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Adaptive Helmholtz Resonator in a Hydraulic System

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Abstract**—** An adaptive Helmholtz resonator was designed and adapted to hydraulics. The resonator was controlled by open- and closed-loop controls so that 20 dB attenuation of the peak-to-peak value of the pulsating pressure was maintained. The closed-loop control was noted to be better, albeit it was slower because of its low pressure and temperature variation, which caused variation in the effective bulk modulus of the hydraulic system. Low-pressure hydraulics contains air, which affects the stiffness of the hydraulics, and temperature variation changes the viscosity of the oil. Thus, an open-loop control loses its efficiency if a condition such as temperature or the amount of air changes after calibration. The instability of the low-pressure hydraulic system reduced the operational frequency range of the Helmholtz resonator when compared with the results of an analytical model.

Different dampers for hydraulics are presented. Then analytical models of a hydraulic pipe and a hydraulic pipe with a Helmholtz resonator are presented. The analytical models are based on the wave equation of sound pressure. Finally, control methods and the results of experiments are presented.

Keywords**—** adaptive, damper, hydraulics, pressure, pulsating

I. Introduction

Pressure accumulators, T-pipes, chamber resonators (line filters), multi-degree-of-freedom mass-spring dampers, dampers based on plate elements and Helmholtz resonators can be used in hydraulic systems to damp out pulsating pressure.

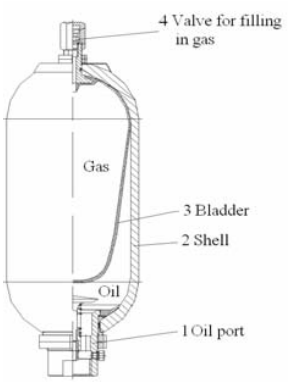

Fig. 1 Cross-section of a pressure accumulator

Accumulators are widely used in hydraulics, but they are not efficient at high frequencies [1]. An accumulator should be placed within 0.3 m of the source of disturbance and the supply line between the accumulator and the main pipe should be short, preferably between 0.05 and 0.1 m [2]. Heat causes problems in continuous use of accumulators because of mechanical movement (stress) of the bladder; see Fig. 1.

A T-pipe (T-filter, band-pass filter) is a side-branch pipe that is closed off at the end; see Fig. 2. The pulsating pressure at a certain frequency is damped if the length of the T-pipe is

$$l_{\text{T-pipe}} = \frac{\lambda}{4} \qquad (1)$$

where λ is the wavelength

$$\lambda = \frac{a}{f} \qquad (2)$$

where a is the pressure wave velocity (sound velocity) in a medium and f is frequency. A long T-pipe is required if a low frequency has to be damped out [1, 2]. For example, if the wave velocity is 1400 $^{m}/_{s}$ and the excitation frequency is 20 Hz, the length of the T-pipe has to be 17.5 m to damp out the pulsating pressure. If the T-pipe is replaced with a short open-ended pipe (orifice branch), a high-pass filter is obtained [3]. However, an orifice branch is unsuitable in hydraulics.

A line chamber resonator (line filter, low-pass filter) is in line with the main pipe so that an enlarged section is inserted into the main pipe; see Fig. 2 [3, 4]. The required length of the chamber resonator can be calculated with equation (1); a long resonator is required for low frequencies. In his dissertation Kiesbauer [4] presented a chamber resonator (*Reihenresonator*) and flow simulations in the frequency domain.

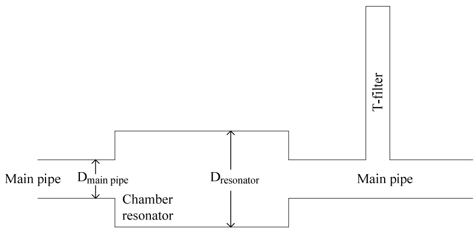

Fig. 2 Example of a line chamber resonator and a T-filter

Mikota [1] presented a multi-degree-of-freedom mass-spring compensator and a compensator based on plate elements; see Figs. 3 and 4, respectively. A multi-degree-of-freedom mass-spring compensator is constructed by adding springs and masses to a cylinder which is added to the hydraulic line so that one of the masses is acted on by pressure P(t). A sealing element is used to prevent oil leakages. The compensator can be designed to damp out several excitation frequencies by adding more masses and springs to the system so that the natural frequencies of the compensator equal the excitation frequencies of the hydraulic system [1].

