentioned general studies, the next step is the development and construction of a HFD prototype as well as the improvement of the simulation models. The main and important part of the development phase is the elaboration of robust solutions for the piston widening considering the bellow fixation. With detailed models, the snap stability will be investigated. With the depicted FE-model and experiments, the forces related to the piston widening have to be investigated.

CONCLUSION

In this work, analytic and simulative models were introduced to show the fundamental behavior of the hydro pneumatic suspension system. It was found that it is possible to control and vary the forces in a wide range by energy input. Although in cases of higher frequency high power is demanded, only small changes in the piston radius are required. This gives the possibility to use piezo actuators for the piston widening.

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