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CONTROL OF AN ADJUSTABLE HELMHOLTZ RESONATOR IN A LOW-PRESSURE HYDRAULIC SYSTEM

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Abstract

The theory of controlling adjustable tuned vibration absorbers (incl. the adjustable Helmholtz resonator) is reviewed. The theory review is completed with analytical models containing a two-degrees-of-freedom spring-mass model in which the spring constant between the primary system and the vibration absorber is controlled. The main focus of this paper is on the Helmholtz resonator in a hydraulic system, so all parameters are adapted to hydraulics. Two control methods are presented, open loop and closed loop. Both methods are modelled analytically and the models are experimentally verified by means of hydraulic test equipment consisting of a main pipe and an adjustable Helmholtz resonator. The open-loop control identifies the disturbance frequency and then adjusts the volume of the adjustable resonator accordingly by using a previously produced control list that contains information on frequency and corresponding cavity volume (piston position). The closed-loop control adjusts in order of different volumes of the resonator while continuously measuring the response of the system, and after this identifying phase the resonator is adjusted to the volume that produced the most favourable response. The peak-to-peak values in the main pipe were measured and the 20 dB attenuation level was measured when the resonator was used.

Keywords: control, helmholtz resonator, hydraulics

1 Introduction

As in mechanics, also hydraulic systems contain harmful vibrations, which are observed as a variation in pressure. In the old days problems arose from hydraulic machine units, but nowadays they are rather steady because of better manufacturing, sophisticated valves, electronic control of the hydraulic machine units and so on. At the same time the precision requirements of the machines have risen. This has led to steadier machines; especially the mechanics of the machines are nowadays so well known that the bases and bodies of the machines are normally steady. However, many machines include hydraulics, and the harmful vibration of the machines can be transferred to the hydraulic system and move along it far away from the source, causing damage and noise in many places. Thus, this paper presents an adjustable Helmholtz resonator in a hydraulic system that attenuates harmful vibrations at different frequencies. The Helmholtz resonator is like a tuned vibration absorber in mechanics, and it has been noted that tuned vibration absorbers are effective dampers at a certain frequency. The problem is that the damped frequency band is very narrow, so that even small variations in the surroundings can cancel out the effectiveness of the tuned vibration absorber. This has limited the usability of tuned vibration absorbers. Fortunately, application areas can be increased noticeably if the physical properties (stiffness or mass) of the tuned vibration absorber can be adjusted. Furthermore, the benefits of an adjustable tuned vibration absorber increase if it can be controlled automatically. Naturally, these possibilities have led research on the subject strongly forward, as noted from the references. It is easy to see that most of the previous studies have been carried out in mechanics and acoustics, and the field of hydraulics has not been taken into consideration until now. Fortunately, same basic control properties are valid in acoustics, mechanics and hydraulics. The use of Helmholz resonators as dampers is discussed in the literature survey.

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2 Literature Survey

Matsuhisa, Ren and Sato (1992) presented a Helmholtz resonator whose cavity volume is automatically controlled so that the system stays in the anti-resonance state. The control system was based on the fact that the phase difference between the pressure in the duct and in the cavity changes 180 degrees at anti-resonance. The controlling was carried out in two steps. First an open-loop control was used where the disturbance frequency was measured and the optimal volume of the cavity was checked from pre-measured data. After rough adjustment, the volume of the cavity was adjusted so that the phase difference was 90 degrees. They verified their system both theoretically and experimentally and noted also that the Helmholtz resonator is an efficient damper – sound pressure was decreased up to 30 dB around the resonance.

de Bedout et al. (1997) presented a tunable, variable-volume Helmholtz resonator controlled by a robust, simple control algorithm to achieve maximum performance. The variation in the cavity volume was realized by rotating an internal wall inside the cavity with respect to an internal fixed wall. An open-loop algorithm was used to ensure convergence of the closed-loop gradient descent algorithm. The correct cavity volume was reached in the open-loop control by comparing noise with theoretically calculated values. Precise tuning was realized by applying the gradient base feedback tuning control law. The algorithm started by making 1 Hz incremental changes in the resonant frequency, and when the sign of the gradient changed the tuning direction was reversed. Every time the slope changed from negative to positive, the increment quantity was decreased by 0.3 Hz. The minimum was passed six times, and the gradient-based tuning was completed and control chattering of the resonator was avoided. Experiments were carried out and up to a 30 dB decrease in the sound pressure level was reached with the tunable Helmholtz resonator.

Kostek and Franchek (2000) used a hybrid noise control system to damp broadband noise. The hybrid system contained active feedback noise control and adaptive-passive noise control (a tunable Helmholtz resonator whose cavity volume is adjustable). A robust tuning algorithm was used to tune the resonator. The algorithm looked through the whole adjustment range of the resonator and stored the values of the microphone's amplitudes. After passing through the adjustment range, the resonator was set to the position where the amplitude was at a minimum. Retuning was executed if the sound pressure level increased, so that optimal tuning was maintained. They noted that the adaptive-passive system minimized the overall band-pass filtered signal and the active noise control results were in line with the results of a sensitivity chart.

Estève and Johnson (2004, 2005) and Johnson and Estève (2002) studied adaptive Helmholtz resonators to control broadband noise inside rocket payload fairing. In their first solution the resonant frequency of the resonator was controlled by varying the length of the neck, and in the second solution the opening of the Helmholtz resonator was varied by using an iris diaphragm. They investigated strategies for controlling the Helmholtz resonators using purely local information. Local strategies made it possible to use a resonator independently (Johnson and Estève, 2002). To control an independent resonator, each of the resonators was equipped with two microphones so that each resonator and its microphones constituted an autonomous damping device (Estève and Johnson, 2004). They compared the global cost function with four local cost functions (HR internal pressure, HR external pressure, pressure difference from an absorber to the base and the cross product between the absorber and the base), which were used to update the stiffness of the absorbers (Johnson and Estève, 2002). The cross product method, which multiplied the velocity of the absorber mass and the velocity of the absorber attachment point to determine the direction of tuning, was noted to be the best tuning criterion. The cross product method was observed to be useful also with broadband noise. Later, Estève and Johnson (2005) presented a tuning law called the dot-product method, which was based on phase information between the velocity of the absorber mass and the velocity of the host structure. They carried out numerical simulations and experiments in the time and frequency domains. In the experiments they used an adjustable opening area (Estève and Johnson 2005) and installed Helmholtz resonators inside a cylinder. They noted that the dot-product method can be implemented using analogue circuitry, which makes the controller cheaper, lighter and more straightforward. The dot-product method can tune resonators to a near-optimal solution over a frequency band that includes multiple resonances. They reached noise attenuation of 6.5 dB in the 95-115 Hz band.