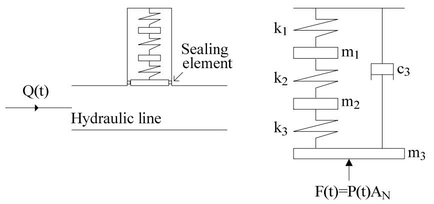

Fig. 3 Principle of a multi-degree-of-freedom mass-spring compensator with three bodies. Q is flow, k_i is a spring, m_i is a mass, c_i is a damper and A_N is piston area. Modified from [1]

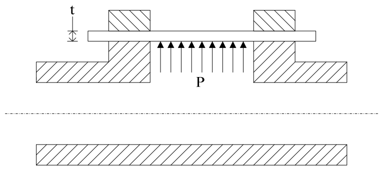

Fig. 4 Principle of a compensator based on plate elements

A compensator based on a plate element is constructed by clamping a plate of homogenous thickness, *t*, to the hydraulic line. The plate has to be tuned so that the natural frequency of the clamped plate equals the excitation frequency. Exceeding the maximum permissible stress of the plate material must be avoided [1]. Mikota [1] noted that the plate has to be large and bulky at high pressures and low frequencies.

Different dampers for hydraulic systems have been developed and presented in the literature, but a Helmholtz resonator, familiar from acoustics, was chosen for this study. Because hydraulics and acoustics are similar in nature, basic knowledge and measurement results from the field are available. The benefits of the Helmholtz resonator, if compared with the aforementioned dampers, are size, temperature stability and adjustability. As presented, the T-filter, in-line chamber resonator and plate-based compensator are longer or bigger if low frequencies are damped. The pressure accumulator and compensator based on plate elements are based on mechanical movement, which generates heat during continuous use. A Helmholtz resonator is also easier to adjust or adapt than is a pressure accumulator or a plate-based compensator.

A Helmholtz resonator in hydraulics has been studied by [1, 2, 4 5, 6]. Viersma [2] studied a Helmholtz resonator in hydraulics in his book, which serves as a basis for all studies in the field. Kiesbauer [4] studied the dynamics of a hydraulic line in his dissertation. He presents different kinds of dampers, such as a line resonator (*Reihenresonator*), a branch resonator (*Abzweigresonator*), a pipe resonator (*Pfeifenresonator*), a Helmholtz resonator, a hydropneumatic damper, multiple resonators (*Mehrkammer-Resonator*) and an active damper. The presented resonators and dampers were also simulated numerically, but the branch resonator and active damper are emphasized in his study, so most of the simulations and experiments are done with them. He also presents variable-volume resonators and examines a variable-volume branch resonator more closely. Two different solutions for making the volume of the branch resonator variable are discontinuous and continuous resonators. The discontinuous resonator was executed so that resonators of different sizes were connected to the main pipe and they were controlled by valves. The continuous resonator was made from a hydraulic cylinder so that both sides of the piston were used by controlling them with a 4/3 valve. Because of the structure of the adjustable cavity, he also studied the effect of seals on the dynamics of an adjustable resonator by means of simulations and experiments. The properties of the oil used and sound velocity in fluid were measured and defined for the experiments. Kiesbauer [4] has presented the control of an adaptive side-branch resonator and has executed simulations and experiments using discontinuous resonators to determine which combination of resonators is needed to obtain maximal attenuation at a certain frequency. The method presented by Kiesbauer [4] could also be regarded as calibration. Kiesbauer [4] did his study at pressures of 15 bar and 80 bar. In the paper written by Mikota [1], a Helmholtz resonator is only mentioned and calculations or experiments are ignored. Ortwig [5] mentioned a Helmholtz resonator as a special type of attenuator, but he did his experiments with other types of attenuators. In his dissertation, Ijäs [6] covered the Helmholtz resonator and conducted experiments, but the resonant frequency of the Helmholtz resonator was constant.

II. ANALYTICAL MODEL

A time domain model can be used to study the interior points of a hydraulic pipe. For example, the well-known four-pole model, which is solved in the frequency domain, can only be used to study the end points of the pipe [2]. The time domain model is solved by using the wave equation of sound pressure, which gives pressure as a function of time at an arbitrary point of the main pipe under the exciting unit volume flow. The chosen method is familiar from acoustics.

Time domain models of the main pipe, the Helmholtz resonator and the main pipe with the Helmholtz resonator are deduced. The dimensions of the pipe and the resonator are assumed to be negligible compared with the wavelength, and sound is assumed to travel as a plane wave.

Fig. 5 presents the principle of the experimental device, which is also the basis of the analytical model, so that the distance between pressure sensors P_1 and P_2 is studied analytically. Thus, both ends of the considered area are assumed to be free, whereupon the effect of the piston and valves can be neglected.

