International Journal of Fluid Power 4 (2003) No. 1 pp. 17-30

DEVELOPMENT OF A COMPACT AND TUNEABLE VIBRATION COMPENSATOR FOR HYDRAULIC SYSTEMS

Josef Mikota(1 **and Herbert Reiter**(2

(1 *Rosenbauer International AG, Paschinger Str. 90, 4060 Leonding, Austria*
(2 *Johannes Kepler University of Linz, Department for Foundations of Machine Design, Altenbergerstr. 69, 4040 Linz, Austria*
josef.mikota@rosenbauer.com

Abstract

This publication is about vibration compensators for the attenuation of fluid flow pulsations in hydraulic systems. After a problem definition and an overview of conventional devices, a compact and adjustable *mass-spring resonator* featuring a *hydraulic spring* will be presented. The main advantages of this design are: simple and compact design, excellent noise attenuation characteristics, suitability for all pressure levels through mean pressure compensation and the possibility to alter the resonance frequency of the device in both a semi-active and active manner.

Besides the description of the working principle, the discussion of some phenomena occurring at high frequencies, the treatment of some design aspects, such as the optimisation of the sealing gap geometry, dimensioning etc. and a section devoted to compactness of vibration compensators, experimental results will be presented which prove the usefulness of the concept.

Keywords: flow pulsations, pressure pulsations, vibration compensation, hydraulic spring, noise attenuation, compactness

This manuscript was received on 28 August 2002 and was accepted after revision for publication on 10 March 2003

1 Introduction

Positive displacement pumps and motors as well as the utilisation of discontinuous control elements, such as fast switching valves in combination with so-called *switching type converters* (resonance converters Garstenauer et al, 1996; Scheidl and Riha, 1999, wave converters and motor converters) create significant flow and pressure pulsations in hydraulic circuits. These unpleasant effects may lead to excessive (acoustic) noise, compromised actuator dynamics or even to fatigue problems of components. Apart from measures to reduce fluid flow pulsations in the first place by optimised pump/motor designs (Ivantysyn and Ivantysynova, 2000) or the utilisation of damping devices (e.g. accumulators, in particular featuring a *Pulse tone* design), different forms of vibration compensators, such as $\lambda/4$ line silencers, $\lambda/4$ side-branch resonators, *Helmholtz resonators* etc. exist which may be used to attenuate flow induced pressure pulsations in hydraulic systems.

This paper is about the utilisation of vibration compensators for the attenuation of fluid flow pulsations in hydraulic systems. After a problem definition and an overview of conventional devices, a compact and adjustable *mass-spring resonator* featuring a *hydraulic spring* will be presented.

The main advantages of this concept are:

- Simple and compact design.
- Excellent noise attenuation characteristics.
- Suitable for all pressure levels by mean pressure compensation.
- Frequency tuning of the device is easily possible in both a semi-active and active manner.

The fundamental working principle of the proposed device will be outlined in section 2, followed by a theoretical analysis of some phenomena occurring at high frequencies in section 3. After the treatment of some aspects regarding the practical realisation, the discussion of both semi-active compensators and the compactness of vibration compensators, experimental results will be presented in section 6 which prove the usefulness of the concept proposed.

1.1 Sources of Hydraulic Noise

There are three main sources of noise in a hydraulic circuit: So-called *positive displacement pumps*, such as piston pumps, gear pumps etc. producing a non-steady flow stream in the first place, *positive displacement motors* and the utilisation of *discontinuous control elements* in the hydraulic circuit. Some details concerning these elements will be discussed in the following paragraphs.

By definition, positive displacement machines are characterised by a finite number of displacement elements, i.e. pistons in case of piston pumps or motors, teeth in gear pumps/motors etc. Hence, the flow stream originating at a pump is not constant over time, i.e., it is characterised by some harmonics repeating at the pump frequency and integer multiples of it. In addition to that, these periodic flow pulsations interact with other elements of the hydraulic circuit, such as piping, valves etc. or even notably with the load itself and complex pressure waves are formed within the hydraulic circuit (Dodson et al, 1998). To the outside world, these fluctuations become notable as audible noise, vibrations of the pipe-work or fatigue problems of components.

According to simple models (Backé, 1994) describing the flow generation mechanism of piston pumps, the first harmonic of the flow pulsation of pumps with an odd number of piston elements would be at $f_1 = 2\, n_{\mathrm{Pist}}\, n$. However, taking the compressibility of the fluid and port timing into account, it is reasonable to say that the first harmonic is always at $f_1 = n_{\mathrm{Pist}}\, n$.

As explained in Garstenauer et al (1996) and Scheidl and Riha (1999), a fast (digital) switching valve (SV) in combination with a mechanical oscillator – the so-called *Resonance converter* – may be utilised to effectively adjust the flow rate in hydraulic systems by periodically switching between a high pressure port (P), a tank line (T) and an output port (A) (see Fig. 1). The flow rate in the output line is thereby controlled by the pressure levels in the pressure and tank lines, the switching frequency *f*, the relative period of the pressure-on-phase and the relative period of the output-on-phase.

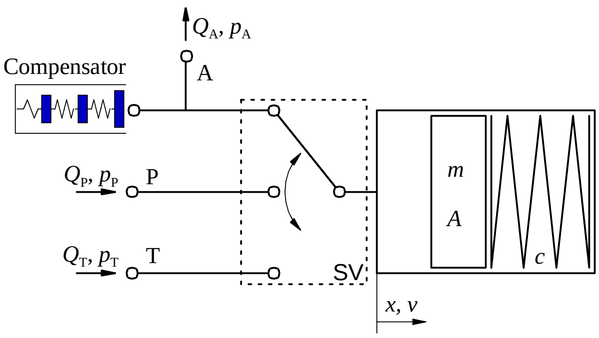


Fig. 1: *Resonance converter*

Although this method of hydraulic control is more energy efficient than conventional resistance control, there is still the problem that without the use of appropriate vibration compensators, considerable pressure transients are introduced in the hydraulic system due to the discontinuous nature of the switching process.

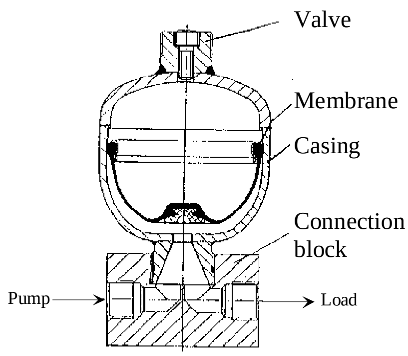
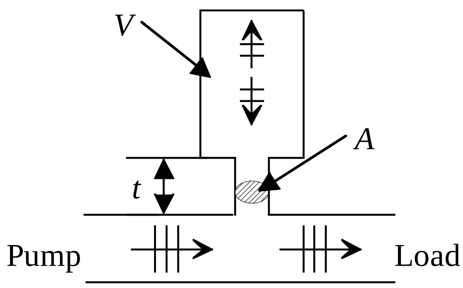
1.2 Conventional Devices

As the attenuation of pressure pulsations, otherwise also commonly known as fluid borne noise, has been a topic in hydraulic engineering for a long time, numerous devices for the suppression of hydraulic noise exist.

However, as can be seen in Table 1 most conventional devices have one or more of the following shortcomings: complex or expensive design, bulky design (e.g. $\lambda/4$ resonators), no simple method of frequency tuning, unsystematic parameter selection (e.g. for accumulators the relevant parameters are volume and precharge pressure), therefore development is done by *trial and error*, and low noise attenuation performance.

Furthermore, no conventional devices but the *Multi-volume resonators* are capable of attenuating a base harmonic and all integer multiples of it up to a certain order.

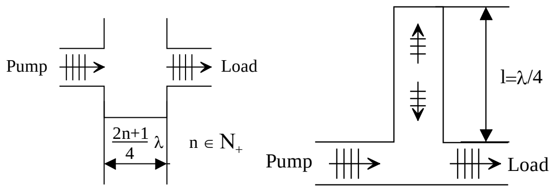
Table 1: Proprietary hydraulic noise suppressors

<table>
<tr><th>Accumulator</th></tr>
<tr><td>+ simple device, readily available
- low noise attenuation performance (especially at high frequencies)
- parameter selection by experimental means

</td></tr>
<tr><th>Helmholtz resonator</th></tr>
<tr><td>+ simple device
+ simple frequency tuning
- attenuation of 1 harmonic only
- builds rather bulky if attenuation over broad frequency range is required

</td></tr>
<tr><th>In-line noise suppressor</th></tr>
</table>

\+ fairly simple device
\- expensive

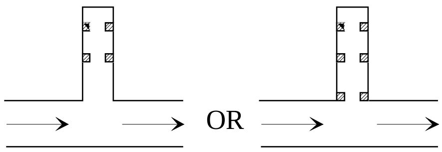
λ/4 Silencers

\+ simple devices
\- attenuation of odd order harmonics only
\- build rather long for low frequencies



Multiple volume resonators

\+ fairly simple devices
\+ may be tuned in such a way to attenuate integer multiples of a base harmonic
\- complex tuning operation



Variable resonance type side branch resonator

Mulit DOF type Helmholtz resonator

2 Vibration Compensator Featuring a Hydraulic Spring

2.1 Working Principle

The device depicted in Fig. 2 is connected to the main hydraulic system (1). A working piston (3) with mass m is excited by the pressure pulsation $p(t)$ in the main system. Furthermore, the neck of the working piston seals off (e.g. by a gap seal between the piston (3) and the housing (0) as depicted in Fig. 2) the hydraulic volume (2) – also called the *hydraulic spring* – from the main system (1) and permits only the (slow) balancing of the mean pressures between (1) and (2).

