A NOVEL PULSATION COMPENSATOR FOR DISPLACEMENT MACHINES

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ABSTRACT

This paper presents a mechanical compensator for hydraulic pressure pulsations induced by a pump or motor. The compensator is based on the combination of pump or motor mass moment of inertia and the torsional compliance of the adjacent coupling. According to the displacement volume, angular deflections of the resulting oscillator correspond to volumes which can compensate for geometric or dynamic pulsations at the natural frequency of the oscillator. Two design concepts are suggested for the coupling. Three models are set up for various configurations of the overall system, accounting for limited inertia of an electric motor and the influence of an outlet pipeline. Model parameters are determined for a 32 ccm radial piston pump and a 250 ccm axial piston pump. Torsional damping from viscous friction is roughly estimated and considered by different levels. At the pump outlet, frequency response functions between flow rate excitation and pressure response are calculated. The results show that the compensation effect is relevant for both pumps and particularly robust with respect to resonances in the hydraulic system.

Keywords: pump, pressure pulsations, compensator, torsional oscillations, coupling

1. INTRODUCTION

The periodic operation of hydraulic pumps and motors usually leads to a pulsating flow which depends on the kinematics of the working principle. Moreover, the precompression of displacement chambers need not match the pressure at the outlet and periodic impulsive flow rates can result from this effect. The overall flow rate pulsation of the machine excites pressure pulsations in the attached hydraulic system. This results in structural vibrations and noise, and may even lead to cavitation. Various countermeasures have been suggested, at the cost of efficiency or additional components [1].

In structural mechanics, vibration compensators are supposed to counteract harmonic forces by natural vibrations at the same frequency. In hydraulic systems, such compensators can be understood to counteract harmonic flow rates by their natural vibrations. A Helmholtz resonator [2], for instance, extracts harmonic flow rates in its attachment point. Its natural frequency is defined by the inertia of the attachment pipe and the compliance of the resonator chamber.

For practical inclusion in hydraulic systems, more compact compensators are required [3]. A special solution is described and investigated in [4]. Compared to a Helmholtz resonator, the inertia of the fluid in the attachment pipe is replaced by a lumped mechanical mass, while the resonator chamber provides a hydraulic spring.

In this paper a flow rate pulsation compensator realised by the inertia of the machine rotor and the torsional compliance of its coupling is proposed [5]. The compensator constitutes a torsional oscillator which counteracts the pulsating flow from the machine into the hydraulic system. This requires a lightly damped coupling with a torsional stiffness selected to match the natural frequency of rotor and coupling with the pulsation frequency. The ratio between flow rate pulsation and angular velocity amplitudes is given by the displacement volume of the machine. By appropriate angular movements, the machine is supposed to compensate its own pulsation. Contrary to separate devices like Helmholtz resonators, pulsation compensation takes place in the displacement chamber itself, which is the source of flow rate pulsation. Moreover, maximum compactness is achieved.

The novel compensator concept promises a reduction of pressure pulsations in hydraulic systems at the expense of a tailored torsional stiffness of the coupling attached to pump or motor. Alternatively, the concept offers more freedom in the design of the machine itself as it may be exchanged against other measures for pulsation reduction (e.g. the number of pistons).

Nevertheless, the practical realisation of the novel compensator faces several challenges. Hydraulic damping is always present and reduces the effectiveness of pulsation compensation. A broadband effect will only be achieved for certain relations between machine and coupling properties. In order to investigate the feasibility of the concept, a frequency domain study is carried out for two pumps of different type and size. In particular, viscous friction in sealing gaps is quantified, torsional vibrations of the entire pump drivetrain are modelled, and the dynamic behaviour of a transmission line is taken into account.

2. SYSTEM CONFIGURATION

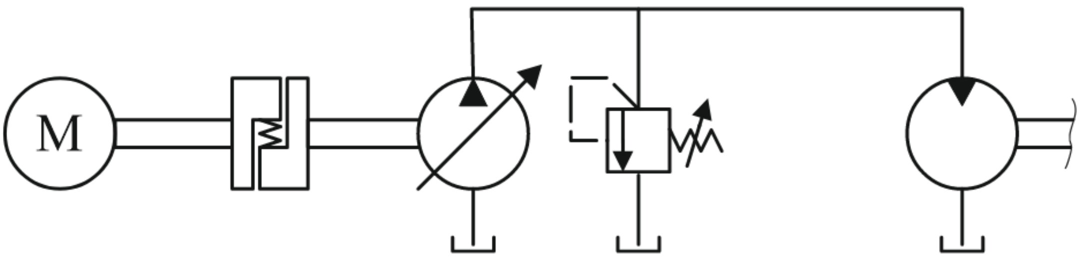

Figure 1: Schematic of overall system

As one out of many application examples, **Figure 1** shows the schematic of an open circuit transmission with a variable displacement pump and a fixed displacement motor. The pump is driven by an electric motor at constant speed, with which it is connected by a coupling. Compared to the pump, the electric motor's mass moment of inertia will be several times higher so that the first torsional natural frequency of the drivetrain can be approximated from the inertia of the pump rotor and the torsional stiffness of the coupling.