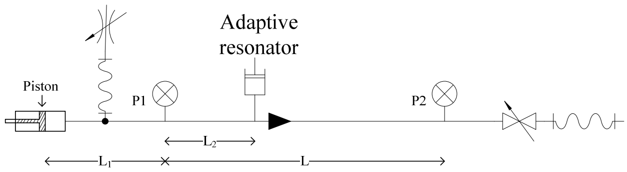

Fig. 5 Principle of the experimental device

The plane wave equation due to external volume flow [7]

$$\psi_0(x,t) = \Psi_0(x)\, e^{i\omega t}, \tag{3}$$

at point x_0 (pressure sensor P_1 in Fig. 5) is

$$\frac{\partial^2 P}{\partial t^2} - a^2 \frac{\partial^2 P}{\partial x^2} = \frac{\rho a^2}{A_P} \cdot \frac{\partial^2 \psi_0}{\partial t^2}\, \delta(x - x_0)\,, \tag{4}$$

where P is pressure, t is time, a is sound velocity, ρ is density, A_P is the cross-sectional area of the main pipe and δ is Dirac's delta function. Remember the assumptions of the free ends. The solution for harmonic vibration is

$$P(x,t) = Ꝑ(x)\, e^{i\omega t}, \tag{5}$$

and the boundary conditions of the free ends are

$$\frac{P(x,t)}{\partial x} = 0\,. \tag{6}$$

Thus, the k:th natural frequency of the main pipe is

$$\omega_k = k\, \frac{\pi a}{L}\,, \tag{7}$$

and its eigenfunction is

$$Ꝑ_k(x) = \sin(k\, \frac{\pi a}{L}\,), \tag{8}$$

where L is the distance between the measurement points; see Fig. 5.

The orthogonality of the eigenfunction (8) can be presented with the help of two eigenfunctions of natural frequencies k and h, respectively.

$$Ꝑ_k''(x) + \frac{\omega_k^2}{a^2}\, Ꝑ_k(x) = 0, \tag{9}$$

$$Ꝑ_h''(x) + \frac{\omega_h^2}{a^2}\, Ꝑ_h(x) = 0. \tag{10}$$

Thus, the orthogonality of the eigenfunction (8) can be presented as

$$\int_0^L Ꝑ_k Ꝑ_h\, dx = \begin{cases} 0, k \neq h \\ \frac{L}{2}, k = h \end{cases}. \tag{11}$$

The solution of equation (4) is given by the series of eigenfunctions as

$$P(x,t) = Ꝑ(x)\, e^{i\omega t} = \sum_{k=1}^{\infty} Ö_k Ꝑ_k \;\; e^{i\omega t}. \tag{12}$$

The unknown coefficient $Ö_k$ is determined by substituting equation (12) for equation (4) and noting the orthogonality of the eigenfunction. Thus, the solution of (12) can be presented as

$$Ꝑ(x) = \sum_{k=1}^{\infty} \frac{\rho a^2 \omega^2 P_k(x) P_k(x_0)}{A_P(\omega^2 - \omega_k^2)\frac{L}{2}} \Psi_0(x) = G(x,x_0)\; \Psi_0, \tag{13}$$

where $G(x,x_0)$ is Green's function, which gives the pressure P at x under the excitation unit volume flow at x_0.

The pressure in the main pipe without a Helmholtz resonator was solved above. Next, the effect of a Helmholtz resonator is added to the model. The fluid in the neck of the resonator is assumed to move as a unit of constant mass without any losses if the length of the neck is short compared to the wavelength. The fluctuating pressure in the cavity is given as

$$P_c = -\frac{\rho a^2 A_n z}{V_c}\,, \tag{14}$$

where A_n is the cross-sectional area of the neck, z is the displacement of fluid in the neck and V_c is the volume of the cavity. The volume flow in the neck can be presented as

$$\psi_n = A_n\, z. \tag{15}$$

The equation of motion of the fluid in the neck is

$$\rho\, A_n\, l_n\, \frac{d^2 z}{dt^2} = A_n\, (P_c - P(x_r,t)), \tag{16}$$

where l_n is the length of the neck of the resonator and $P(x_r,t)$ is the pressure in the main pipe below the neck of the Helmholtz resonator; see Fig. 5. By eliminating z from equations (15) and (16), the following pressure equation is obtained

$$\frac{d^2 P_c}{dt^2} + \frac{A_n a^2}{V_c l_n}\, (P_c - P(x_r,t)) = 0. \tag{17}$$