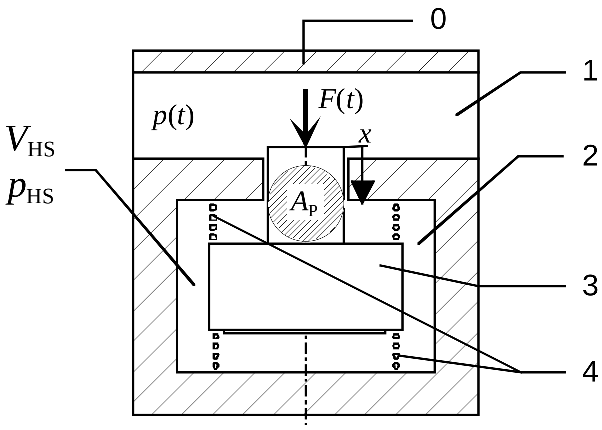


Fig. 2: *Proposed vibration compensator*

In order to keep the working piston (3) in a centred position after the mean pressure compensation, centring springs (4) may be used. Due to the pressure pulsation $p(t)$ in the main system or the excitation force

$$F(t) = \big(p(t) - p_{HS}(t)\big) A_P \tag{1}$$

Respectively, the piston (3) is accelerated and a displacement $x(t)$ of the piston (3) takes place. In effect, this yields an increased pressure p_{HS} in the hydraulic spring (2). Neglecting the cross flow between the hydraulic spring (2) and the main system (1) and assuming that $V_{HS} >> A_P\, x$, the evolution equation for p may be simplified to

$$\dot{p} = \frac{B'_{Tot}}{V_{HS} - A_P\, x} A_P \dot{x} \approx \frac{B'_{Tot}}{V_{HS}} A_P\, \dot{x} \tag{2}$$

Accordingly, a displacement Δx of the piston yields an increase in pressure in the hydraulic spring

$$\Delta p_{HS} = \frac{B'_{Tot}}{V_{HS}} A_P\, \Delta x \tag{3}$$

Multiplying Eq. 3 by the piston area A_P yields the stiffness of the hydraulic spring c_{HS}

$$\Delta F = A_P\, \Delta p_{HS} = \underbrace{\frac{B'_{Tot}\, A_P^{\,2}}{V_{HS}}}_{c_{HS}} \Delta x \tag{4}$$

The behaviour of the system represents a resonator with mass m (the mass of the working piston (3)[1)]), stiffness c_{HS} (due to the stiffness of the hydraulic spring (2)[2)]) and damping d due to shear stresses in the fluid and leakage, where B'_{Tot} represents the combined bulk modulus of hydraulic oil considering the flexibility of the surrounding enclosure, A_P represents the area of the working piston and V_{HS} represents the oil volume in the hydraulic spring.

As known in Engineering Mechanics (Hunt, 1979), a properly tuned secondary system called the vibration

[1)] Neglecting both the proportionate mass of the oscillating oil column in the hydraulic spring and the proportionate mass of the centring springs.

[2)] Neglecting the stiffness of the centring springs.

compensator may be used to greatly attenuate the effects of an excitation of the primary system if the frequency of excitation Ω is close to the natural frequency ω of the vibration compensator. In our case, the excitation is given by the flow pulsation, the vibration compensator is represented by the presented device and the primary system is the main hydraulic system.

3 Resonance Frequency

3.1 Basic Model

Neglecting the influence of the centring springs, the natural frequency of the vibration compensator is given as

$$\omega_0 = \sqrt{\frac{c_{\mathrm{HS}}}{m}} = A_{\mathrm{P}}\sqrt{\frac{B'_{\mathrm{Tot}}}{V_{\mathrm{HS}}\, m}} \tag{5}$$

Basic consideration, i.e. considering the stiffness but not their respective masses, of the centring springs yields

$$\omega = \sqrt{\frac{c_{\mathrm{HS}} + 2\,c_{\mathrm{CS}}}{m}} \tag{6}$$

3.2 Advanced Model[3)]

Although the equations presented in the previous section are sufficiently accurate at low frequencies (e.g. prototype I – see section 6.1), additional effects need to be considered at higher frequencies. Some of these effects will be discussed in the subsequent paragraphs.

Linear damping

Assuming linear damping in the vibration absorber, the resonance frequency ω of the device may be written as

$$\omega = \sqrt{1-\zeta^2}\,\omega_0 \tag{7}$$

where the dimensionless damping ratio is

$$\zeta = \frac{d}{2\sqrt{m\, c_{\mathrm{HS}}}} \tag{8}$$

Since the influence of leakage between the piston and the housing does appear as additional damping, only the influence of shear stresses in the sealing gap between the housing and the piston will be investigated in this paragraph (see section 4.1 for a more detailed investigation of leakage). The coefficient of damping due to shear stresses between the piston and the housing may be written as

$$d = \frac{D\,\pi\, l_{\mathrm{Gap}}\,\eta}{s_{\mathrm{Gap}}} \tag{9}$$

since

$$F_{\mathrm{Damp}} = d\, v = D\,\pi\, l_{\mathrm{Gap}}\,\tau \tag{10}$$

where

$$\tau = \eta\frac{\partial v}{\partial z} \tag{11}$$

Wave propagation, mass of the hydraulic spring and damping

In order to investigate the effects of wave propagation, mass of a hydraulic spring and (nominal) damping[4)], the simplest possible arrangements – a system with a hydraulic spring – depicted in Fig. 4 will be investigated. The approach and the nomenclature presented here are taken from Ingard (1988).

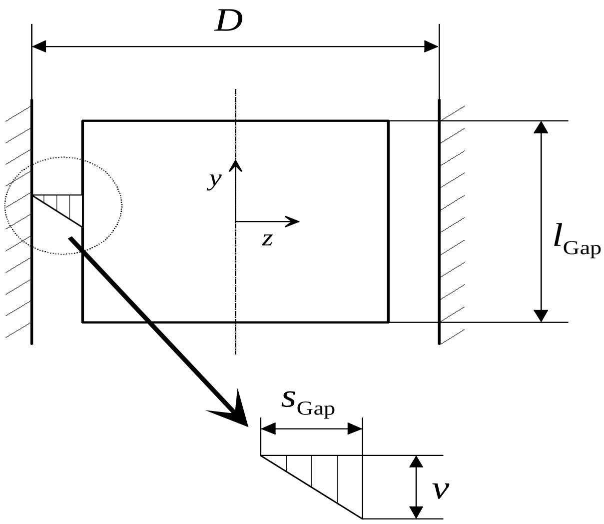


Fig. 3: *Damping in sealing gap*

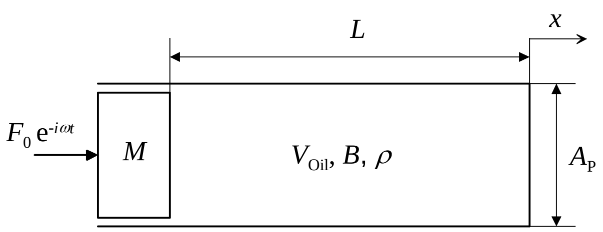


Fig. 4: *Resonances of hydraulic system*

Using the common notation for the wave number k, the speed of wave propagation v and the wave resistance Z for hydraulic systems

$$k = \frac{\omega}{v} \tag{12}$$

$$v = \sqrt{\frac{B}{\rho}} \tag{13}$$

3) In order to make the following descriptions and calculations as clearly as possible, the influence of c_{CS} on ω will be subsumed in c_{HS} ($c_{\mathrm{HS}} \rightarrow c_{\mathrm{HS}} + 2\, c_{\mathrm{CS}}$).