In the hydraulic system, the pump excites pressure pulsations at a fundamental frequency depending on rotational speed and number of displacement chambers, which are usually accompanied by several higher harmonics. In principle, all torsional natural modes of the pump drivetrain could be used for pulsation compensation. The corresponding natural frequencies must match the pump harmonics to be compensated. In order to compensate a single pulsation frequency, the torsional stiffness of the coupling is chosen for an appropriate match with the first natural frequency of the pump drivetrain.

Coupling design should avoid any torsional damping since the torsional oscillator includes unavoidable viscous damping at the pump rotor. Depending on the type of pump, the rotor's mass moment of inertia may vary. In particular, this is the case for radial piston pumps, where a mean value will be used for compensator design. Of course, the system which is driven by the hydraulic motor could also be equipped with an appropriate coupling to realise another compensator.

3. COUPLING CONFIGURATONS

The compensator coupling requires a relatively low torsional stiffness in the absence of damping, as opposed to conventional couplings of comparable size. Therefore, two different concepts are suggested with individual advantages of each.

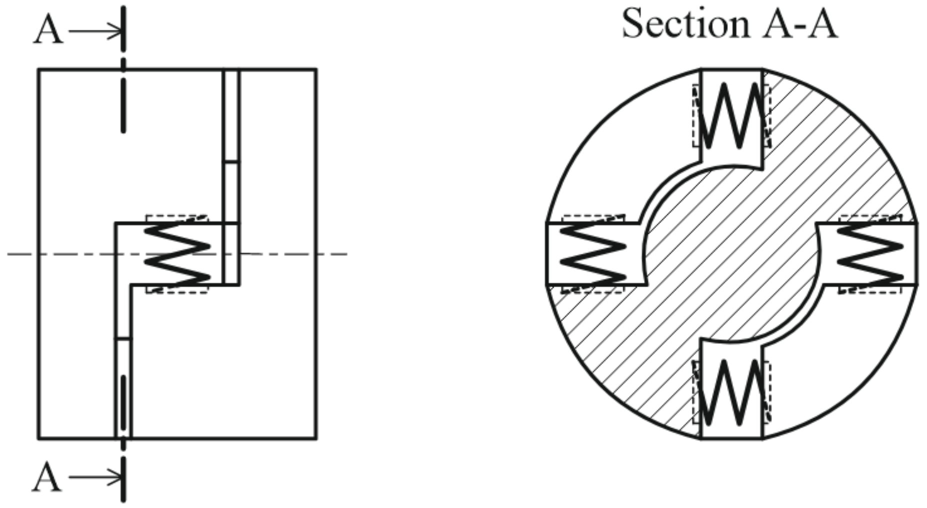

Figure 2: Coil spring concept

Figure 2 shows a concept where two coupling halves engage in such a way that four tangential coil springs can be arranged in between, which define the torsional stiffness of the coupling. This concept is favoured by the fact that it covers a wide range of stiffness while broken springs are easily exchanged.

The concept in **Figure 3** uses a radial arrangement of flat rectangular beams, whose tangential deflection under torque defines the torsional stiffness. The beams are connected to an inner and an outer ring, all of which can be manufactured as an integral part. The outer ring is held in the carrier flange by a shrink fit, while the inner ring may be connected to the pump shaft by a key. This concept seems compact enough to be integrated in a pump housing and could also be cascaded for the compensation of several pulsation frequencies. An example for two pulsation frequencies is shown in **Figure 4**. For such an arrangement, it may be necessary to adapt the intermediate flange inertia to obtain feasible values of both torsional stiffnesses.