The resonant frequency of the Helmholtz resonator can be solved if the pressure in the main pipe below the neck, $P(x_r,t)$, is assumed to be zero. Thus, the resonant frequency of the Helmholtz resonator is

$$\omega_{HR} = a\sqrt{\frac{A_n}{l_n V_c}} \,. \tag{18}$$

The harmonic volume flow from the neck to the main pipe is

$$\psi_n = A_n\, z\, e^{i\omega t} = \Psi_n\, e^{i\omega t}, \tag{19}$$

where Ψ_n is the amplitude of the harmonic volume flow. If the pressure equation (17) is solved for

$$P_c = \not{P}_c\, e^{i\omega t}, \tag{20}$$

and for

$$P(x_r,t) = \not{P}(x_r)\, e^{i\omega t}, \tag{21}$$

and equations (18) and (19) are noted, the relationship between the pressure and the volume flow is obtained

$$\not{P}(x_r) = \frac{l_n \rho}{A_n}\,(\omega^2 - \omega_{HR}^{\,2})\,\Psi_n = \not{X}\,\Psi_n, \tag{22}$$

where x_r is the position of the resonator.

The differential equation of flow from the resonator is solved by substituting equation (14) for equation (16) and noting equation (19)

$$\frac{d^2\Psi_n}{dt^2} + \omega_{HR}^2 \Psi_n = -\frac{A_n}{\rho l_n} P(x_r, t)\,. \tag{23}$$

The wave equation of the main pipe with volume flows from two sources (primary excitation at x_0 and the resonator at x_r) can be presented as

$$\frac{\partial^2 P}{\partial t^2} + a^2 \frac{\partial^2 P}{\partial x^2} = \frac{\rho a^2}{A_P}\Big[\frac{d^2\psi_0}{dt^2}\,\delta(x - x_r) + \frac{d^2\psi_n}{dt^2}\,\delta(x - x_r)\Big]\,. \tag{24}$$

Equations (23) and (24) are simultaneous differential equations that determine the flow from the Helmholtz resonator and the pressure in the main pipe. The solution of equation (24) is (note equation (13))

$$\not{P}(x) = G(x,x_0)\,\Psi_0 + G(x,x_r)\,\Psi_n, \tag{25}$$

and the pressure at $x = x_r$ is

$$\not{P}(x_n) = G(x_r,x_0)\,\Psi_0 + G(x_r, x_r)\,\Psi_n. \tag{26}$$

The flow is solved from equations (22) and (26) by eliminating $\not{P}$

$$\Psi_n = \frac{G(x_r, x_0)}{\not{X} - G(x_r, x_r)}\,\Psi_0\,. \tag{27}$$

The pressure at an arbitrary point, x, in the main pipe to which the Helmholtz resonator is added can be calculated using

$$\not{P}(x) = \frac{\not{X}G(x,x_0) - G(x_n,x_n)G(x,x_0) + G(x,x_n)G(x_n,x_0)}{\not{X} - G(x_n,x_n)}\,\Psi_0\,, \tag{28}$$

which is obtained by substituting equation (27) for equation (25).

First it should be noted that the modelled results do not agree with hydraulics if the pressure responses of the whole pipe are considered, as presented in Figs. 6 and 7. Fig. 6 presents the pressure in the main pipe. The result is obtained by solving the continuity equation (29) and the equation of motion (30) using the method of characteristics. The continuity equation is

$$\frac{1}{B}\frac{\partial P}{\partial t} = -\frac{\partial u}{\partial x}\,, \tag{29}$$

where B is the bulk modulus and u is flow velocity. The equation of motion is

$$\rho\frac{\partial u}{\partial t} = -\frac{\partial P}{\partial x} - \frac{f_D}{4}\frac{\rho u}{2r}\lvert u\rvert\,, \tag{30}$$

where ρ is density, f_D is Darcy's friction factor and r is radius.

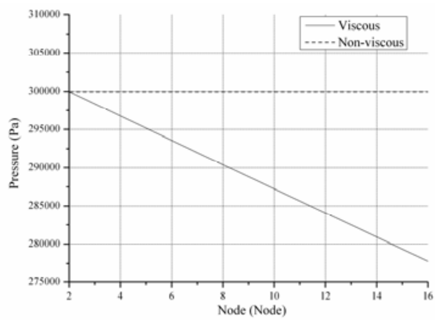

Fig. 6 Iterated pressure in the main pipe without the Helmholtz resonator. The length of the node is $^{L}/_{N-1}$ and the first and last nodes are ignored from the results because those nodes are affected by the boundary conditions