4) Although most texts do not consider damping in combination with wave propagation, a combined treatment makes particularly sense if one is interested in the quantitative response close to resonance and not just in the natural frequency itself.

$$Z = \rho v = \sqrt{B \rho} \tag{14}$$

the complex amplitudes of pressure and velocity assuming linear wave propagation may be written as

$$p(x) = A^* e^{ikx} + B^* e^{-ikx} \tag{15}$$

$$u(x) = \frac{1}{Z}\left(A^* e^{ikx} - B^* e^{-ikx}\right) \tag{16}$$

where consideration of the boundary condition

$$u(x)\big|_{x=0} \equiv u(0) = 0 \tag{17}$$

yields

$$p(x) = 2A^* \cos(k\,x) \tag{18}$$

$$u(x) = \frac{1}{Z} 2\,A^* i \sin(k\,x) \tag{19}$$

since

$$\begin{aligned} u(0) &= \frac{1}{Z}\left(A^* - B^*\right) = 0 \\ &\Rightarrow A^* = B^* \end{aligned} \tag{20}$$

Taking the *Fourier* transform to the equation describing the dynamics of mass M yields

$$M\left(-i\omega\right) u\left(-L\right) + d\,u\left(-L\right) = F_0 - p(-L) A_P \tag{21}$$

where

$$d = 2\zeta \sqrt{c_{HS}\, M} \tag{22}$$

Consequently, the complex amplitude A^*

$$A^* = \frac{F_0}{DEN} \tag{23}$$

and the amplitudes of velocity $u(-L)$ and displacement $\xi(-L)$ respectively may be written as

$$u(-L) = \frac{-2F_0 i \sin(kL)}{Z\,DEN} \tag{24}$$

$$\zeta(-L) = \frac{u(-L)}{-i\omega} = \frac{2F_0 \sin(kL)}{Z\,\omega\,DEN} \tag{25}$$

where

$$\begin{aligned} DEN = {} & -2\omega \frac{M}{Z} \sin(kL) - \\ & 4i\zeta \frac{\sqrt{M\,c_{HS}}}{Z} \sin(kL) + 2\cos(kL) A_P \end{aligned} \tag{26}$$

The resonance frequency of the hydraulic system depicted in Fig. 4 may be evaluated by taking the maximum of Eq. 25.

Channel length from main system to absorber

Making use of the analogy[5)] between hydraulic and mechanical systems, the channel from the main system to the vibration absorber may be approximated by an additional (mechanical) system which also shifts the natural frequencies of the coupled system.

Neglecting damping, the dynamics of the coupled system may be described by

$$\mathbf{M}\;\;\ddot{\mathbf{x}} + \mathbf{C}\;\;\mathbf{x} = \mathbf{F} \tag{27}$$

where

$$\mathbf{M} = \begin{pmatrix} m_C & 0 \\ 0 & m_{HS} \end{pmatrix}, \quad \mathbf{C} = \begin{pmatrix} c_C & -c_C \\ -c_C & c_C + c_{HS} \end{pmatrix} \quad \mathbf{F} = \begin{pmatrix} F_0 \\ 0 \end{pmatrix} \tag{28}$$

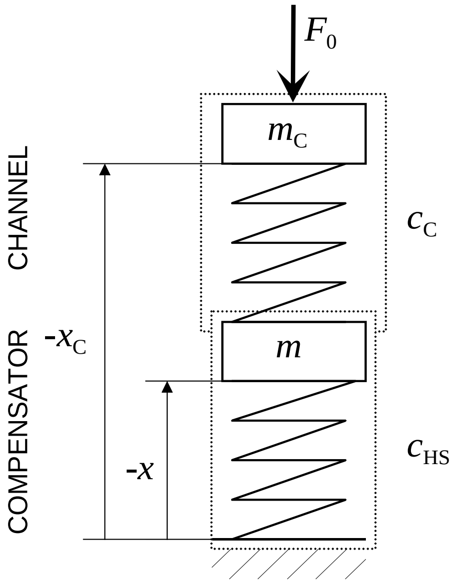
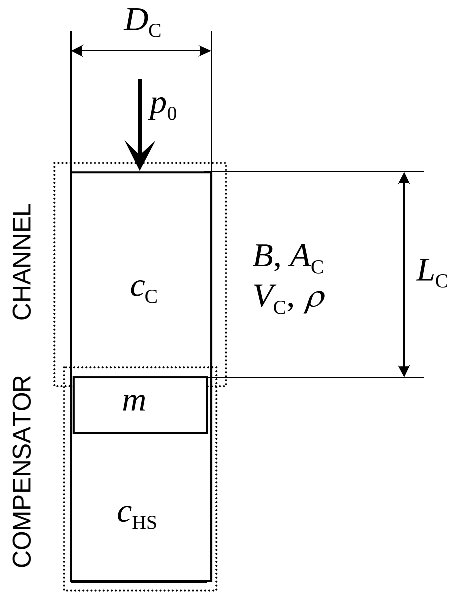


Fig. 5a: *Coupled oscillation: Channel – vibration absorber Mechanical system*



5) Also the effects of wave propagation, mass of a hydraulic spring and damping could have been explained using the analogy between mechanical and hydraulic systems.

Fig. 5b: *Coupled oscillation: Channel – vibration absorber Hydraulic system*

and

$$m_{\mathrm{C}} = V_{\mathrm{C}}\,\rho \tag{29}$$

$$V_{\mathrm{C}} = A_{\mathrm{C}}\,L_{\mathrm{C}} \tag{30}$$

$$A_{\mathrm{C}} = \frac{D_{\mathrm{C}}^{2}\pi}{4} \tag{31}$$

$$c_{\mathrm{C}} = \frac{B\,A_{\mathrm{C}}^{2}}{V_{\mathrm{C}}} \tag{32}$$

The natural frequencies of the coupled system Eq. 27 may be calculated as the square-root of the eigenvalues of the characteristic matrix $\tilde{\mathbf{A}}$, where

$$\tilde{\mathbf{A}} = \mathbf{C}\cdot\mathbf{M}^{-1} \tag{33}$$

4 Design Aspects

So far, there is no clear rule of how big a vibration compensator needs to be for a given flow pulsation Q_{P}, thus the following very simple procedure was developed.

Assume that a vibration absorber[6] is parametrised in the way depicted in Fig. 6 with independent design parameters D, H and d_{P}. Also assuming that the height of the hydraulic spring is equal to the diameter[7], $H = D$, the mass m, the volume of the hydraulic spring V_{HS} and hence also the resonance frequency of the absorber ω_0 are fully expressed as functions of the independent parameters.

However, since the resonance frequency ω_0 of the absorber is defined by two parameters, one additional equation needs to be found to fully constrain the system.

[6] The parameterisation follows roughly the design of the first prototype (see Mikota, 2002).

[7] For the sake of a compact shape of the compensator.

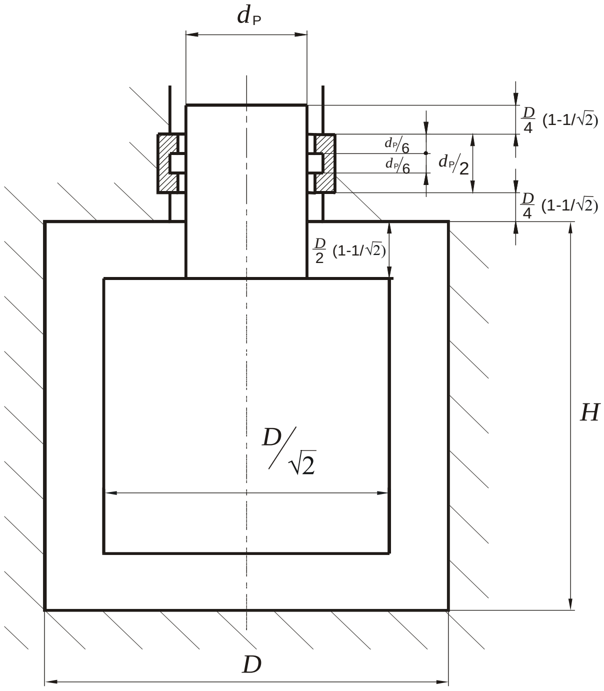


Fig. 6: *Parameterised vibration absorber*

4.1 System Dynamics

Neglecting the eccentricity of the piston in the sealing gap, the equations describing the dynamic behaviour of the system depicted in Fig. 7 are

$$m\,\ddot{x} + d\,\dot{x} + 2\,c_{\mathrm{CS}}\,x = A_{\mathrm{P}}\left(p_{\mathrm{Sys}} - p_{\mathrm{HS}}\right) \tag{34}$$

$$\dot{p}_{\mathrm{HS}} = \frac{B}{V_{\mathrm{HS}}}\left(A_{\mathrm{P}}\dot{x} + \frac{d_{\mathrm{P}}\pi\,s_{\mathrm{Gap}}^{3}}{12\eta\,l_{\mathrm{Gap}}}\left(p_{\mathrm{Sys}} - p_{\mathrm{HS}}\right)\right) \tag{35}$$

where

$$d = \frac{l_{\mathrm{Gap}}\,\eta\,d_{\mathrm{p}}\pi}{s_{\mathrm{Gap}}} \tag{36}$$