Singh et al. (2006) presented a cost function that enabled tuning of an adjustable Helmholtz resonator without any in-duct pressure sensors. The cost function was based on the phase difference between the pressure at the closed end of the cavity of the resonator and the pressure at the neck. Using the cost function, they controlled the volume of the cavity and reached a maximum decrement of in-duct acoustic power transmission by minimizing their cost function.

3 Test Equipment

The Helmholtz resonator we used is presented in Fig. 1 and 2. A shaker moves a small piston inside the main pipe (hydraulic pipe, $Ø_{in}$ 8 mm and $Ø_{out}$ 11 mm), which causes harmonic disturbance pressure variations in the system. The disturbances are damped out by the adjustable Helmholtz resonator, whose volume is adjusted by an inner movable piston. This solution probably allows the widest tuning range, as the natural frequency varies as the square root of the volume (Estève and Johnson, 2004). Other reasons for choosing this damping solution were a simple and reliable structure and straightforward fabrication. The Helmholtz resonator consists of a hydraulic cylinder pipe whose length is 0.998 m ($Ø_{in}$ 100 mm and $Ø_{out}$ 110 mm). However, the maximal adjusting range is 200 mm, from a position of 0.043 m to a position of 0.243 m, measured from the bottom of the resonator. The movement of the piston is executed by another hydraulic cylinder controlled by an external control unit. This cylinder is in another hydraulic circuit so that oil is pumped by a big hydraulic machine unit.

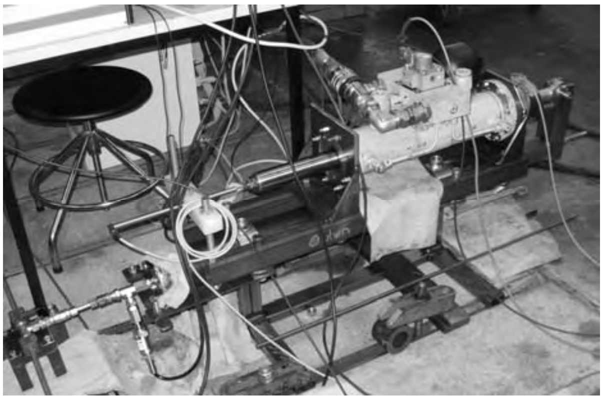

Fig. 1: *Helmholtz resonator used in the experiments. On the top of the resonator is a hydraulic cylinder which moves the piston inside the resonator (adjusting the volume of the cavity)*

The maximum pressure used in the experiments was 3 bar. The temperature of the surroundings varied between 18˚C and 20˚C during the measurements, and the temperature of the oil varied between 20˚C and 29˚C. The variability of the fluid temperature was noted also in the measured results. The measurement system consists of a Kyowa PG-10KU pressure transducer, two Kyowa PG-20KU pressure transducers, a Kyowa Strain Amplifier DPM-6H (for the Kyowa pressure transducers), a National Instruments SCB-68 MIO-16E Series DAQ card, an Intel Pentium 4 CPU 2.4 GHz computer with Microsoft Windows XP, DasyLab v.8.00.04 and DasyLab v.9.00.02 measurement software and a Measurement&Automation Explorer v.4.5.0F0 application, a Thermocouple PT100 TD-TV/PT1A temperature sensor, a Digitron thermometer (max 850˚C), a Dana Exact generator, an 8500+ Instron control unit for the hydraulic machine unit and a Kemo filter between the DAQ card output and the Instron control unit. The measurement frequency was 500 Hz and the block size was either 1024 or 32 samples, depending on the control method. Open-loop control uses a FFT, which requires a bigger block size for accuracy.

The fluid used in the test equipment was commercial mineral oil-based hydraulic oil (Teboil Larita Oil 10) whose bulk modulus is estimated to be 1.67 GPa. The estimation of the bulk modulus is based on the results given in a previous article (Kela and Vähäoja, 2009). Naturally, this estimation is inaccurate because the oil has been changed for the measurements of this research. However, the open-loop control requires calibration measurements – accurate open-loop control cannot be based on calculations. Thus, the inaccurate bulk modulus estimation affects only analytical modelling; whose accuracy is not the main focus of this article.

During three months of experiments, the measured responses were found to continuously vary slightly even though environmental conditions were almost constant the whole time (inside the laboratory). Small variations in the results were expected because the temperature of the oil varied between 20˚C and 30˚C during the measurements, but the variations were larger than expected. The reason for the oil temperature variation was the small, old hydraulic pump unit we used, whose oil tank was small and located around the pump and an electric motor, so that they warmed up the oil. The loading of the oil was also increased by using a strict pressure control circuit. Against expectations, the variations were noted to be hysteretic in nature, so that after a while the original starting point was not reached. The responses were returned to the starting point only by changing the oil in the test equipment, and they started to vary again after the oil change. The viscosity and density of the new oil and the used oil were measured, and it was noted that the density of the new oil varied from 839 g/l to 829 g/l if the temperature was raised from 20˚C to 30˚C. The corresponding densities of the used oil were 833 g/l and 825 g/l. The viscosity of the new oil varied from 17 cP to 8 cP when the temperature was raised from 20˚C to 45˚C.