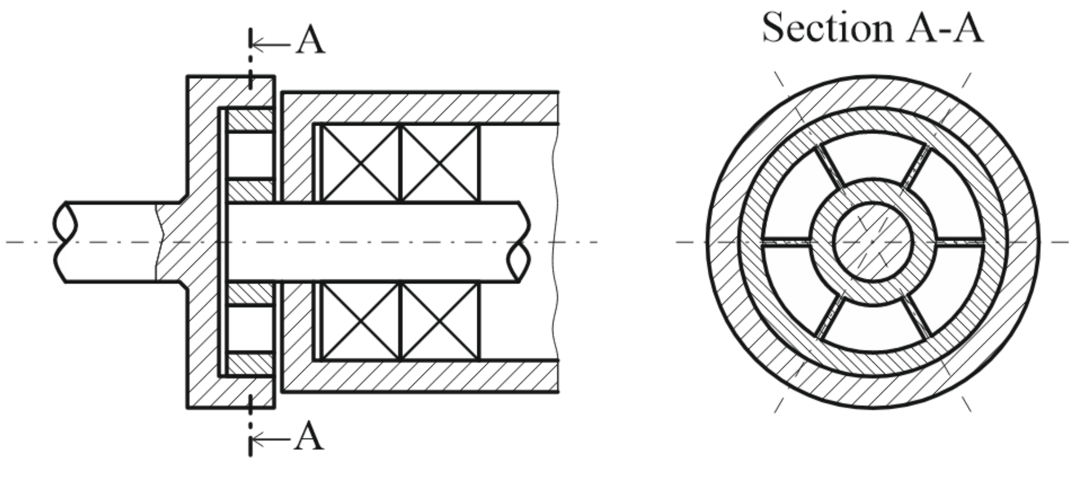

Figure 3: Beam concept

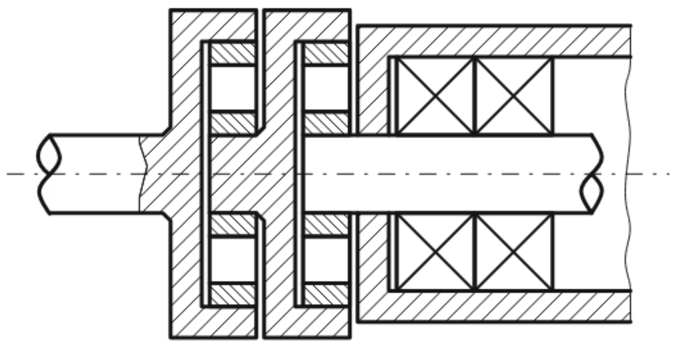

Figure 4: Cascaded beam concept

4. FREQUENCY DOMAIN MODEL

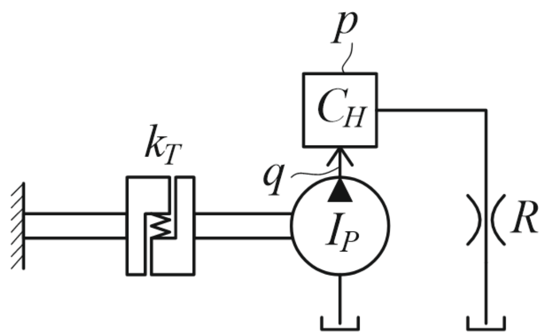

Figure 5: Schematic of system with fixed boundary condition at electric motor end and hydraulic system with throttle towards tank

A first numeric assessment of the novel compensator concept is made by frequency domain models of the systems in **Figure 5**, **Figure 6**, and **Figure 7**. To demonstrate the compensation effect with a minimum number of components, **Figure 5** includes a coupling with a fixed boundary condition at the electric motor end, a pump rotor, the inevitable hydraulic capacity at the pump outlet, and a throttle towards tank. Due to viscous fluid friction, torsional damping acts on the pump rotor. With the torsional stiffness k_T of the coupling, the mass moment of inertia I_P of the pump rotor, the torsional damping constant c_T, the hydraulic capacity C_H, the hydraulic resistance R, and the displacement volume V_P of the pump, the time domain model reads

$$\begin{bmatrix} I_P & 0 \\ -\frac{V_P}{2\pi} & C_H \end{bmatrix} \begin{bmatrix} \ddot{\varphi}_P \\ \ddot{p} \end{bmatrix} + \begin{bmatrix} c_T & 0 \\ 0 & \frac{1}{R} \end{bmatrix} \begin{bmatrix} \dot{\varphi}_P \\ \dot{p} \end{bmatrix} + \begin{bmatrix} k_T & \frac{V_P}{2\pi} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \varphi_P \\ p \end{bmatrix} = \begin{bmatrix} 0 \\ \dot{q} \end{bmatrix}, \tag{1}$$

where q is the flow rate excitation into the hydraulic capacity C_H, p is the pressure in this capacity, and φ_P is the rotation angle of the pump. Equation (1) represents a linear two-degree-of-freedom model with asymmetric system matrices due to hydraulic-mechanical interaction. The second order mechanical subsystem describes an oscillator, while the hydraulic subsystem is only first order. The corresponding frequency domain model reads

$$\left(-\omega^2 \begin{bmatrix} I_P & 0 \\ -\frac{V_P}{2\pi} & C_H \end{bmatrix} + \mathrm{i}\omega \begin{bmatrix} c_T & 0 \\ 0 & \frac{1}{R} \end{bmatrix} + \begin{bmatrix} k_T & \frac{V_P}{2\pi} \\ 0 & 0 \end{bmatrix}\right) \begin{bmatrix} \Phi_P(\mathrm{i}\omega) \\ P(\mathrm{i}\omega) \end{bmatrix} = \begin{bmatrix} 0 \\ \mathrm{i}\omega Q(\mathrm{i}\omega) \end{bmatrix}, \tag{2}$$

where capital letters for excitation and states denote the respective Laplace transforms, and ω is the angular frequency.

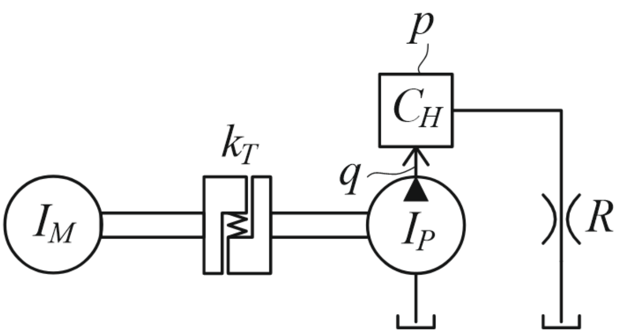

Figure 6: Schematic of system with electric motor and hydraulic system with throttle towards tank

Compared to **Figure 5**, **Figure 6** also accounts for the electric motor's limited mass moment of inertia I_M. The corresponding model also includes the Laplace transformed rotation angle Φ_M of the electric motor and reads

$$\left(-\omega^2\begin{bmatrix} I_M & 0 & 0 \\ 0 & I_P & 0 \\ 0 & -\frac{V_P}{2\pi} & C_H \end{bmatrix} + \mathrm{i}\omega\begin{bmatrix} 0 & 0 & 0 \\ 0 & c_T & 0 \\ 0 & 0 & \frac{1}{R} \end{bmatrix} + \begin{bmatrix} k_T & -k_T & 0 \\ -k_T & k_T & \frac{V_P}{2\pi} \\ 0 & 0 & 0 \end{bmatrix}\right)\begin{bmatrix} \Phi_M(\mathrm{i}\omega) \\ \Phi_P(\mathrm{i}\omega) \\ P(\mathrm{i}\omega) \end{bmatrix} = \begin{bmatrix} T_M(\mathrm{i}\omega) \\ 0 \\ \mathrm{i}\omega Q(\mathrm{i}\omega) \end{bmatrix}, \quad (3)$$

where T_M denotes the Laplace transform of any torque from the electric motor.