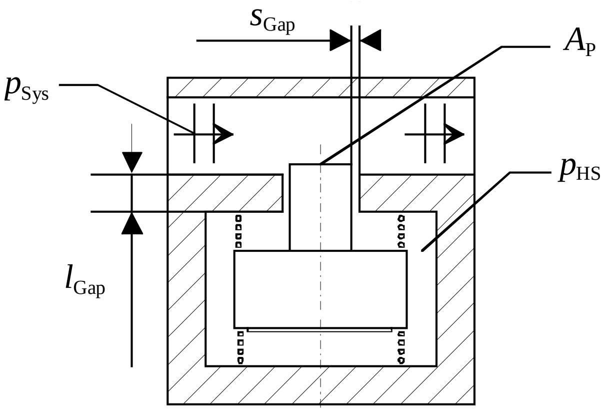


Fig. 7: *System overview: Sealing gap optimisation*

In the frequency domain, the transfer function describing the piston displacement $\hat{x}(s)$ as a function of the system pressure $\hat{p}_{\mathrm{Sys}}(s)$ is

$$G(s) = \frac{\hat{x}(s)}{\hat{p}_{\mathrm{Sys}}(s)} = \frac{12\, s V_{\mathrm{HS}}\, \eta\, l_{\mathrm{Gap}}\, d_{\mathrm{P}}^{\,2}\, \pi\, s_{\mathrm{Gap}}}{DEN} \tag{37}$$

where

$$\begin{aligned} DEN = {} & 48 s^3 V_{\mathrm{HS}}\, \eta\, l_{\mathrm{Gap}}\, m\, s_{\mathrm{Gap}} + 48 s^2 V_{\mathrm{HS}}\, \eta^2\, l_{\mathrm{Gap}}^{\,2}\, d_{\mathrm{P}}\, \pi + {} \\ & 4 s^2 B\, d_{\mathrm{P}}\, \pi\, s_{\mathrm{Gap}}^{\,4}\, m + 96\, s V_{\mathrm{HS}}\, \eta\, l_{\mathrm{Gap}}\, c_{\mathrm{CS}}\, s_{\mathrm{Gap}} + {} \\ & 4 s B\, d_{\mathrm{P}}^{\,2}\, \pi^2\, s_{\mathrm{Gap}}^{\,3}\, \eta\, l_{\mathrm{Gap}} + 3 s B\, d_{\mathrm{P}}^{\,4}\, \pi^2\, \eta\, l_{\mathrm{Gap}}\, s_{\mathrm{Gap}} + {} \\ & 8 B\, d_{\mathrm{P}}\, \pi\, s_{\mathrm{Gap}}^{\,4}\, c_{\mathrm{CS}} \end{aligned} \tag{38}$$

Eq. 37 may now be used to optimise the sealing gap of the absorber.

4.2 Optimisation of Sealing Gap

Evaluating the magnitude of $G(s)$ at the design frequency of the compensator for different parameters s_{Gap} yields an optimised sealing gap.

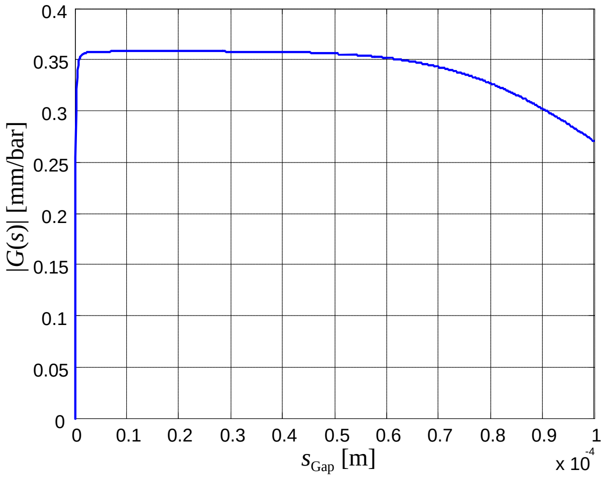


Fig. 8: *|G(s)| for compensator specified in Table 7*

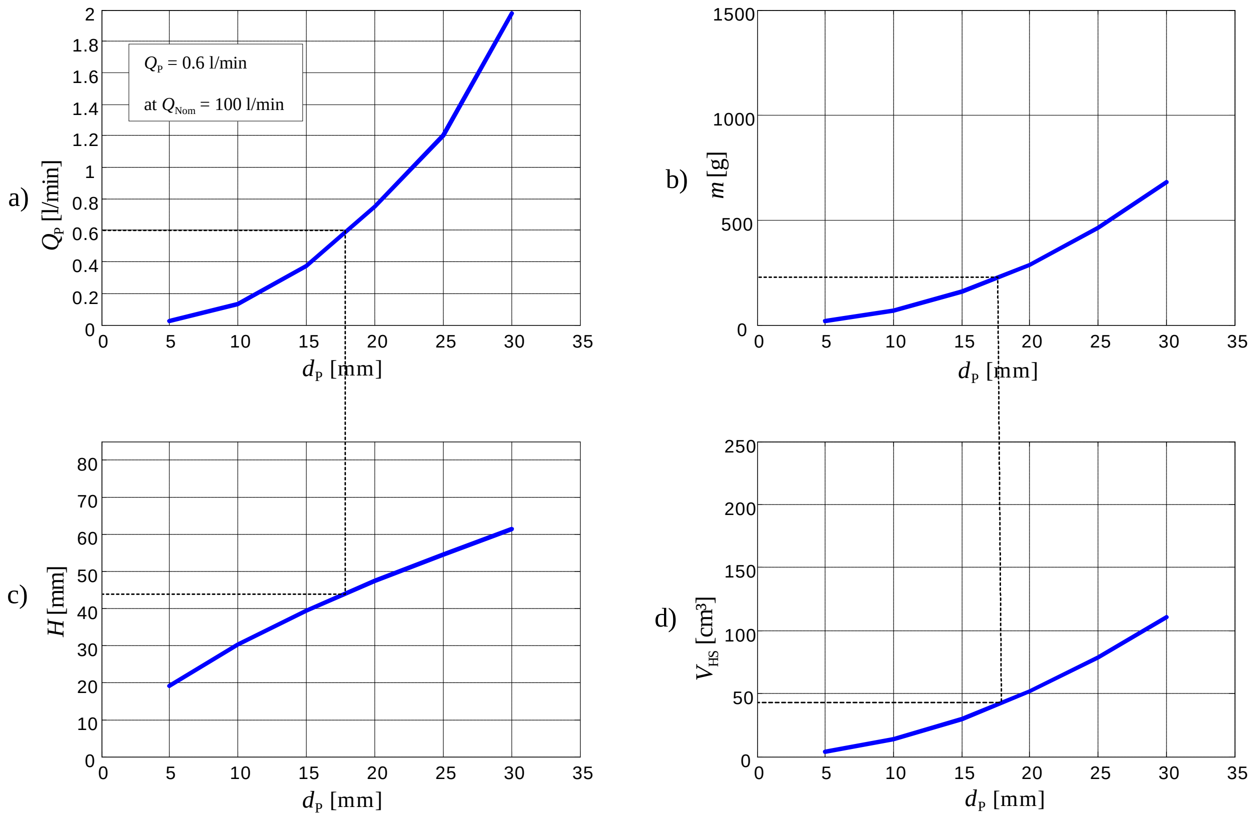


Fig. 9: *Dimensioning of compensator with f_{Res}=450 Hz (assuming B=12000 bar and p_{Sys}=2 mbar)*

4.3 Dimensioning

Assuming a purely sinusoidal motion of the piston with frequency ω_0, the flow Q_{P} taken-up by the displacement of the compensator due to the pressure pulsation p_{Sys} may be written as

$$|x| = |G(s)|\,|p_{\mathrm{Sys}}| \tag{39}$$

$$\Rightarrow |\dot{x}| = |x|\,\omega \tag{40}$$

$$\Rightarrow |Q_{\mathrm{P}}| = A_{\mathrm{P}}\,|\dot{x}| = A_{\mathrm{P}}\,\omega\,|G(s)|\,|p_{\mathrm{Sys}}| \tag{41}$$

However, since only one additional equation was missing according to the last paragraph of section 4, Eq. 37 in combination with Eq. 41 and the parameterisation given in Fig. 6 are therefore sufficient to fully constrain the compensator for a given resonance frequency ω_0, flow pulsation Q_{P} and permissible pressure pulsation p_{Sys}.

As a worked out example, the dimensioning charts for compensators with f_{Res} = 450 Hz and p_{Sys} = 2 mbar using the parameterisation given in Fig. 6 are depicted in Fig. 9.

Using these charts, the general procedure of absorber dimensioning is as follows:

- Select d_{P} according to the pulsating flow stream Q_{P}

at frequency f_{Res} (see Fig. 9a)

- Read height H of the hydraulic spring (see Fig. 9c)
- For a given diameter d_{P} of the piston, read the mass of the piston m (see Fig. 9b) and the volume V_{HS} of the hydraulic spring (see Fig. 9d)

4.4 Semi-active Compensators

Although various methods of frequency tuning are possible (Mikota and Manhartsgruber, 2001), perhaps the most straight forward frequency tuning principle makes use of the very nature of the hydraulic spring itself, i.e., by changing the volume V_{HS} in an either discrete or continuous manner. The schematics of systems featuring continuous and discrete adaptation respectively are depicted in Fig. 10.