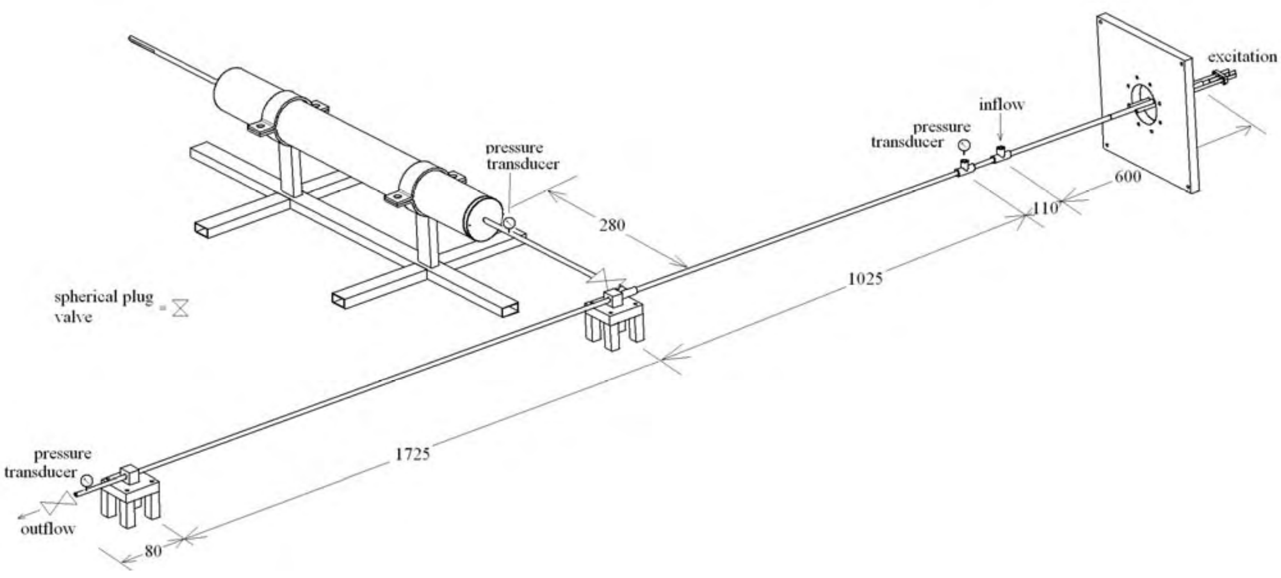

Fig. 2: *Test equipment; hoses are not included. Dimensions are in millimeters*

The corresponding values for the used oil were 18 cP and 8 cP. As one can observe, the measured variations are within the tolerance of common practice. However, the previous article (Kela and Vähäoja, 2009) presented the effect of oil density on its bulk modulus, and that article can be used as an aid when predicting changes in the system due to the oil density variations. The effect of viscosity on the resonant frequency of the Helmholtz resonator is less studied in the literature, so we decided to omit its influence. One explanation for the hysteretic nature of the response might be air bubbles, or even small air pockets in the oil, which dissolve gradually during the experiments. Therefore, the effective bulk modulus (stiffness) of the system would vary and cause recognized hysteresis. Because of the low pressure and shape of the structures, dissolved air and entrained air (air bubbles) possibly exist in the system in spite of deaeration, which was done through the bleeding screws of the pressure sensors and the bleeding screw of the resonator. The relative amount of air could be decreased by increasing the pressure of the system if all the components of the system (e.g. the pressure sensors) would withstand it. In analytical modelling the amount of air is taken into consideration, so that the main pipe and hoses are expected to be airless (the value of the fluid bulk modulus, 1.67 GPa, is used) and in the Helmholtz resonator the volume of air is expected to be 0.2 % of the total volume. The results of the modelling are comparable with the measurement results, as can be noted.

4 Analytical Modelling

In his book, Viersma (1980) presented four-pole equations to simulate the dynamics of hydraulic systems. Those equations are omitted from this paper because they are designed to be used in the frequency domain, and controlling will be done here in the time domain. Thus, a spring-mass model, like in mechanics, is used here, but so that its parameters are adapted to be suitable for hydraulics. The spring-mass model clearly visualizes the control methods and is primarily solved in the time domain, but is easily transformed to the frequency domain. Fig. 3 presents a primary system (main pipe) with an undamped vibration absorber (Helmholtz resonator) that is the basis of this article's analytical modelling.

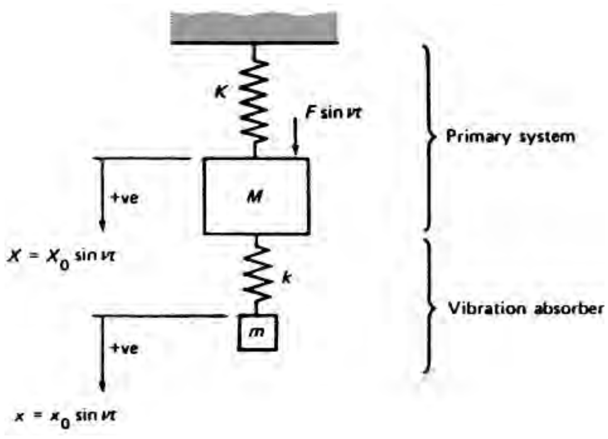

Fig. 3: *Primary system with an undamped tuned vibration absorber (Beards, 1995)*

The equation of motion of the primary system is

$$X(t) = \frac{F(k - m\omega^2)}{(k - m\omega^2)(K + k - M\omega^2) - k^2} \sin(\omega t), \qquad (1)$$

and the equation of motion of the vibration absorber is

$$x(t) = \frac{Fk}{(k - m\omega^2)(K + k - M\omega^2) - k^2} \sin(\omega t), \qquad (2)$$

where F is the force directed to the primary system, K and M are the effective stiffness and mass of the primary system, k and m are the effective stiffness and mass of the vibration absorber, ω is the angular velocity and t is time (Beards, 1995). In the hydraulic test equipment we used, X is the pressure of the main pipe and x is the pressure of the resonator, K is the stiffness of the hydraulic oil in the main pipe and hoses, M is the mass of the hydraulic oil in the main pipe and hoses, k is the stiffness of the hydraulic oil in the resonator's cavity and m is the mass of the hydraulic oil in the neck of the resonator.

Table 1: Dimensions of the test equipment used

	Main pipe	Hoses	Neck	Cavity
Diameter [m]	0.008	0.00635	0.006	0.100
Length [m]	2.75	12.9	0.28	varies
Area [m²]	$5.03 \cdot 10^{-5}$	$3.17 \cdot 10^{-5}$	$2.83 \cdot 10^{-5}$	$7.85 \cdot 10^{-3}$
Volume [m³]	$1.38 \cdot 10^{-4}$	$4.09 \cdot 10^{-4}$	$7.92 \cdot 10^{-6}$	varies

The bulk modulus of the hydraulic oil is estimated to be 1.67 GPa (Kela and Vähäoja 2009). The estimation of the stiffness of the main pipe and hoses is based on the assumption that the oil in the main pipe moves as a rigid part and the oil in the hoses is a spring. Thus, the stiffness of the main pipe is

$$K_{\text{main pipe}} = \frac{B_{\text{fluid}} A_{\text{main pipe}}^{2}}{A_{\text{hose}} L_{\text{hose}}} \qquad (3)$$

where B_{fluid} is the bulk modulus of the fluid (airless), $A_{\text{main pipe}}$ is the cross-sectional area of the main pipe, A_{hose} is the cross-sectional area of the hose, and L_{hose} is the length of the hoses. Thus, the stiffness of the main pipe is 10328 N/m. The mass of the oil in the main pipe and hoses is 0.48 kg. Thus, the natural frequency of the main pipe would be 23 Hz, which agrees with the measured result presented in Fig. 7.