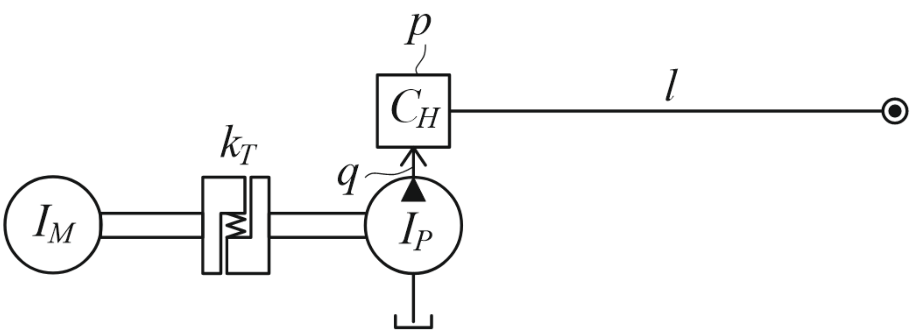

Figure 7: Schematic of system with electric motor and hydraulic system with pipeline towards constant pressure boundary condition

To account for resonances in the hydraulic subsystem, the throttle is replaced by a pipeline from the hydraulic capacity to a constant pressure boundary condition. This configuration is shown in **Figure 7**. Using the pipeline description for dynamic laminar flow as developed in [6], the system model reads

$$\left(-\omega^2\begin{bmatrix} I_M & 0 & 0 \\ 0 & I_P & 0 \\ 0 & -\frac{V_P}{2\pi} & C_H \end{bmatrix} + \mathrm{i}\omega\begin{bmatrix} 0 & 0 & 0 \\ 0 & c_T & 0 \\ 0 & 0 & S_{12} \end{bmatrix} + \begin{bmatrix} k_T & -k_T & 0 \\ -k_T & k_T & \frac{V_P}{2\pi} \\ 0 & 0 & 0 \end{bmatrix}\right)\begin{bmatrix} \Phi_M(\mathrm{i}\omega) \\ \Phi_P(\mathrm{i}\omega) \\ P(\mathrm{i}\omega) \end{bmatrix} = \begin{bmatrix} T_M(\mathrm{i}\omega) \\ 0 \\ \mathrm{i}\omega Q(\mathrm{i}\omega) \end{bmatrix}, \quad (4)$$

in which

$$S_{12} = -\frac{\mathrm{i}A}{\sqrt{E\rho}f(\mathrm{i}\omega)}\cot\left(\frac{f(\mathrm{i}\omega)\omega l}{\sqrt{E/\rho}}\right) \tag{5}$$

with

$$f(\mathrm{i}\omega) = \sqrt{-\frac{\mathrm{J}_0\left(\mathrm{i}r\sqrt{\mathrm{i}\omega/\nu}\right)}{\mathrm{J}_2\left(\mathrm{i}r\sqrt{\mathrm{i}\omega/\nu}\right)}}, \tag{6}$$

where J_0 and J_2 are Bessel functions of first kind, l, r, and A denote pipeline length, radius, and cross-sectional area, respectively, E is the bulk modulus, ρ the mass density, and ν denotes the kinematic viscosity of the fluid.

The torsional stiffness of the coupling is calculated in order to match the pulsation frequency f_P with the torsional natural frequency of the pump drivetrain. This results in

$$k_T = \frac{(2\pi f_P)^2}{\frac{1}{I_M} + \frac{1}{I_P}}. \tag{7}$$

A torsional damping constant c_{TG} is estimated for different types of sealing gap with gap height h. A cylindrical sealing gap appears at the inner radius r_i of the radial piston pump rotor. The relative motion of gap walls due to the angular velocity $\dot{\varphi}_P$ of the rotor causes a uniform shear stress

$$\tau_R = \nu\rho\frac{r_i\dot{\varphi}_P}{h}, \tag{8}$$

resulting in a shear torque

$$T_{\tau R} = 2\pi r_i^2 l_P\,\tau_R \tag{9}$$

over the rotor length l_P and a torsional damping constant

$$c_{TGR} = \frac{T_{\tau R}}{\dot{\varphi}_P} = 2\pi\nu\rho\frac{r_i^3 l_P}{h}. \tag{10}$$

Between axial piston pump rotor and valve plate, the sealing gap assumes the form of a disc with inner radius r_i and outer radius r_a. The shear stress

$$\tau_A = \nu\rho\frac{x\dot{\varphi}_P}{h} \tag{11}$$

depends on the intermediate radius x and causes the shear torque

$$T_{\tau A} = \int_{r_i}^{r_a} 2\pi x^2 \tau_A dx. \tag{12}$$

In this case, the torsional damping constant becomes

$$c_{TGA} = \frac{T_{\tau A}}{\dot{\varphi}_P} = \frac{\pi}{2}\nu\rho\frac{r_a^4 - r_i^4}{h}. \tag{13}$$

Accounting for comparable viscous damping from the slippers, the overall torsional damping constant becomes $c_T = 2c_{TG}$.