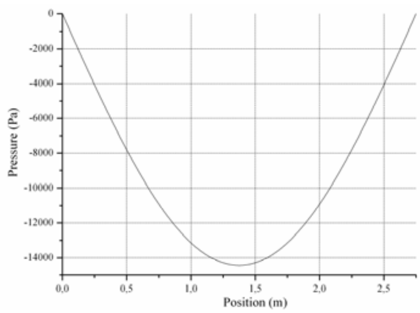

Fig. 7 Sound pressure in the main pipe without the Helmholtz resonator

Fig. 6 includes two cases, viscous and non-viscous. The response with viscosity agrees better with the measurements. However, both results are presented because the effect of viscosity is ignored in the sound pressure calculations, too. Although the method of characteristics describes the hydraulic system better, it is ignored because the results must be iterated. Iteration takes time and convergence can not be guaranteed every time. Thus, control systems can not be modelled reliably by using the method of characteristics. Also, sound pressure should be iterated if the effect of viscosity were to be included in the model. For example, Matsuhisa *et al.* [7] used the Adams method to numerically solve the damped sound pressure.

Although the sound pressure in the whole main pipe does not agree with the hydraulic model, the sound pressure equation can be used to model the hydraulic system if a certain point in the main pipe is considered, as presented in Figs. 8 and 9. Fig. 8 presents the hydraulic pressure at node 7, which is point x = 1.03 m in the experimental device. Fig. 9 presents sound pressure in the main pipe of the experimental device at the 1.03 m point. As can be noted, the results are similar if a certain point is considered at a time. Thus, the experimental device can be modelled by using the presented sound pressure model if the effect of the resonator is considered at a certain point of the main pipe at a time.

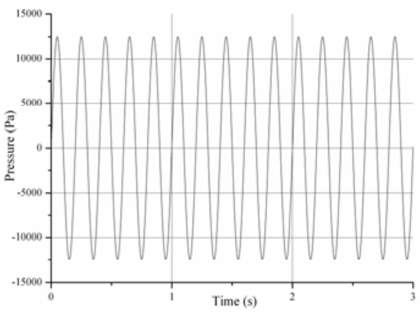

Fig. 8 Hydraulic pressure in the main pipe at the 1.03 m point (node 7)

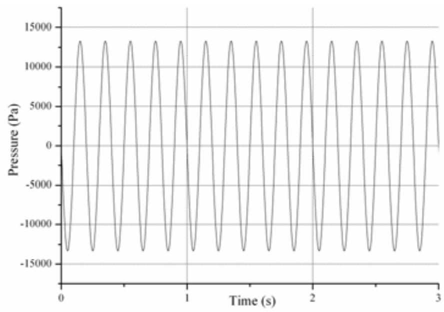

Fig. 9 Sound pressure in the main pipe at the 1.03 m point

Figs. 10 and 11 present the pressure at the 1.03 m position in the main pipe without a Helmholtz resonator and with a Helmholtz resonator whose resonant frequency is 23.4 Hz, respectively. Four different excitation frequencies are presented in each case: 5, 15, 23.4 and 35 Hz. The transition between frequencies is presented strictly stepwise to pack the results in the same plot. The transition would be continuous in the real world. As Figs. 10 and 11 indicate, the Helmholtz resonator lowers the amplitude of vibration at every frequency, but the best efficiency is reached at its resonant frequency.

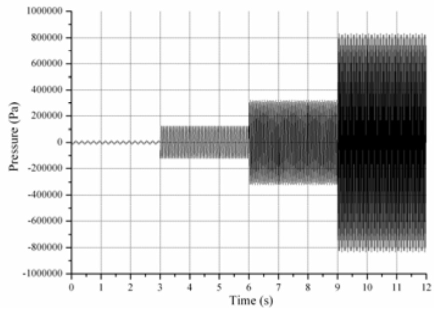

Fig. 10 Pressure at the 1.03 m point in the main pipe

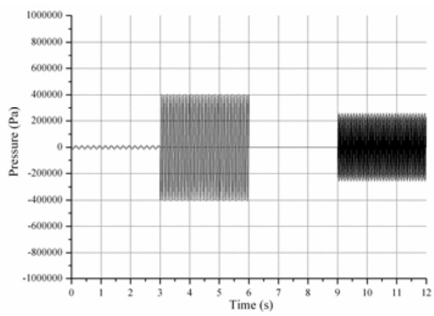

Fig. 11 Pressure at the 1.03 m point in the main pipe which also has a Helmholtz resonator at the same point

III. DESIGN OF THE ADAPTIVE HELMHOLTZ RESONATOR

Fig. 12 presents the adaptive Helmholtz resonator and the hydraulic cylinder that moves the piston inside the adaptive Helmholtz resonator, and Fig. 13 presents the principle of the adaptive Helmholtz resonator. The Helmholtz resonator was made of a hydraulic cylinder whose inner diameter was 100 mm. The resonant frequency of the adaptive Helmholtz resonator was controlled by a piston inside the resonator, so that the length of the cavity could be changed continuously between 0.043 and 0.243 m.