The air content in a low-pressure hydraulic system is larger than in high-pressure systems (> 50 bar). The pressure used in the test equipment is 3 bar. Also the resonator (cylinder) is difficult to deaereate, despite the bleeding screw on the top corner, whereupon the possibility of entrained air being present in the test equipment increases when the volume of the system is increased, for example by adding the Helmholtz resonator to the main pipe. Thus, it is supposed that the test equipment with the Helmholtz resonator contains air whose effect has to be included in the value of the fluid bulk modulus. The air content is assumed to be 0.2 % of the total volume. This assumption is based on measurements and calculations done by using the test equipment previously. It was calculated that the air content varied between 0 and 1 % of the total volume, depending on the pressure and dimensions of the resonator. Thus, the effective bulk modulus of the test equipment B_{eff} with the resonator is 0.187 GPa if the air content is 0.2 %. The effective bulk modulus is

$$\frac{1}{B_{\text{eff}}} = \frac{1}{B_{\text{fluid}}} + \chi \frac{1}{1.4P} \tag{4}$$

where χ is air content and P is the pressure. The resonant frequency of the Helmholtz resonator is calculated by

$$f_{\text{resonator}} = \frac{\sqrt{\frac{B_{\text{eff}}}{\rho}}}{2\pi} \sqrt{\frac{A_{\text{n}}}{l_{\text{n}} l_{\text{c}} A_{\text{c}}}}, \tag{5}$$

where ρ is the density of the fluid, A_{n} is the cross-sectional area of the neck, l_{n} is the length of the neck, l_{c} is the length of the cavity and A_{c} is the cross-sectional area of the cavity. The length of the cavity varies between 0.043 m and 0.243 m. Thus, the resonant frequency of the Helmholtz resonator varies between 17 Hz and 41 Hz. The calculated values are close to the measured values, see Fig. 7.

5 Results of Calculations and Experiments

After the theory review, open-loop control and closed-loop control were chosen to be modelled analytically and verified by experiments. The open-loop control simply identifies the disturbance frequency and then checks the corresponding piston position from a previously produced list containing the disturbance frequencies and corresponding piston positions. The closed-loop control moves the piston through the whole adjusting range and observes the peak-to-peak values in the left pressure transducer, see Fig. 2. After the flow-through, the piston returns to the position where the peak-to-peak value is the smallest and stays there until the peak-to-peak value exceeds the adjusted limit. Kostek and Franchek (2000) have used a similar method in acoustics.

The pressure pulsation's peak-to-peak value (p-p value) was chosen to describe the steadiness of the system because it is very illustrative; simply, the smaller the p-p value, the steadier the system. However, in the control system the greatest advantage of the p-p value is its sensitivity to variations in the system. The p-p value reacts immediately if the pressure, temperature, properties of the oil etc. vary. Thus, the control can lean to the p-p value. For example, the closed-loop control used in this study holds the piston position until the p-p value exceeds the appointed limit. Besides, positive changes can also occur in the system so that the system becomes steadier, in which case the p-p value becomes smaller. In that case there is no sense in carrying out a new adjusting ramp, but instead the position is held. Thus, from now on in this article the main focus is on minimizing the p-p value at different frequencies at the pressure transducer in the left end of the main pipe.

With both control cases it should be remembered that their application field is now hydraulics, which sets some limitations on the control, e.g. the speed of the piston, which causes pressure variations in the system. Thus, the pace of the piston movements is limited already in the analytical models, because otherwise the program would force the piston to a new position so quickly that destructive pressure variations could not be avoided in the test equipment.

5.1 Analytically Calculated Test Drives

Equation 5 is used to calculate the resonant frequency of the Helmholtz resonator as a function of the cavity length (piston position) in 42 different positions, and the results are fitted to the list (resonant frequency vs. piston position). The list is the basis of the open-loop control. The control program identifies the excitation frequency and compares it with the values of the resonant frequencies on the list. After the corresponding value is found, the piston is moved to the correct position, where it stays until the resonant frequency varies. If the identified excitation frequency is not mentioned on the list, the program interpolates the excitation frequency and the corresponding piston position. If the identified excitation frequency is outside the list (above or below), the piston is moved to the corresponding limit position.

Figure 4 presents the result of an analytically calculated test drive. The upper level depicts the excitation frequency, the middle level depicts the corresponding piston position, which is checked from the list, and the lowest level depicts the corresponding p-p value of the primary system (main pipe).

As presented in Fig. 4, the open-loop control maintains the p-p value nearest to zero in tuned conditions. Of course, because the model does not include any damping, the response of the primary system should be zero in the tuned conditions, but in this case this will not be reached. The reason is that the control list is created from analytically calculated results where the piston position is fitted to the equation and the corresponding frequency is calculated. In the model we used, the excitation frequency is identified from the time domain by first doing a FFT. Thus, the accuracy of the FFT affects the results. The measurement frequency of 500 Hz and the block size of 1024 samples do not make sufficient accuracy possible.

For example, if the exact excitation frequency is 17 Hz, after the FFT the control program sees a value of 17.09 Hz. The error might sound trivial, but on the list the corresponding piston positions are 240 mm and 237.5 mm. Thus, it is understandable that the zero level is not reached. This problem is emphasized in the vicinity of natural frequency of the main pipe (23 Hz) where amplitude raises quickly and in the low-frequency range, as seen in Fig. 4, wherein the p-p value differs most from zero at low excitation frequencies.