For a medium size radial piston pump driven by a servo motor and a large axial piston pump driven by an induction motor, model parameters are shown in **Table 1**.

Table 1: Model parameters

	Radial piston pump	Axial piston pump
I_M	0.018 kg m^2	2.55 kg m^2
I_P	0.0061 kg m^2	0.0959 kg m^2
V_P	32 cm^3	250 cm^3
l_P	40 mm	
r_i	20 mm	30 mm
r_a		70 mm
h	0.04 mm	0.03 mm
z	7	9
n	25 Hz	25 Hz
f_P	zn	zn
V_d	V_P	$V_P/2$
E	1400 MPa	1400 MPa
C_H	V_d/E	V_d/E
R	10^{11} Pa m^{-3} s	10^{10} Pa m^{-3} s
ρ	860 kg m^{-3}	860 kg m^{-3}
ν	40 mm^2 s^{-1}	40 mm^2 s^{-1}
l	1.1 m … 2.1 m	0.7 m … 1.7 m
r	7.5 mm	16 mm
A	$r^2\pi$	$r^2\pi$

5. RESULTS AND DISCUSSION

The following figures show frequency response functions between flow rate excitation and pressure response in the hydraulic capacity at the pump outlet. Different curves are shown for systems with and without compensator. To account for parameter uncertainty and damping mechanisms not included in the model, a damping constant of $10\ c_T$ is used for a compensator with light damping, while $100\ c_T$ is used for a compensator with high damping.

From the model in **Figure 5**, the frequency response functions in **Figure 8** and **Figure 9** have been obtained for radial piston pump and axial piston pump, respectively. The compensator causes a local antiresonance (151 Hz for the radial piston pump and 221 Hz for the axial piston pump), which is followed by a resonance (166 Hz for the radial piston pump and 232 Hz for the axial piston pump). Both antiresonance and resonance are more pronounced for light damping. At excitation frequencies around the antiresonance, pressure response is reduced by the compensator. The contrary happens around the resonance, whose excitation should be avoided. To use the compensation effect, the pulsation frequency of the pump should match the antiresonance. Since the torsional stiffness of the coupling is determined from Eq. (7) and the model in **Figure 5** does not account for the electric motor's limited mass moment of inertia, the model must be refined to obtain the intended match.

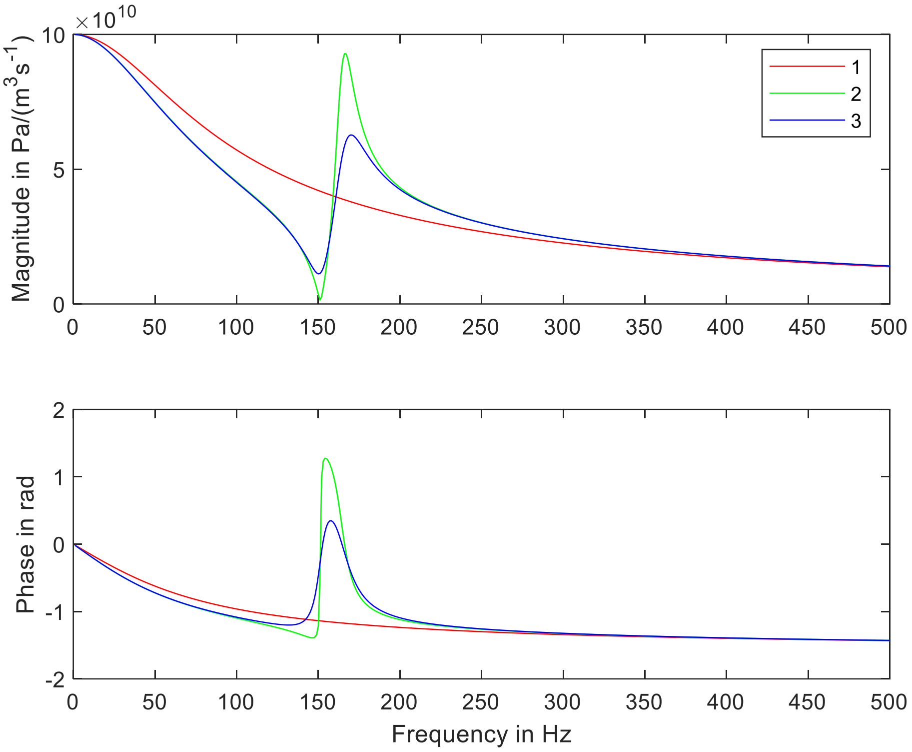

Figure 8: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, radial piston pump, pump inertia, capacity and resistance at the outlet, 1: without compensation, 2: light damping compensator, 3: high damping compensator