In the case of continuous adaptation (see Fig. 10a and b), the volume V_{HS} of the hydraulic spring may be modified by means of an actuator (5). Switching valves (SV1) and (SV2), as depicted in Fig. 10a, possibly in combination with a hydraulic orifice (D1) may be utilised for a quicker balancing of mean pressures between the main system and the hydraulic spring. Alternatively, the resonance frequency of the compensator may be modified in a discrete manner[8)] too (see Fig. 10c). In that case, a simple switching valve (SV3) may be used to connect the initial volume V_{HS} of the hydraulic spring to an additional volume (5). For the sake of quicker settling times of the resonator after frequency modification, an additional switching valve (SV4) between the main system and volume (5) may be utilised.

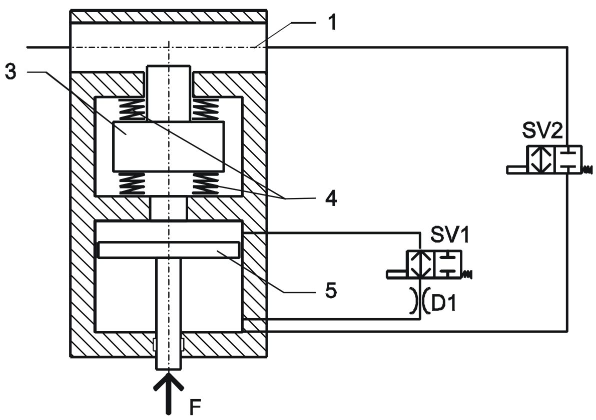


Fig. 10a: *Frequency tuning by modification of V_{HS}: Continuous adaptation (hydraulic drive)*

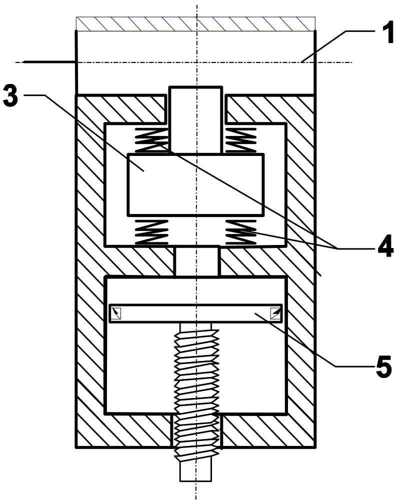


Fig. 10b: *Frequency tuning by modification of V_{HS}: Continuous adaptation (mechanical drive)*

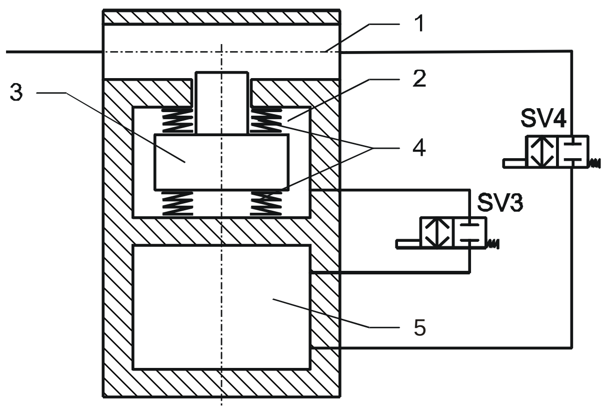


Fig. 10c: *Frequency tuning by modification of V_{HS}: Discrete adaptation*

5 Compactness of Vibration Absorbers

5.1 Attenuation Performance

The equation describing the dynamics of the analogous mechanical system to the hydraulic system Fig. 11 neglecting both damping and the "end correction" of the aperture neck (Ingard, 1953) reads as

$$m\,\ddot{x} + c\,x = p\,A_{\mathrm{P}} \tag{42}$$

$$m = \underbrace{\frac{d_{\mathrm{P}}^{\,2}\pi}{4}}_{A_{\mathrm{P}}} l_{\mathrm{Neck}}\,\rho_{\mathrm{Oil}} \tag{43}$$

$$c = \frac{B\,A_{\mathrm{P}}^{\,2}}{V} \tag{44}$$

8) More than one additional volume may also be utilized thus allowing (digital) tuning of the compensator.

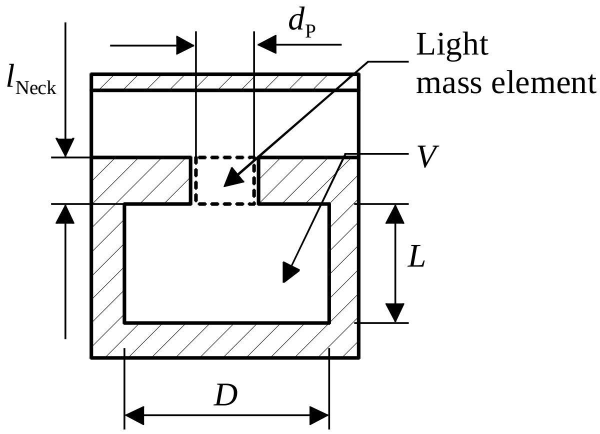


Fig. 11: *Hydraulic system*

Hence, in the frequency domain the displacement $\hat{x}$ of the mass m due to the pressure pulsation $\hat{p}$ is

$$\hat{x} = \frac{\hat{p}\, A_P}{m s^2 + c} \tag{45}$$

$$s = i\omega \tag{46}$$

and the velocity of the mass may be written as

$$u = s\,x = \frac{s\,p\,A_P}{m s^2 + c} \tag{47}$$

Consequently, the flow taken up by the compensator is

$$\hat{Q} = A_P\,\hat{u} = \frac{s\,\hat{p}\,A_P{}^2}{m s^2 + c} \tag{48}$$

According to Kojima and Edge (1994)[9], the attenuation performance of vibration compensators may be described by the inverse of the entry impedance $\left|1/Z_r\right|$, where

$$\frac{1}{Z_r} = \frac{\hat{Q}}{\hat{p}} = \frac{s\,A_P{}^2}{m s^2 + c} \tag{49}$$

$$\left|\frac{1}{Z_r}\right| = \frac{\omega\,A_P{}^2}{\left|-\omega^2 m + c\right|} \tag{50}$$

[9] Note that Eq. 16 in Kojima, Edge (1994) contains an obvious typing error.

5.2 Comparison of Absorbers with Identical Dynamic Behaviour

The compactness of classical Helmholtz resonators (HR), modified Helmholtz resonators featuring a heavy mass element (MHR) and the resonators presented in Sec. 2 (MR) will be compared for resonance frequencies of f_{Res} = 225 Hz and f_{Res} = 450 Hz. For the MR, the parameterisation follows Fig. 6 and the common parameters used in the comparison are given in Table 2.

Table 2: Common parameters

d_P	20	mm
B	16000	bar
ρ_{Fe}	7800	kg/m³
ρ_{Oil}	850	kg/m³

Table 3: Compactness of absorbers with identical dynamics (f_{Res} = 225 Hz)

	m_{Osc} [g]	m_{Tot} [g]	l_{Neck} [mm]	l_{Max} [mm]	V_{Oil} [l]	V_{Tot} [l]	D [mm]	Diag [mm]
MR			19	83		0.21	63	104
MHR	623	731	254	309	0.13		54	314
HR			2334	2388		0.86		2389

Table 4: Compactness of absorbers with identical dynamics (f_{Res} = 450 Hz)

	m_{Osc} [g]	m_{Tot} [g]	l_{Neck} [mm]	l_{Max} [mm]	V_{Oil} [l]	V_{Tot} [l]	D [mm]	Diag [mm]
MR			17	67		0.1	49	83
MHR	327	378	133	176	0.06		43	181
HR			1225	1269		0.45		1269

5.3 Comparison of Absorbers Featuring Identical Diameter of the Hydraulic Spring

Since the diameter D of the hydraulic spring of MRs is greater than those of MHRs and HRs (see Table 3 and 4), a comparison will also be given for identical diameter D of the resonators. Since $|1/Z_r|$ is unbounded at $f = f_{Res}$ (resonance of an undamped system), the value of $|1/Z_r|$ was evaluated at $f = 0.9\, f_{Res}$.