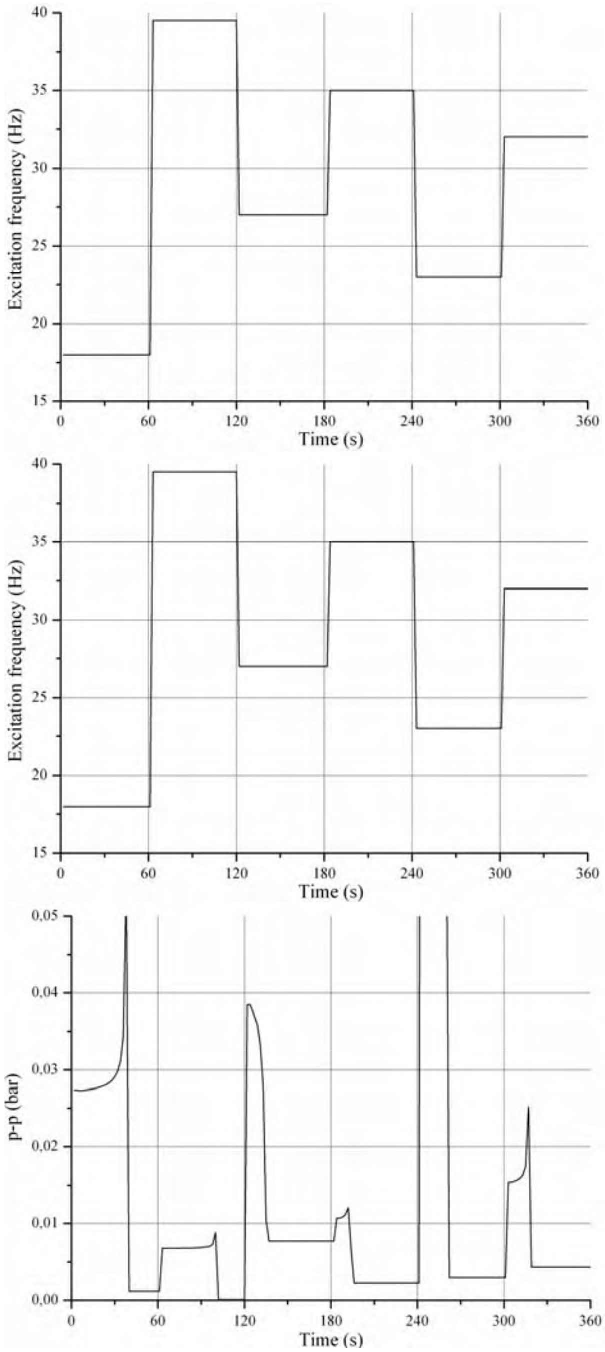

Fig. 4: *Analytically calculated test drive of the open-loop control. The top line depicts the excitation frequencies, the middle line depicts the corresponding piston positions and the lowest line depicts the measured p-p values for the primary system (main pipe)*

This happens because in the low-frequency range a change of 0.2 Hz in the excitation frequency causes a 5 mm movement in the piston position. The problem could be avoided by raising the block size, but then controlling would be delayed even more. Another way could be to include the error caused by the FFT on the control list beforehand, but then the control list would correspond exactly with only a control unit with the same measurement frequency and block size. For example, on the control list the piston position for an excitation frequency of 17 Hz is 0.240 m, but on the "modified" list the piston position for the excitation frequency of 17 Hz should be 0.238 m (which on the original list corresponds to a frequency of 17.1 Hz). The third way could be a two-part program wherein the excitation frequency would be identified accurately by the first part and the other part would only "ask" the correct value from the first part to execute the control. The time delay caused by an accurate FFT (for example 8192 samples) could be avoided by programming the control so that the first part of the program would give a rough piston position immediately, so that the piston movement could be started before the exact value is collated. Naturally, the best way would be to organize the frequency identification without the FFT from the time domain.

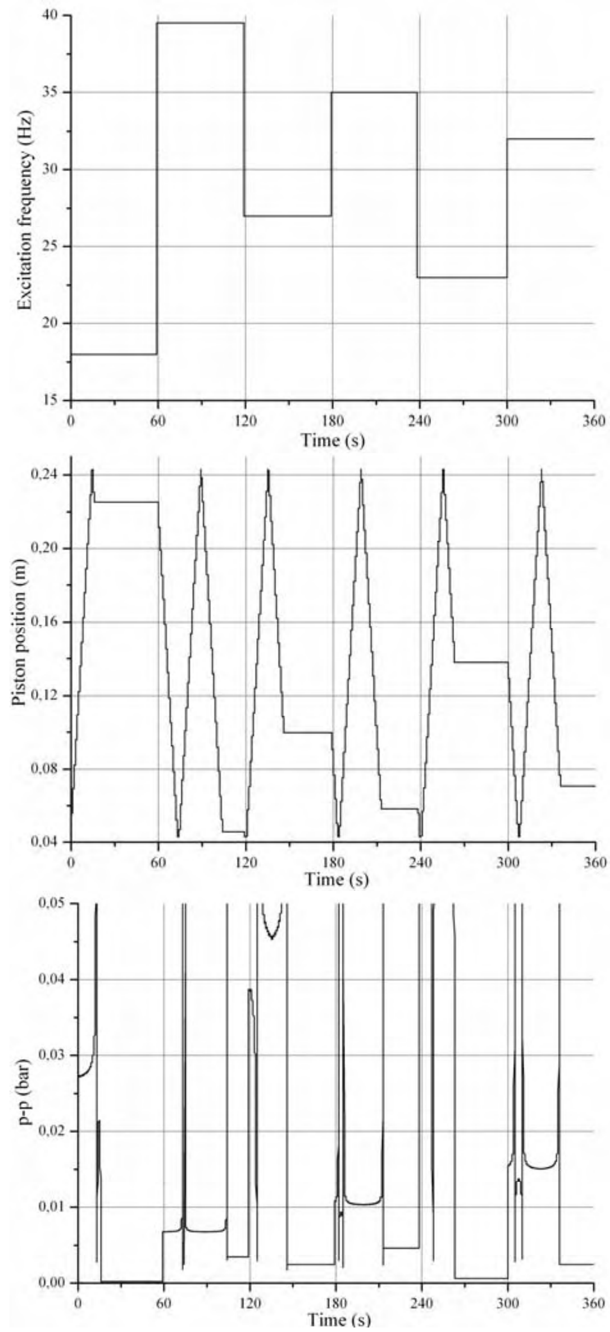

Fig. 5: *Analytically calculated test drive of the closed-loop control. The top line depicts the excitation frequencies, the middle line depicts the corresponding piston positions and the lowest line depicts the measured p-p values of the primary system (main pipe)*

Figure 5 presents an analytically calculated test drive corresponding to the one in Fig. 4, but the closed-loop control is used. As noted from the results, first the piston is moved through the adjusting range and at the same time the primary system's (main pipe's) p-p value is detected. After the piston has reached the limit of the adjustment range, it is returned to the position where the minimal p-p value was measured. This position is maintained until the p-p value exceeds a certain limit. Now, the exceeding is caused by varying the excitation frequency, which influences the p-p value. After the exceeding of the limit value, the piston is returned to the starting point and a new tuning loop is started automatically. In the closed-loop control the FFT is not used and the block size is only 32 samples to guarantee faster control.

5.2 Experimental Test Drives

Figure 6 presents the measured p-p values as a function of the excitation frequencies in the left pressure transducer when the Helmholtz resonator is restricted from the main pipe. The excitation is caused by a piston inside the main pipe, see Fig. 2, which is moved at different frequencies. The frequency of the piston movement is controlled manually. The measured p-p values without the Helmholtz resonator should be decreased by using an adjustable Helmholtz resonator and open-loop and closed-loop controls.