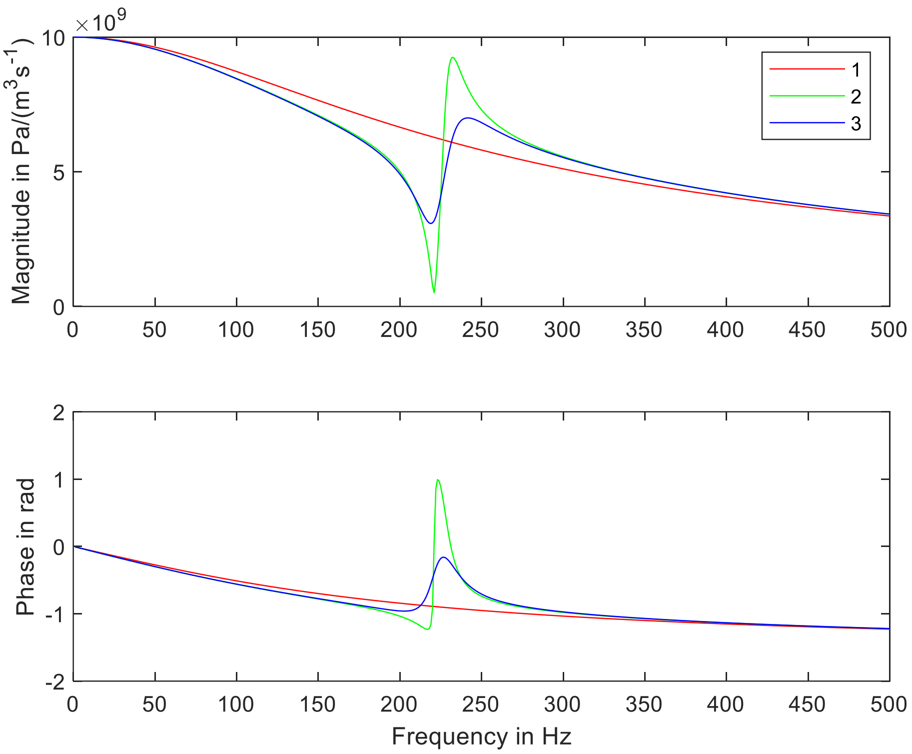

Figure 9: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, axial piston pump, pump inertia, capacity and resistance at the outlet, 1: without compensation, 2: light damping compensator, 3: high damping compensator

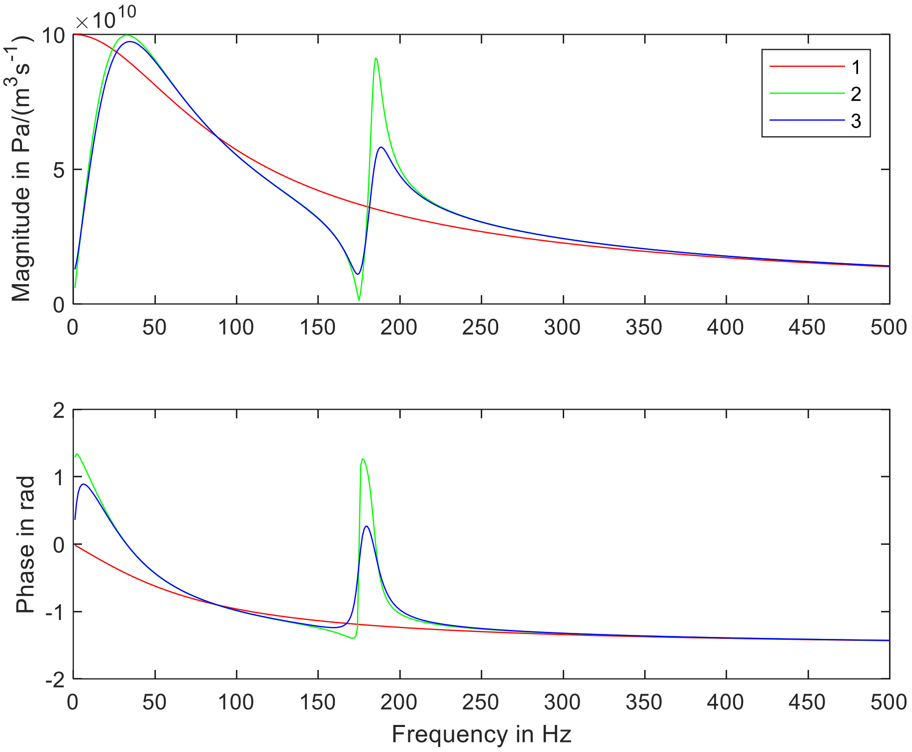

Figure 10: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, radial piston pump, motor inertia, pump inertia, capacity and resistance at the outlet, 1: without compensation, 2: light damping compensator, 3: high damping compensator