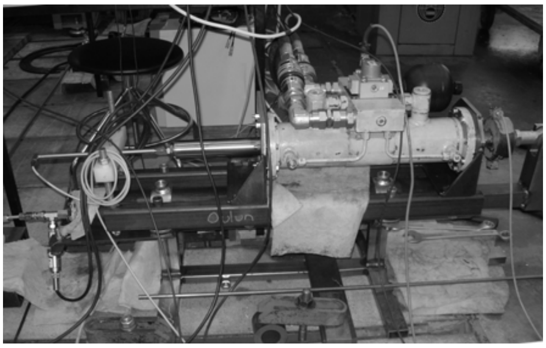

Fig. 12 Helmholtz resonator (below) and the hydraulic cylinder (white, above) used to adjust the piston position inside the resonator

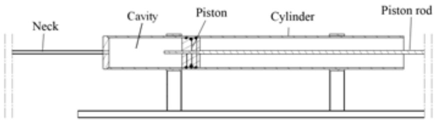

Fig. 13 Principle of an adaptive Helmholtz resonator for hydraulics

As presented in equation (18), the resonant frequency of the Helmholtz resonator is defined by sound velocity and the dimensions of the neck and cavity. All the mechanical dimensions can be measured reliably. Despite this, sound velocity in a fluid is a sensitive parameter, which is affected by, among others things, pressure and temperature, so that its value should be measured instead of using the value presented in theory. A low-pressure hydraulic system, in particular, may contain air (dissolved or entrained), in which case the effective bulk modulus, and thereby sound velocity, is lower than that presented in theory, because the value presented in theory is often defined in a pure fluid (no air or other components). Thus, sound velocity in the main pipe of the experimental device was measured using the impact test, and the results are presented in a previous paper [8].

The resonant frequencies of the adaptive Helmholtz resonator as a function of piston position were measured and the results are presented in paper [9]. As presented in paper [9], the measured frequencies do not agree with the analytically calculated results. Because the other factors of equation (18) are noted to be reliable, the measured sound velocity in the main pipe must differ from the sound velocity of the whole system. This deviation is caused by air content, because the total volume and thereby the amount of air was increased after the Helmholtz resonator was added to the system. The low pressure and the geometry of the experimental device, especially the geometry of the resonator, enable hydraulics to contain air bubbles or even air pockets inside the system. Because air is softer than oil, the effective stiffness of the system decreases as the amount of air increases. The amount of air in the system was estimated by calculating the effective bulk modulus of the system

$$B_{eff} = \rho \frac{4\pi^2 f^2 l_n V_c}{A_n}, \tag{31}$$

where f is the measured resonant frequency and V_C is the volume of the cavity. After the effective bulk modulus of the system was calculated, the amount of air could be estimated from equation

$$\frac{1}{B_{eff}} = \frac{1}{B_{fluid}} + \frac{V_c}{V_{tot}}\frac{1}{B_c} + \frac{V_p}{V_{tot}}\frac{1}{B_p} + \frac{V_n}{V_{tot}}\frac{1}{B_n} + \frac{V_{air}}{V_{tot}}\frac{1}{B_{air}} \tag{32}$$

where B_{fluid} is the bulk modulus of the fluid (defined by the impact test [6]), V_{tot} is the total volume of the system, V_p is the volume of the main pipe, B_p is the bulk modulus of the main pipe, V_n is the volume of the neck, B_n is the bulk modulus of the neck, V_{air} is the volume of air and B_{air} is the bulk modulus of air (1.4*P).

Up to 1 % of the total volume of the hydraulics of the experimental device was noted to be air. The amount of air varied with pressure and the volume of the cavity. Thus, it was estimated that the effective bulk modulus of the hydraulics of the experimental device was 140 MPa if the pressure was 0.3 MPa. The estimated value was used in modelling, as presented in Figs. 10 and 11.

IV. CONTROL

The purpose of the study was to develop an adaptive dynamic vibration absorber for a hydraulic system so that the harmonic pulsating pressure can be damped out even though its frequency varies. A Helmholtz resonator, which changes its properties automatically so that its resonant frequency always corresponds with the excitation frequency, was chosen as an absorber. The length of the cavity of the Helmholtz resonator was changed so that the resonant frequency of the resonator varied. The length of the cavity was controlled with both open- and closed-loop controls so that the outlet pressure of the main pipe remained stable even though the excitation frequency of the inlet pressure varied. The principles of the control methods are presented in Fig. 14.