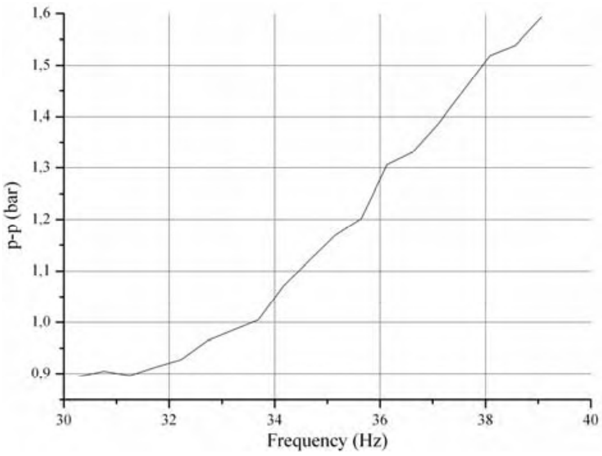

Fig. 6: *Example of the measured pressure's peak to peak values as a function of frequency when the Helmholtz resonator is restricted from the system*

The control list was produced for the open-loop control by defining the maximum attenuation of the pressure pulsations in the left pressure transducer at eleven different piston positions. Thus, the control list included eleven pairs of frequencies and corresponding piston positions, so that the control program compares the excitation frequency with the frequencies mentioned on the list and moves the piston to the correct position. The definitions of the list (calibration) were produced so that the piston was moved to a certain position and two excitation frequency ramps were driven, first one from 5 Hz to 50 Hz and then another from 50 Hz to 5 Hz. Fig. 7 presents an example of the measured responses at one piston position. It was decided to construct the control list so that the minimum response of TFE (transfer function estimation by Matlab™) between the pressure sensors at the beginning and end of the main pipe was reached.

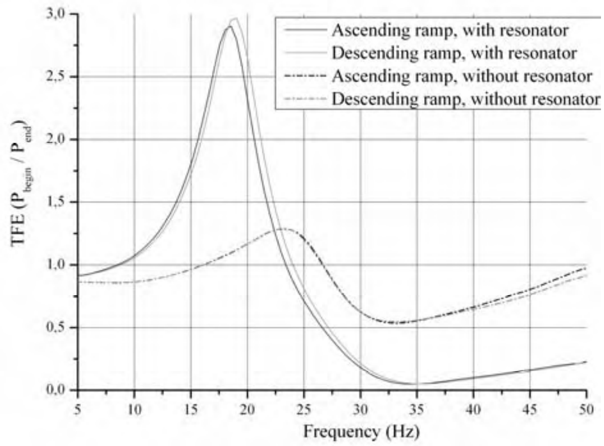

Fig. 7: *Example of the measurement results (transfer function estimation vs. frequency) that were used to construct the open-loop control list*

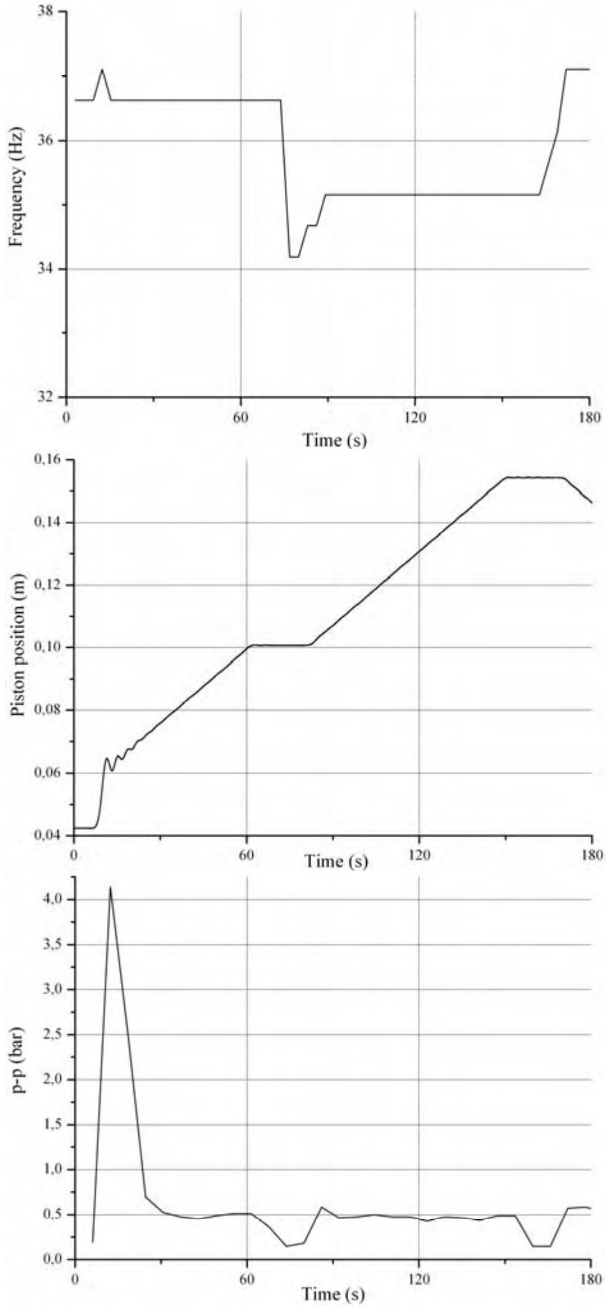

Fig. 8: *Example of an open-loop control test drive*

Figure 8 presents a test drive of the open-loop control with the test equipment depicted in Fig. 1 and 2. The control program has identified the disturbance frequency and adjusted the piston to the correct position, where the pressure's p-p value has been attenuated dramatically; see Fig. 6, which presents the pressure's p-p values in the system without the Helmholtz resonator. The piston movement causes pressure variations in the system so that the measured pressure's p-p values fluctuate significantly during the movement and a steady state is not reached before the piston has stopped.

As observed, the adjusting frequency range is narrow. More differences between the measurements and the calculations arise when the system warms up above the designed temperature, as seen in Table 2, which presents the results of three different open-loop control test drives. The temperature increases between the test drives, and this is noted also from the results, because as the temperature varies the resonant frequency of the Helmholtz resonator also varies. This means that maximum attenuation is reached at a different piston position if the excitation frequency remains constant. In other words, the value pointed out on the control list will no longer correspond to reality. And as seen from Table 2, maximum attenuation decreases as the temperature of the fluid increases during the measurements. The temperature of the hydraulic oil varied between 20˚C and 22˚C as the values of the control list were defined.