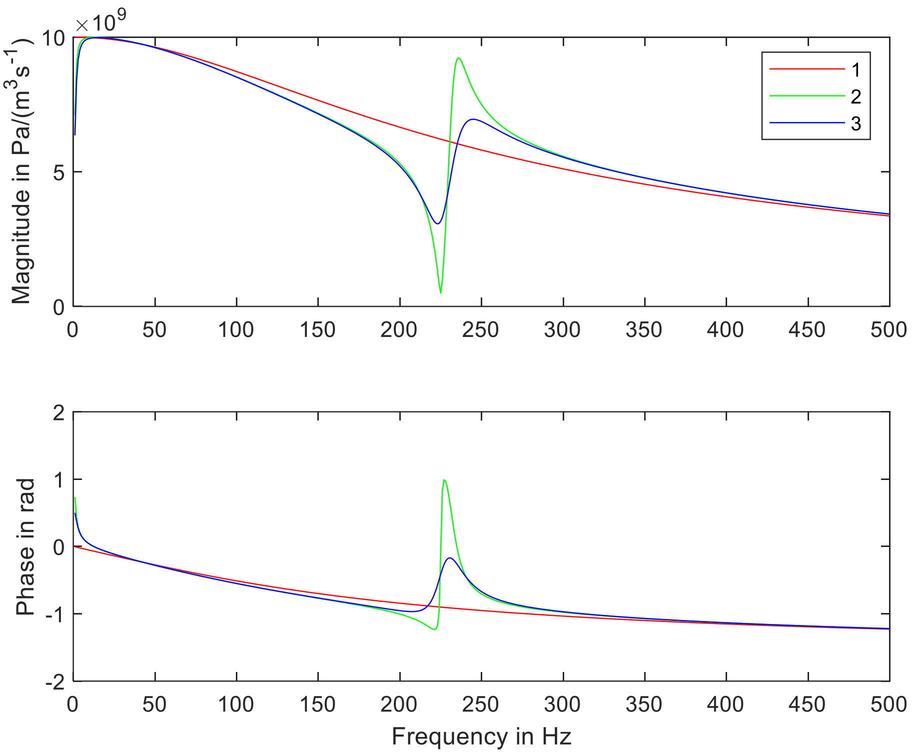

Figure 11: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, axial piston pump, motor inertia, pump inertia, capacity and resistance at the outlet, 1: without compensation, 2: light damping compensator, 3: high damping compensator

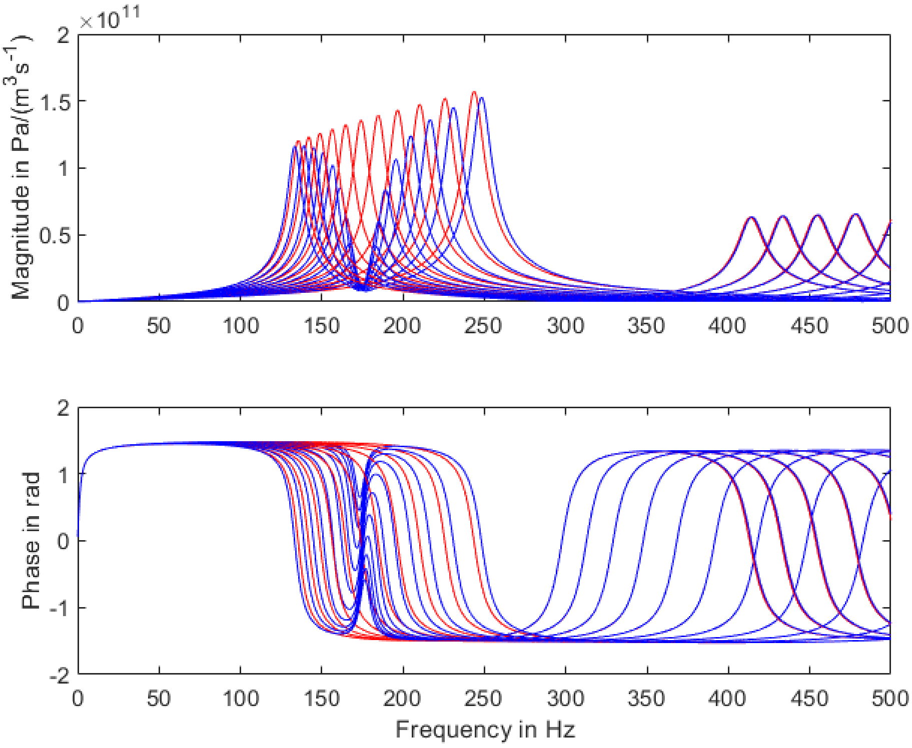

Figure 12: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, radial piston pump, motor inertia, pump inertia, capacity and pipeline at the outlet, red curves: different pipeline lengths without compensation, blue curves: different pipeline lengths with high damping compensator