Table 5: Compactness of absorbers with identical diameter D (f_{Res} = 225 Hz, D = 63mm)

	m_{Osc} [g]	m_{Tot} [g]	l_{Neck} [mm]	l_{Max} [mm]	V_{Oil} [l]	V_{Tot} [l]	Diag [mm]	$\|1/Z_r\|$ [m³/Pas]
MR	623	731	19	83	0.13	0.21	104	5.3E-10
MHR	394	564	161	224	0.2	0.25	233	8.4E-10
HR			1475	1538		0.66	1540	

Table 6: Compactness of absorbers with identical diameter D (f_{Res} = 450 Hz, D = 49mm)

	m_{Osc} [g]	m_{Tot} [g]	l_{Neck} [mm]	l_{Max} [mm]	V_{Oil} [l]	V_{Tot} [l]	Diag [mm]	$\|1/Z_r\|$ [m³/Pas]
MR	327	378	17	67	0.06	0.1	83	5.1E-10
MHR	210	290	86	135	0.09	0.12	144	7.9E-10
HR			786	835		0.34	837	

6 Experimental Results

After designing and building the prototypes I and II (see Table 7 and 8 for the design parameters, Fig. 14 for the actual design of prototype II and Fig. 15 and 17 for photos) featuring natural frequencies of f_{Res} = 225 Hz and f_{Res} = 450 Hz respectively, the prototypes were subject of an experimental investigation (for a more complete reference see Reiter, 2002) in order to specify the insertion loss (Beranek et al, 1992)

$$IL = 20 \log \frac{p_{i\,\mathrm{FFT}}}{p_{i\,\mathrm{RefFFT}}} \tag{51}$$

of the devices.

In order to receive practically relevant data, the volume V_{HS} of the compensator was kept constant during the experiments and the pulsations were created by a 9 piston displacement axial piston-pump driven at a nominal speed of n = 1500/min (see Fig. 12, 13 and 16).

6.1 Prototype I (f_{Res} ≈ 225 Hz)

Table 7: Parameters of prototype I

Piston diameter	d_{P}	20	mm
Length of gap seal	l_{Gap}	5	mm
Sealing gap	s_{Gap}	17	µm
Piston mass	m	298	g
Oil volume of hydr. spring	V_{HS}	195…285	cm^3
Stiffness centring springs	c_{CS}	4.24	N/mm

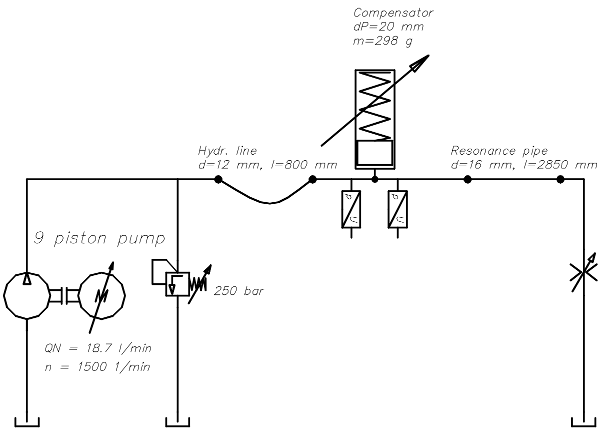


Fig. 12: *Experimental setup: Prototype I*

6.2 Prototype II (f_{Res} ≈ 450 Hz)

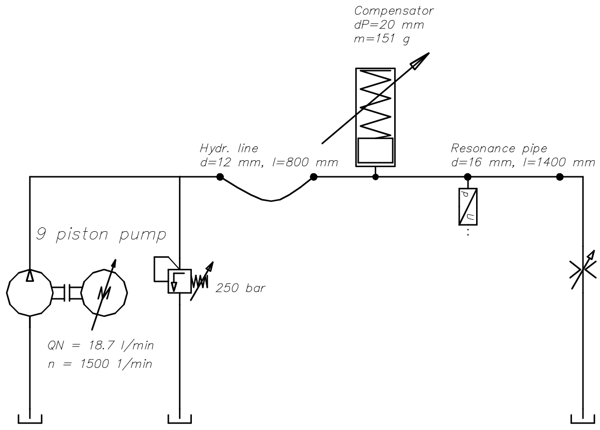


Fig. 13: *Experimental setup: Prototype II*

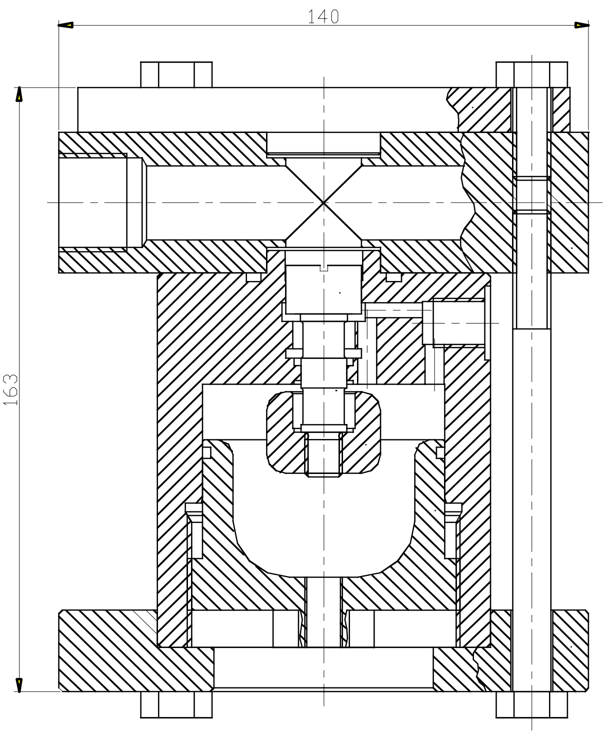


Fig. 14: *Design of prototype II*

Table 8: Parameters of prototype II

Piston diameter	d_{P}	20	mm
Length of gap seal	l_{Gap}	5	mm
Sealing gap	s_{Gap}	17	µm
Piston mass	m	151	g
Oil volume of hydr. spring	V_{HS}	50…130	cm^3
Stiffness centring springs	c_{CS}	4.24	N/mm

Fig. 15a: *Pictures of prototype II: Front view*

Fig. 15b: *Pictures of prototype II: Exploded view*

6.3 Prototypes I and II

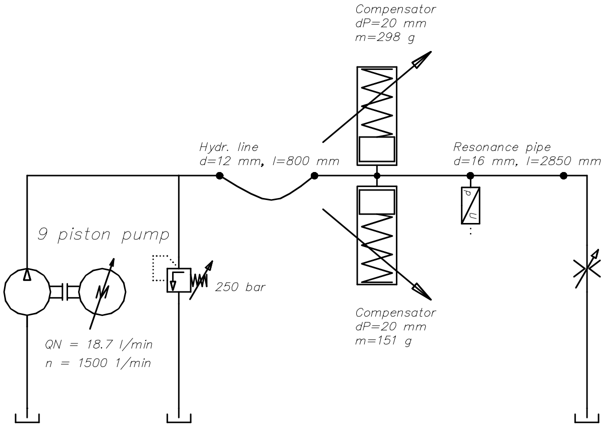


Fig. 16: *Experimental setup: Prototypes I + II*

As can be seen in Fig. 18, both prototypes show excellent noise attenuation characteristics of *IL* = -22 dB and *IL* = -29 dB at resonance frequencies of f_{Res} = 225 Hz and f_{Res} = 450 Hz respectively. In case of systems where both harmonics are somehow problematic, the use of both compensators may be an option.

Even in this case and although the resonance frequency of the first compensator appears to be slightly shifted (the reason of which has not yet been found), the noise attenuation is excellent with *IL* = -18 dB and *IL* = -29 dB at the first and second order harmonic respectively.

Fig. 17: *Arrangement of prototypes I + II*

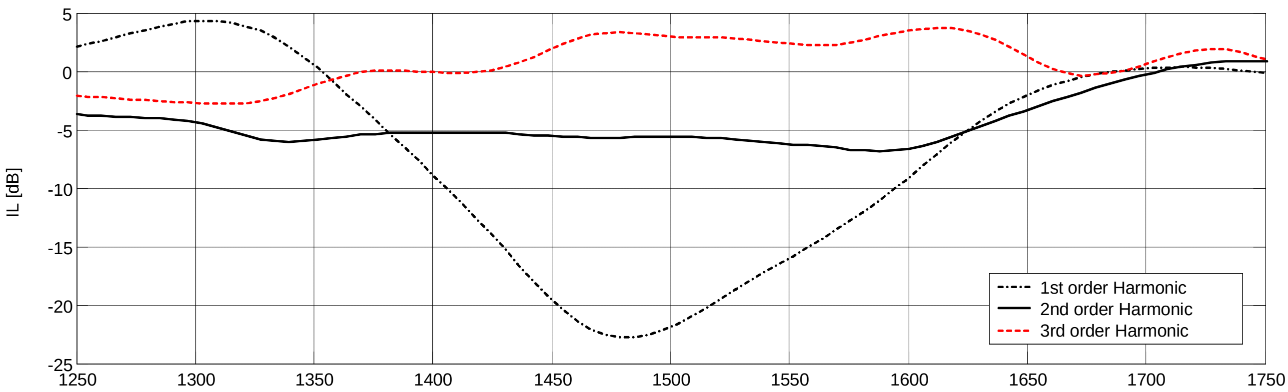


Fig. 18a: *Attenuation performance of prototypes I and II: Prototype I active, prototype II inactive*