Table 2: Measured maximum attenuations of the open-loop control

Temperature	Excitation frequency	Piston position	p-p with the resonator	p-p without the resonator	Attenuation
[˚C]	[Hz]	[mm]	[bar]	[bar]	dB
20…	35.6	134	0.12	1.20	-20
	38.0	44	0.16	1.52	-20
21	34.2	190	0.12	1.07	-19
21.5…	36.6	100	0.15	1.33	-19
	35.2	154	0.14	1.17	-18
22.5	37.1	81	0.17	1.39	-18
23…	35.2	154	0.16	1.17	-17
	31.7	239	0.16	0.91	-15
24	38.6	42	0.23	1.54	-17

Figure 9 presents a test drive of the closed-loop control. As noted from Fig. 9, the piston is moved 15 mm or 20 mm and then the piston movement is paused for a while to check the corresponding p-p value. This procedure is repeated so that the whole adjusting range is checked (there is a safety margin of 5 mm at both ends of the hydraulic cylinder) and then the piston is returned to the position where the minimal p-p value was reached. This position is kept until the pressure's p-p value exceeds 0.175 bar. In the test equipment the p-p value is controlled by the excitation frequency, which can be changed, whereupon the pressure's p-p value changes. To guarantee the efficiency of the control, a maximal amount of samples are handled and averaging of the results is avoided. Thus, the measured p-p values fluctuate, as presented in Fig 9. However, the control program notes the lowest p-p value even if it is only one value (point), thus it is reasonable to observe every measurement value.

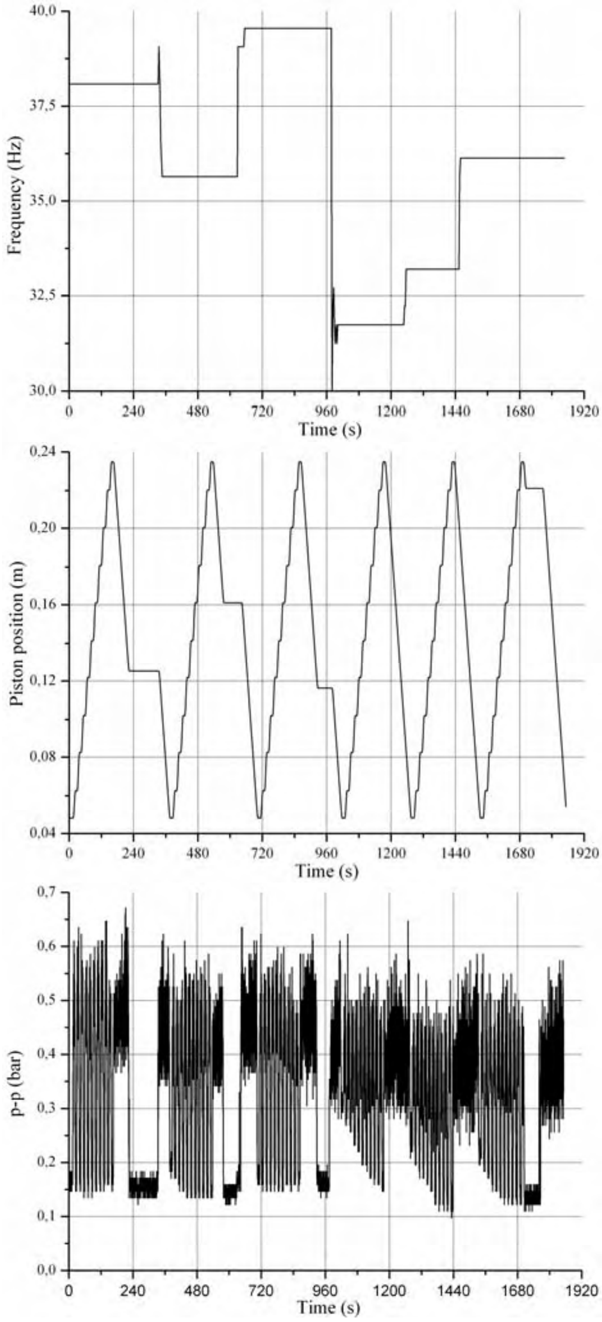

Fig. 9: *Example of a closed-loop control test drive. Note that the frequencies 31.5 Hz and 33 Hz were outside the adjusting range*

The results depicted in Fig. 9 include two frequencies, 31.7 Hz and 33.2 Hz, which are outside the adjustment range. This is also noticed from the measured p-p values, which decrease rapidly during piston movement. The control program would try to find the minimum position again and again, but in the experiments the excitation frequency was varied so that the system finally returned to within the adjustment range.

During the measurements the temperature variation was noted once again. As seen in Fig. 9, in the beginning of the measurements the smallest p-p value for the frequency of 36 Hz was found at the piston position of 0.161 m, and at the end of the test drive the smallest p-p value for the frequency of 35.6 Hz was found at the piston position of 0.221 m. This difference might be caused by the temperature variation, which was over two degrees during the measurement. Another explanation might be the air content of the low-pressure system. Entrained air dissolves from the system, whereupon the stiffness of the system changes. However, this kind of phenomenon supports the closed-loop control as long as it remains within the adjusting range of the Helmholtz resonator.

The presented closed-loop program is slow, but it is simple to code and reliable. However, in hydraulic systems fast piston movements cannot be used because of pressure variations, which might cause damages. Thus, the controls used in hydraulic systems are always slow if compared with mechanics. Nevertheless, the chosen method should be developed more. The next-generation program should include information on the optimal direction so that the piston could avoid unnecessary movements, the piston is controlled in the correct direction and a new optimum could be reached without the entire measuring loop. Enhancement of the control could be achieved by measuring phase differences.

Table 3 presents the maximum attenuation in the left pressure transducer, see Fig. 2, when the closed-loop control is used. As noted from the results, the adjusting range varied as the hydraulic oil aged. However, the closed-loop control maintained -20 dB attenuation while the excitation frequency was within the adjusting range, which moved slightly between the measurements because of temperature and air content.

Table 3: Measured maximum attenuations of the closed-loop control

Temperature	Excitation frequency	Piston position	p-p with the resonator	p-p without the resonator	Attenuation
[˚C]	[Hz]	[mm]	[bar]	[bar]	dB
26…	38.0	125	0.17	1.64	-19
	35.6	161	0.15	1.38	-19
29	39.0	116	0.17	1.72	-20
	36.1	221	0.15	1.44	-20
21…	37.6	166	0.16	1.56	-20
	36.6	182	0.16	1.51	-19
23	41.0	134	0.18	1.77	-20
24…	44.4	110	0.19	1.64	-19
25	46.4	90	0.19	1.53	-20

6 Conclusions

The main focus of the study was to test an adjustable Helmholtz resonator in a low-pressure hydraulic system and to point out that it can be controlled so that maximal damping can be maintained even though the excitation frequency varies. Thus, two control methods, open loop and closed loop, were modelled analytically and verified experimentally. The models and experiments prove that the adjustable Helmholtz resonator, which is well known in acoustics, also works in a low-pressure hydraulic system.