An open-loop control was the first control method. A pressure sensor, P_1 in Fig. 5, was used to identify the disturbing frequency. The identified disturbing frequency was compared with the frequencies presented in a control list compiled beforehand so that the optimal piston position was found. The control list included frequency – piston position pairs and the control programme checked from there the correct piston position by comparing the identified frequencies with the frequencies presented in the list. After the piston was in the correct position, maximum attenuation of the pulsating pressure in pressure sensor P_2 was reached.

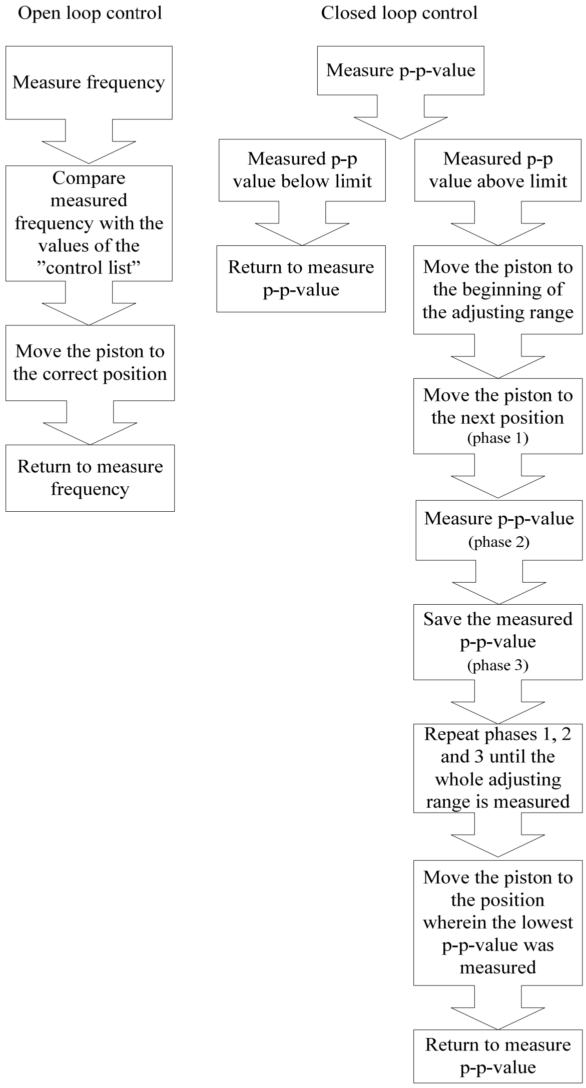

Fig. 14 Block diagrams of the open-loop and closed-loop controls, respectively

The closed-loop control moved the piston 0.015 m (note the safety margins (0.005 m) at both ends of the cavity) or 0.020 m at a time and checked the corresponding p-p value of the pressure from pressure sensor P_2 between movements; see Fig. 5. This procedure was repeated until the whole adjusting range was checked. Then the piston returned to the position wherein the smallest p-p value was observed. This position was kept until the p-p value of the pulsating pressure exceeded 17.5 kPa.

Fig. 15 presents the efficiency of the adaptive Helmholtz resonator in the time domain. Two measurements are scaled to the same figure. The first one is measured without a Helmholtz resonator and the second one with a Helmholtz resonator at pressure sensor P_2; see Fig. 5. The excitation frequency is 31.7 Hz in both cases.

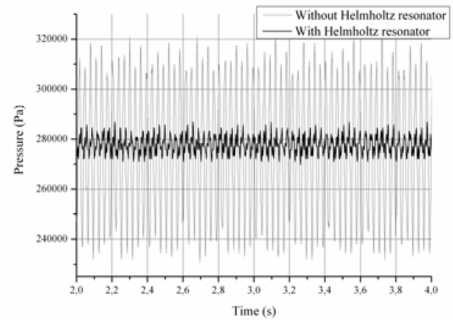

Fig. 15 Measured pressures at pressure sensor P_2

The adjusting frequency range was noted to be narrower than expected (24 Hz) and attenuation of 20 dB was reached only between the frequencies of 34.2 and 38 Hz during the open loop experiments. The difference between what was expected and the experiments was caused by temperature variation and dissolution of air. The temperature of the hydraulic oil varied between 20 and 22˚C when the values of the control list were defined. However, the temperature of the hydraulic oil rose from 20 to 24˚C during the open loop experiments. The deviation between theory and experiments increased more when the system warmed up from the planned temperature, so that at a temperature of 24˚C only 17 dB attenuation of the p-p value of pressure was reached. The rise in temperature also changes the bulk modulus of the fluid, so the resonant frequency of the Helmholtz resonator also varies and the viscosity decreases, which decreases inertial damping in which case the attenuation range narrows. Also, the amount of air inside the experimental device may vary during long-term experiments, so that the effective bulk modulus of hydraulics varies a little bit all the time. Thus, the open-loop control is very case-specific and operates reliably only in a stabile system. In addition, an accurate open-loop control cannot be based on calculations; calibration measurements are needed before start-up.