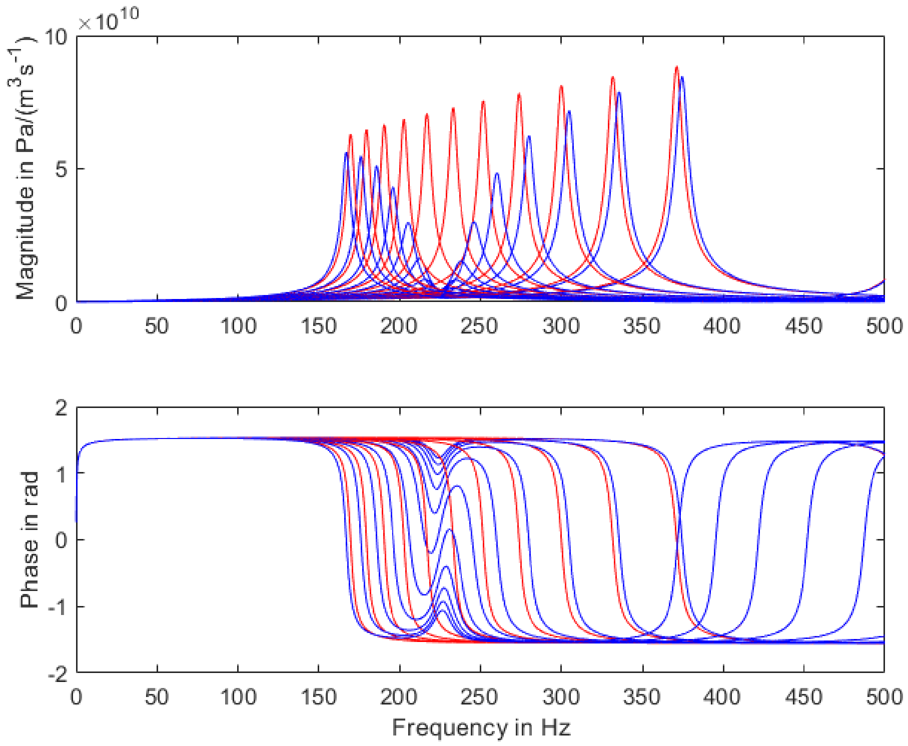

Figure 13: $P(\mathrm{i}\omega)/Q(\mathrm{i}\omega)$, axial piston pump, motor inertia, pump inertia, capacity and pipeline at the outlet, red curves: different pipeline lengths without compensation, blue curves: different pipeline lengths with high damping compensator

Figure 10 and **Figure 11** result from the model in **Figure 6**. Accounting for the motor inertia, another resonance emerges (33 Hz for the radial piston pump and 13 Hz for the axial piston pump). The radial piston pump antiresonance shifts up to 175 Hz. For the axial piston pump, the electric motor's mass moment of inertia is higher in relation, resulting in an antiresonance shift up to 225 Hz. In both cases, one can see that the entire pump drivetrain dynamics is essential for the antiresonance frequency, which now matches the pulsation frequency of the respective pump.

Figure 12 and **Figure 13** refer to the model in **Figure 7**. For radial piston pump outlet pipeline lengths from 1.1 m to 2.1 m in steps of 0.1 m, **Figure 12** shows frequency response functions with and without the high damping compensator. In **Figure 13** such frequency response functions are shown for axial piston pump outlet pipeline lengths from 0.7 m to 1.7 m in steps of 0.1 m. One can see that the compensation effect is largely independent of the outlet pipeline resonance frequency. The benefit of the compensator becomes most apparent where the intended antiresonance coincides with the original outlet pipeline resonance. At the cost of two side resonances, the intended antiresonance remains.

6. CONCLUSION AND OUTLOOK

For hydraulic pumps and motors, a novel pulsation compensation device has been suggested. With a suitable coupling, the drivetrain of the hydraulic machine forms a lightly damped torsional oscillator whose natural vibrations compensate flow rate excitations occurring in the displacement chamber. The device can be realized by merely defining the torsional properties of the coupling. Two possible coupling design concepts have been indicated. For a medium size radial piston pump and a large axial piston pump, frequency domain models show significant attenuation of pressure pulsation at a fixed excitation frequency and high robustness with respect to response characteristics of the hydraulic system at the pump outlet.

In a next step, laboratory tests shall be carried out with a compensator tailored to the radial piston pump. Coupling design shall be detailed to allow for sufficient torsional compliance despite strength requirements. Under excitation from the pump itself, pressure response shall be compared between setups with compensator and standard coupling, respectively. In both cases, different hydraulic networks will be installed at the pump outlet. As the flow rate excitation from the pump cannot be measured, similar experiments with external valve excitation shall be carried out to enable direct comparisons with frequency domain models. If experimental results come near the predicted behaviour, the novel compensator offers effective pulsation reduction at little cost for many displacement machine applications at constant rotational speed.

NOMENCLATURE

A	Pipeline cross-sectional area	m^2
C_H	Hydraulic capacity	$m^3\ Pa^{-1}$
c_T	Torsional damping constant	$N\ m\ s\ rad^{-1}$
E	Bulk modulus	Pa
f_P	Pulsation frequency	Hz
h	Gap height	m
I_M	Electric motor mass moment of inertia	$kg\ m^2$
I_P	Pump mass moment of inertia	$kg\ m^2$
k_T	Torsional stiffness	$N\ m\ rad^{-1}$
l	Length of pipeline	m
l_P	Length of pump rotor	m

n	Rotation frequency	Hz
p	Pressure at pump outlet	Pa
P	Laplace transformed pressure at pump outlet	Pa
q	Flow rate at pump outlet	$m^3\ s^{-1}$
Q	Laplace transformed flow rate at pump outlet	$m^3\ s^{-1}$
r	Radius of pipeline	m
R	Resistance	$Pa\ m^{-3}\ s$
r_a	Outer radius of pump rotor	m
r_i	Inner radius of pump rotor	m
T_M	Laplace transformed electric motor torque	N m
T_τ	Shear torque	N m
V_d	Dead volume at pump outlet	m^3
V_P	Displacement volume	m^3
x	Intermediate radius	m
z	Number of pistons	-
ν	Kinematic viscosity	$m^2\ s^{-1}$
ρ	Fluid density	$kg\ m^{-3}$
τ	Shear stress	Pa
Φ_M	Laplace transformed rotation angle of electric motor	rad
$\dot{\varphi}_P$	Angular velocity of pump	$rad\ s^{-1}$
φ_P	Rotation angle of pump	rad
Φ_P	Laplace transformed rotation angle of pump	rad
ω	angular frequency	$rad\ s^{-1}$

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