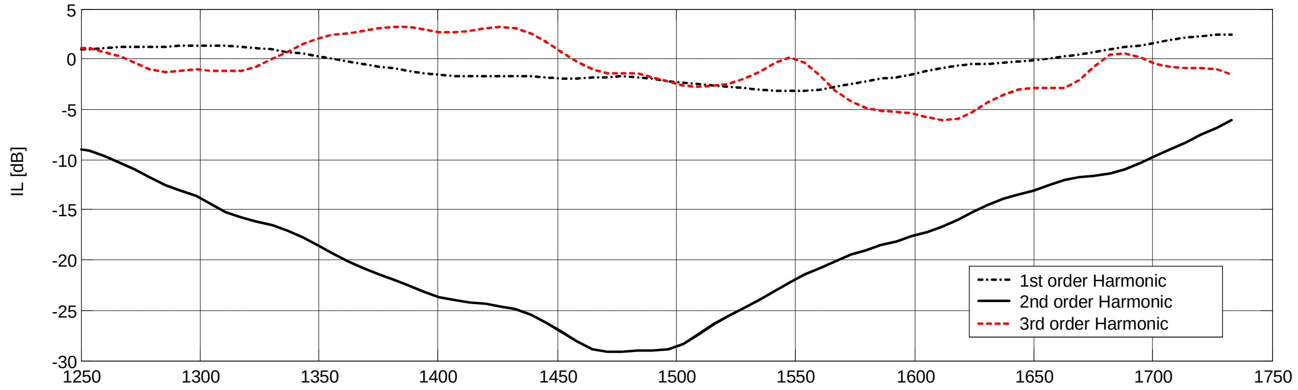


Fig. 18b*:* *Attenuation performance of prototypes I and II: Prototype II active, prototype I inactive*

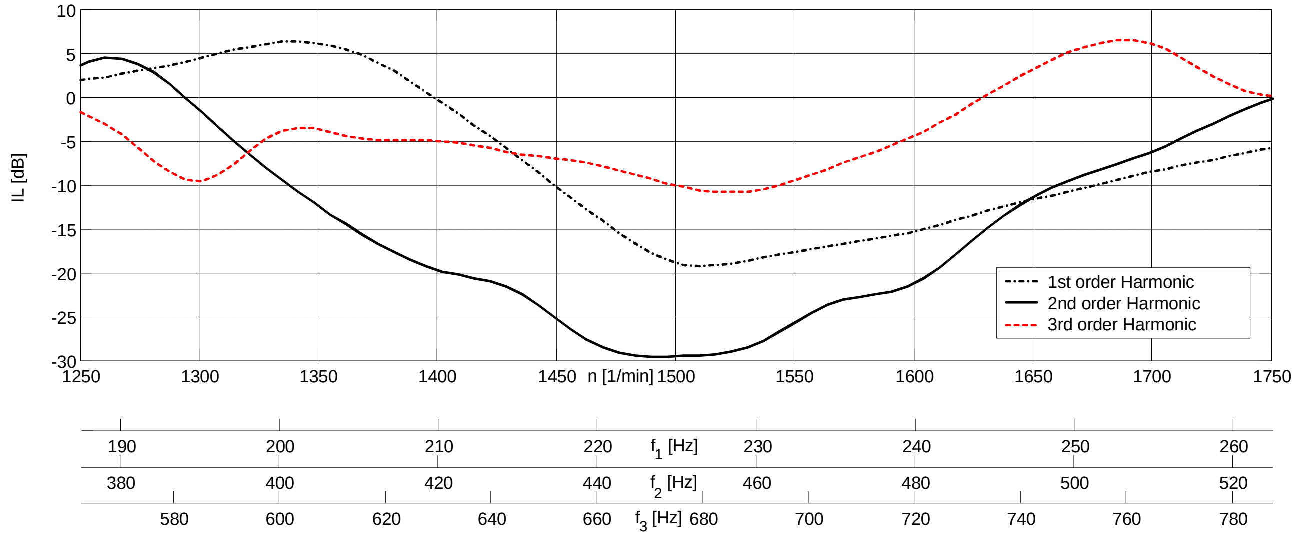


Fig. 18c: *Attenuation performance of prototypes I and II: Prototypes I and II active*

6.4 Determination of the Entry Impedance Z_r

Since the insertion loss is strongly dependent on the wave propagation characteristics of both the upstream and downstream line including the pump source impedance and the load, respectively, besides the resonator itself, as well as the harmonic frequency in question, the entry impedances Z_r of the vibration absorbers – prototype I and II – were determined experimentally using the test procedure described in Kojima et al (2000).

The first step in this procedure was the determination of the source impedance of the pump Z_s by the “2 pressures/2 systems” method (see Fig. 19 and Kojima, 1992) at a given speed n_{Pump} = 1500/min. Therefore, the pressures p_0 and p_1 were recorded for two different distances x, namely x = 140 mm (system A) and x = 360 mm (system B). With this information, the pump impedance Z_s was determined for 2 harmonics (see Table 9).

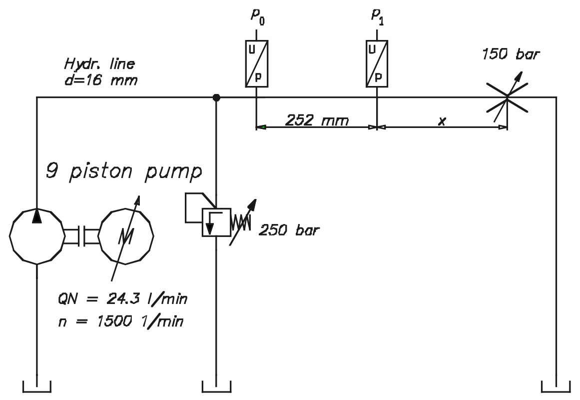


Fig. 19: *Determination of pump impedance Z_s*

Table 9: Pump impedance Z_s

f [Hz]	$\lvert Z_s \rvert$ [Pa s/m^3]	angle(Z_s) [rad]
225	4.2E9	0.36
450	5.5E9	0.25

The second step in the procedure was the determination of $\lvert 1/Z_r \rvert$, where the entry impedance of the compensator may be found in the transfer matrix [**T**] of the vibration compensators

$$\mathbf{T} = \begin{bmatrix} 1 & 0 \\ 1/Z_\mathrm{r} & 1 \end{bmatrix} \tag{52}$$

(see Kojima and Edge, 1994). As can be seen in Fig. 20, four pressure signals $p_0 \ldots p_3$ were recorded for two different distances x. In case of prototype I, the distances used were x = 140 mm (system A) and x = 280 mm (system B). The distances used for prototype II were x = 140 mm (system A) and x = 360 mm (system B).

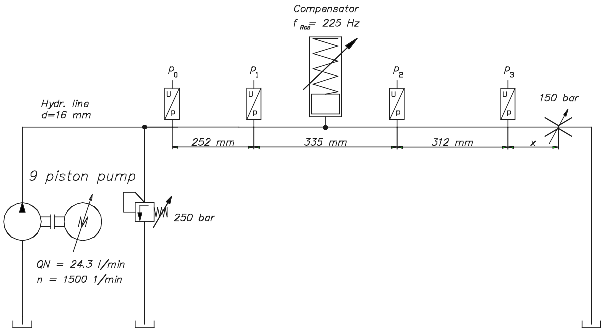


Fig. 20a: *Determination of transfer matrix [**T**]: 450 Hz*

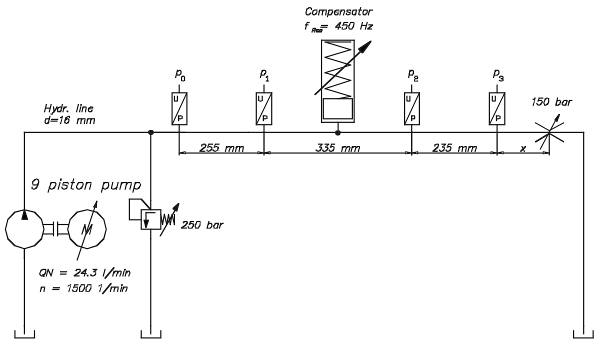


Fig. 20b: *Determination of transfer matrix [**T**]: 225 Hz*

The absolute values of the entry impedances of prototypes I and II, the corresponding transmission losses TL[10] (see Kojima et al, 2000) and the equivalent damping according to Eq. 53 (ref. Eq. 50 for the expression neglecting damping) may be found in Table 10.

$$\left| \frac{1}{Z_\mathrm{r}} \right| = \frac{\omega A_\mathrm{P}^{2}}{\sqrt{\left(-\omega^2 m + c\right)^2 + \left(2\,\omega\,\zeta\,\sqrt{c\,m}\right)^2}} \tag{53}$$

Table 10: Inverse entry impedance, transmission loss and equivalent damping ζ

f [Hz]	$\lvert 1/Z_\mathrm{r} \rvert$ [m^3/Pas]	TL [dB]	ζ [1]
225	1.62E-9	15.2	0.07
450	1.96E-9	19	0.055

Note that $\zeta << 1$ indicates good attenuation performance of vibration compensators.