The literature review presented many studies with results of control methods of adjustable Helmholtz resonators. It was clearly seen that there is an obvious lack of research on the subject presented in this paper. The theory review was used as a basis for analytical modelling and experiments, so that the open-loop and closed-loop methods were chosen to be applied in this study. The open-loop method used a calibration list that defines the correct volume of the cavity (piston position) for the identified excitation frequency to reach maximal steadiness in the hydraulic system. The calibration list must be produced beforehand and is case-specific, so that it works in a certain machine and in certain conditions. The closed-loop method searches for maximum attenuation of the pressure variations in the hydraulic system by itself. In this article the closed-loop method used the method wherein the resonator adjusted itself throughout the whole adjusting range and then returned to the position that caused the smallest pressure p-p value at the measurement point.

After the theory review, analytical modelling was carried out and the spring-mass model was found to be suitable for controlling hydraulics if the parameters are correct. The most important benefit of the spring-mass model is the time domain wherein it is solved. The time domain is easy to adapt to the frequency domain, and the time domain is more illustrative when the control methods of absorbers, like the ones of the Helmholtz resonator, are studied. Of course, the effect of damping (viscosity) should be taken into account if hydraulics is studied. However, the effect of damping was omitted in this study because it was difficult to find any comparable results to support the assumptions made during the research. In addition, the low pressure we used caused some uncertainty in the modelling because of the temperature variation and air content. The main pipe in the system was expected to be airless, but the Helmholtz resonator was expected to contain some dissolved and entrained air (air bubbles) because of the shape of the construction and increased volume. The amount of air was expected to be 0.2 % of the total volume of the system, whereby the modelled results were comparable with the measured results. Thus, because of temperature and air content, taking viscosity into account would only add one unfamiliar factor to the model without achieving any considerable benefit. However, during the measurements it was noted that the quality of the hydraulic oil has an obvious effect on the results, which should be studied further.

The experiments were the most complicated part of the research, because the system properties did not reach a steady state due to air content and temperature variation. Although the control methods were proven to be efficient - even -20 dB continuous attenuations were determined - the test equipment was not in total control. The most complicated part was the used oil, whose properties were found to change during the measurements. These changes were noticed clearly in the measurements with the Helmholtz resonator. If the Helmholtz resonator was restricted from the system, the sensitivity to the quality of the oil weakened. In addition, the variation in the oil properties had a hysteretic nature, so that the properties did not return back to the starting point after the system was returned to the starting point. In fact, the original properties were returned only by changing all of the oil in the system. However, these drawbacks were accepted because the main purpose was to prove that the adjustable Helmholtz resonator can be controlled also in a hydraulic system. It is also known that real hydraulic machine conditions remain almost constant at a given speed while in operation. Thus, the variation problems that were noted during the experiments are not as destructive in real machines. Presumably in steady conditions, like in mills, even the open-loop control would work perfectly. But the closed-loop control is more reliable if environmental conditions, e.g. temperature, vary. Such variations in the environmental conditions should not be allowed to be too large. Nevertheless, the closed-loop control is very robust and steady, but unfortunately much slower than the open-loop control.

Thus, to continue the research, a new hydraulic circuit must be chosen, or at least the small hydraulic machine unit has to be changed. Also another adjustable Helmholtz resonator should be constructed so that extreme dimensions could be avoided; cf. the Helmholtz resonator we used had a neck diameter of 6 mm and a cavity diameter of 100 mm. And definitely work with a model of the Helmholtz resonator that takes into account the effect of viscosity should be continued.

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Nomenclature

ρ	density
$\O_{in}$	inner diameter
$\O_{out}$	outer diameter
χ	air content
ω	angular velocity
A_i	cross-sectional area of the component i
B_i	bulk modulus of the component i
F	force
K	spring constant (stiffness) of the primary system
k	spring constant (stiffness) of the vibration absorber
L, l	length
M	mass of the primary system
m	mass of the vibration absorber
P	pressure
t	time
X	position of the primary system
x	position of the vibration absorber

References

Beards, C. F. 1995. Engineering Vibration Analysis with Application to Control Systems. Edward Arnold, Great Britain.

de Bedout, J. M., **Franchek, M. A.**, **Bernhard, R. J.** and **Mongeau, L.** 1997. Adaptive-passive noise control with self-tuning Helmholtz resonators. *Journal of Sound and Vibration,* Vol. 202, No. 1, pp. 109-123.

Estève, S.J. and Johnson, M.E. 2004. Development of an adaptive Helmholtz resonator for broadband noise control. *2004 ASME International Mechanical Engineering Congress*, Anaheim CA, pp. 7.

Estève, S. J. and **Johnson, M. E.** 2005. Adaptive Helmholtz resonators and passive vibration absorbers for cylinder interior noise control. *Journal of Sound and Vibration*, Vol. 288, pp. 1105-1130.

Johnson, M. E. and **Estève, S.** 2002. Comparison of local and global adaptive strategies for the control of broadband noise in an enclosure using adaptive Helmholtz resonators. *ACTIVE 2002, ISVR*, Southampton, UK, pp. 1111-1120.

Kela, L. and **Vähäoja, P.** 2009. Measuring pressure wave velocity in a hydraulic system. *Proceedings of World Academy of Science, Engineering and Technology*, Vol. 37, WCSET 2009, Dubai, United Arab Emirates, pp. 610-616.

Kostek, T. M. and **Franchek, M. A.** 2000. Hybrid noise control in ducts. *Journal of Sound and Vibration*, Vol. 237, No. 1, pp. 81-100.

Matsuhisa, H., **Ren, B.** and **Sato, S.** 1992. Semiactive Control of Duct Noise by a Volume-Variable Resonator. *The Japan Society of Mechanical Engineers*, Vol. 35, No. 2, pp. 223-228.

Singh, S., **Howard, C. Q.** and **Hansen, C. H.** 2006. Tuning a semi-active Helmholtz resonator. *Active 2006*, Adelaide, Australia, pp. 12.

Viersma, T. J. 1980. *Studies in Mechanical Engineering I – Analysis, Synthesis and Design of Hydraulic Servosystems and Pipelines*. Elsevier Scientific Publishing Company, Netherlands.

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