The measured maximum attenuations varied from 19 to 20 dB at pressure sensor P_2 when the closed-loop control was used. Once again, the adjusting range was noted to vary as the experiments continued and oil temperature varied. However, the closed-loop control maintained at least -19 dB attenuation of the pulsating pressure if the excitation frequency was inside the adjusting range, even though the temperature varied between 21 and 29˚C during the experiments. Maximum attenuation was achieved between the frequencies of 35.6 and 46.4 Hz.

The reason for the variations in oil temperature was the hydraulic pump unit, whose small oil tank was positioned around the pump and the electric motor, so that they warmed up the oil. The loading of the oil was also increased by using a strict pressure control circuit. In addition, variations in the measured responses were noted to be hysteretic in nature, so that the original starting point was not reached even though the system was cooled down. The responses were returned to the initial stage only by changing the oil in the experimental device. The oil properties started to vary again after the experiments were continued, despite the oil change. This phenomenon was caused by air bubbles and pockets that accumulated in certain points of the experimental device every time despite deaeration through the bleeding screws. Continuous usage of the experimental device (flow) slowly dissolved the entrained air from the device so that the effective stiffness of the experimental device increased until the initial stage was returned by changing the oil.

In hydraulics, the effect of the volume change of the resonator (piston movement) has to be taken into account the whole time. Sharp and fast changes might break the system (for example its pressure sensors). Anyway, a change in volume causes pressure variation in the system, and the measured p-p values of pressure fluctuate during piston movement and stability is not reached until the piston has been stopped.

V. Conclusions

The main objective of the study was reached and the p-p value of the pulsating pressure was attenuated 20 dB by the adaptive Helmholtz resonator when compared with the measured p-p values without the Helmholtz resonator. The measured attenuation, 20 dB, is similar to the results of other studies of ADVAs or Helmholtz resonators. For example, de Bedout *et al*. [10] reduced the pressure level 30 dB in an acoustic system by using a controlled tunable resonator, Singh *et al.* [11] measured 18 dB attenuation in net acoustic power transmission in a duct downstream from a Helmholtz resonator and Franchek *et al.* [12] measured 24 dB(A) maximum reduction in vibration in four-degrees-of-freedom test equipment with an adaptive absorber, and 17.2 dB(A) reduction was measured during the entire test period. Their damper consisted of a mass and a helical spring, whose number of active coils was changeable.

The presented open-loop control is fast and reliable as long as the system or environment remains constant so that properties like pressure or temperature do not vary from the state of calibration. Variations in the system or environment can be taken into account by making different control lists for different conditions, but it should be remembered that the DVA can act as an amplifier if it is untuned. The chosen open-loop control would be workable in mill conditions, wherein the properties of the system and surroundings are stabile during operation.

The presented closed-loop control is slow but reliable as long as the disturbing frequency is inside the adjusting range of the Helmholtz resonator. The closed-loop control independently finds, without any controlling list, the optimal piston position to attenuate the p-p value of pulsating pressure. The presented control also notices variations in the system or surroundings so that re-adjusting can be done. The slowness of the presented control could be avoided by helping the control, for example with phase-difference-based directional control, so that the piston could be moved in an optimal direction without moving through the whole adjusting range. However, as noted in the experiments, it would not be beneficial to the performance of the hydraulic system if the properties of the resonator were to change continuously, because every time the volume of the cavity is varied, differences in the pressure level are caused. Thus, changes in the system would be difficult to identify rapidly.

The peak-to-peak value was chosen in this study to represent the condition of the system and surroundings because it is affected by several factors of the system and surroundings. Thus, this kind of control is suitable for various applications, even in outdoor locations wherein conditions might vary. However, the system properties have to be inside the adjusting range the whole time.

The usability of the Helmholtz resonator could be increased if an electrically controlled valve could be installed between the main pipe and the Helmholtz resonator. This feature would widen the range of applications where Helmholtz resonators could be used. The possibility of disconnecting or connecting a resonator to a hydraulic system would be a useful function because the addition of a resonator to a hydraulic system increases the volume of the hydraulic system. However, increased volume means increased mass, which is not always a desirable feature. Thus, easy connectivity of the resonator would be a good feature, especially if emptying of the Helmholtz resonator could be implemented.

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