10) Since the insertion loss IL is only valid for one particular system, the transmission loss TL (for a system with unechoic termination) is indicated. In order to calculate the TL, both the pump source impedance Z_s and the entry impedance Z_r of the compensator are required.

7 Summary

In this paper, a novel vibration compensator for the attenuation of fluid flow pulsations was presented.

After a brief discussion of the sources of fluid flow pulsations in hydraulic systems, conventional devices, such as accumulators, Helmholtz resonators, $\lambda/4$ resonators etc. were presented and their strengths and shortcomings were briefly discussed.

In section 2, a vibration compensator featuring a so-called *hydraulic spring* was introduced. The description of the fundamental working principle and the derivation of the resonance frequency of the device were followed by the investigation of some aspects shifting the natural frequency particularly at high frequencies. Amongst these influences were linear damping, wave propagation and the mass of the hydraulic spring as well as the channel length from the main system to the absorber.

Following those investigations, some aspects regarding the practical implementation of vibration compensators with a hydraulic spring, such as system dynamics, the optimisation of sealing gap geometry and dimensioning were covered. Since semi-active compensators are sometimes of interest, some aspects concerning hydraulic adaptation (modifying the volume V_HS of a hydraulic spring) were presented.

In section 5, an analytical expression for the compactness of branch type vibration compensators was derived and it was demonstrated that so-called MR type resonators build more compact than HR and MHR type resonators.

In section 6, the experimental results of two prototypes were presented. The first prototype featured a resonance frequency of f_Res = 225 Hz with a maximum achievable insertion loss of IL = -22 dB. The second prototype featuring a resonance frequency of f_Res = 450 Hz showed a maximum achievable insertion loss of IL = -29 dB. Also the utilisation of both compensators in a hydraulic circuit at the same time resulted in excellent noise attenuation characteristics of IL = -18 dB and IL = -29 dB at f_Res = 225 Hz and f_Res = 450 Hz respectively. Finally, a comparison of the entry impedances Z_r between the analytical expression in section 5 and the experimental data of prototypes I and II highlights the small damping ratio of the compensator and therefore the great effectiveness of MR type resonators.

Nomenclature

A	Area	[m^2]
A^*	Complex amplitude of a wave running in positive x-direction	
B	Bulk modulus	[Pa]
B'_Tot	Bulk modulus considering flexibility of the enclosure	[Pa]
B^*	Complex amplitude of a wave running in negative x-direction	

c	Spring stiffness	[N/m]
c_{CS}	Stiffness of centring spring	[N/m]
c_{HS}	Stiffness of hydraulic spring	[N/m]
$\mathbf{C}$	Stiffness matrix	[N/m]
d	Damping ratio	[N/(m/s)]
d_P	Diameter of working piston	[m]
D	Diameter	[m]
$Diag$	Diagonal ($\sqrt{l_{Max}{}^2 + D^2}$)	[m]
η	Dynamic viscosity of fluid	[N s/m^2]
f	Frequency	[Hz]
θ	Angle	[rad]
F	Force	[N]
$\mathbf{F}$	Force matrix	[N]
G	Transfer function	
H	Height	[m]
k	Wave number	[1/m]
i	Imaginary unit	[-]
l	Length	[m]
l_{Max}	Maximum length	[m]
λ	Wave length	[m]
IL	Logarithmic measure of noise attenuation	[dB]
m	Mass	[kg]
m_{Osc}	Oscillating mass	[kg]
m_{Tot}	Total mass (mass element and oil mass)	[kg]
μ	Specific mass	[kg/m]
$\mathbf{M}$	Mass matrix	[kg]
n	Pump speed	[1/s]
n_{Pist}	Number of pistons	[-]
p	Pressure	[Pa]
p_{HS}	Pressure in hydraulic spring	[Pa]
Q	Volume flow	[m^3/s]
Q_{Spec}	Specific pump flow	[m^3/rev]
ρ	Specific mass	[kg/m^3]
s_{Gap}	Sealing gap measured at radius	[m]
t	Time	[s]
τ	Shear stress	[N/m^2]
$\mathbf{T}$	Transfer matrix of vibration compensator	
u, v	Velocity	[m/s]
V	Volume	[m^3]
V_{HS}	Volume of hydraulic spring	[m^3]
V_{Tot}	Total volume (mass and hydr. spring)	[m^3]
ω, Ω	Angular frequency	[rad/s]
x	Displacement	[m]
Z	Wave resistance	
Z_r	Entry impedance of vibration absorber	[Pa s/m^3]
Z_s	Source impedance of pump	[Pa s/m^3]
ζ	Dimensionless damping ratio	[-]

Acknowledgements

The authors wish to thank the Department of Foundations of Machine Design, Johannes Kepler University of Linz and Prof. Dr. Rudolf Scheidl in particular for their support.

References

Backé, W. 1994. *Grundlagen der Ölhydraulik* (in German). Lecture notes, RWTH Aachen, Germany.

Beranek, L. and **Vér, I.** 1992. *Noise and Vibration control Engineering*. John Wiley, New York.

Dodson, J., Dowling, D. and **Grosh, K.** 1998. Experimental investigation of quarter wavelength silencers in large-scale hydraulic systems. *Noise control Engineering Journal*, 46(1), pp. 15-22.

Garstenauer, M. Grammer, S. and **Scheidl, R.** 1996. The resonance converter – a novel method for hydraulic fluid power control. *Proceedings of Mechatronics '96*, University of Skövde, Sweden.

Hunt, J. 1979. *Dynamic vibration absorbers*. Mechanical Engineering Publications, London, UK.

Ivantysyn, J. and **Ivantysynova, M.** 2000. *Hydrostatic Pumps and Motors – Principles, Designs, Performance, Modelling, Analysis, Control and Testing*. Academia Books International, New Delhi.

Ingard, U. 1953. On the theory and design of acoustic resonators. *Journal of the Acoustical Society of America*, 25(6).

Ingard, U. 1988. *Fundamentals of waves and oscillations*. Cambridge University Press.

Kojima, E. 1992. A new method for the experimental determination of pump fluid borne noise. *Fifth Bath International Fluid Power Workshop*, University of Bath, UK.

Kojima, E. and **Edge, K.A.** 1994. Experimental determination of hydraulic silencer transfer matrices and assessment of the method for use as a standard test procedure. *Seventh Bath International Fluid Power Workshop*, University of Bath, UK.

Kojima, E. and **Ichiyanagi, T.** 2000. Research on pulsation attenuation characteristics of silencers in practical fluid power systems. *International Journal of Fluid Power*, Vol. 1, No. 2.

Mikota, J. 2001. *Device for damping pressure pulses in a hydraulic system*. International patent application, No.WO02090815.

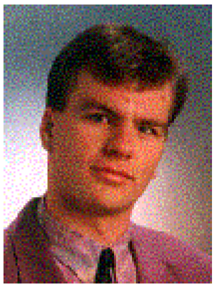
Mikota, J. and **Manhartsgruber, B.** 2001. Transient response dynamics of dynamic vibration compensators for the attenuation of fluid flow pulsations in hydraulic systems. *Power Transmission and Motion control (PTMC '01)*, University of Bath, UK.

Mikota, J. 2002. *Contributions to the development of compact and tuneable vibration compensators for hydraulic systems*. PhD thesis, Johannes Kepler University Linz, Austria.

Reiter, H. 2002. *Simulation, Konstruktion und Vermessung eines neuartigen Schwingungstilgers für die Ölhydraulik (in German)*. Master thesis, Johannes Kepler University Linz, Austria.

Scheidl, R. and **Riha, G.** 1999. Energy efficient switching control by a hydraulic resonance converter. *Power Transmission and Motion control (PTMC '99)*, University of Bath, UK.

Josef Mikota
Born on 24 February 1971 in Steyr (Austria). Degree studies of Mechatronics at the Johannes Kepler University Linz. Master of Science course in Mechatronics and optical Engineering at Loughborough University, UK. Development Engineer at the Research Division of J C Bamford Excavators, Rocester, UK. From 1999 –2002 University assistant and PhD studies at the Department of Foundations of Machine Design (JK University Linz). Currently working as development Engineer in the field of Fire fighting technology for Rosenbauer International AG, Austria.



Herbert Reiter
Born on 7 October 1972 in Kollerschlag (Austria). From 1994-2002 degree studies of Mechatronics at the Johannes Kepler University Linz, Austria. Currently working as Research assistant at the Department of Foundations of Machine Design (JK University Linz).