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RESEARCH ON PULSATION ATTENUATION CHARACTERISTICS OF SILENCERS IN PRACTICAL FLUID POWER SYSTEMS

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Abstract

The pulsation attenuation characteristics of silencers in practical fluid power systems are investigated theoretically and experimentally in terms of the insertion loss characteristics, taking the every key circuit-design factors including the pump and the load into consideration, so as to develop an useful CAE design tool for reducing the system audible noise level through the use of a fluid-borne vibration control technique. A new expression for the insertion loss characteristics is also proposed which makes the physical elucidation of its complex characteristics easy and helps to search for the optimum insertion location efficiently.

Particular attention has been paid to both the theoretical determination of the optimum insertion location and the optimum design of silencer for the specified hydraulic circuit. The simulated results of the insertion loss characteristics agree with the experimental results with sufficient accuracy for practical use up to around 5th harmonic of the pump-induced harmonic pressure pulsation. Further, it is shown that our developed new silencer called a “variable-resonance mode type side-branch resonator” has proven to be very successful for reduction of audible noise also in application to a real hydraulic excavator.

Keywords: fluid-borne noise, pressure pulsation, hydraulic silencer, side-branch resonator, insertion loss, noise reduction

1 Introduction

In fluid power systems, pump-induced harmonic pressure pulsations known as the fluid-borne vibration are usually the primary source of system noise because they are very easily transmitted throughout the entire system and then excite the mechanical vibrations that generate audible noise. One of the most effective means of reducing the pump-induced fluid-borne noise is to decrease the amplitude of the pressure pulsations by inserting a reactive type of silencer in a pump discharge line. To what extent this is realized is determined by the performance of a silencer. The transmission loss (*TL*) and the insertion loss (*IL*) have been commonly used as a silencer performance criteria (Henderson, 1980). Of these, transmission loss is the inherent characteristics of a silencer itself, but it is merely the performance in the specific system where the end load is standardized as an infinitely long pipe, which corresponds to a characteristic impedance (i.e. an anechoic termination) at the outlet of a silencer. On the other hand, the insertion loss is the difference in amplitude of pressure pulsations measured at a point in the downstream line with and without a silencer and is the final measure of how effective a silencer is. However, since the insertion loss involves the complex interaction of the wave propagation characteristics in both the upstream line including the pump pulsation source and the downstream line including the load as well as in the silencer, it is not easy to understand fully its properties for the realistic fluid power systems. For this reason, little detailed study of the insertion loss has been undertaken in the past which takes into consideration the key factors in practical fluid power systems and is validated by the reliable experimental results, except the studies in simple and special conditions (Gassman, 1987; Washio et al, 1988; Strunk, 1991; Bais and Toet, 1992). Consequently, in reality, the fluid power industry has resorted to empirical methods and subjective evaluations in inserting the silencer.

In this paper, firstly, the general theoretical expression of the insertion loss characteristics of reactive type silencers is produced taking all the key circuit-design factors of practical fluid power systems including the pump pulsation source (source flow pulsation and source impedance) and the load into consideration, and then a new expression for the insertion loss characteristics of a branch type silencer is proposed which makes

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the physical elucidation of its complex characteristics easy and help to search for the optimum insertion location efficiently. Next, the accuracy of the present theoretical model is examined by comparison with many experimental results carried out over the wide test conditions in the bench circuit changing the insertion location variously for the different two types of silencers under the different two load conditions. Finally, it is shown that the new silencer called a “variable-resonance mode type of side-branch resonator”, which was devised by one of the authors (Kojima), has proven to be very successful for reduction of audible noise as well as fluid-borne vibration also in the application to a real hydraulic excavator.

2 Theoretical Model of Insertion Loss Characteristics in a Practical Fluid Power System

The typical way of evaluating the performance of a silencer is to determine the insertion loss. This is the difference in pressure amplitudes in the downstream line with and without a silencer, P and P', which is defined with the following expression in units of decibels

$$IL = 20\log_{10}\left|\frac{P'}{P}\right| \quad (1)$$

Figure 1 shows the hydraulic circuit in which a silencer is inserted at some position in the pump discharge line. Where, the pump source characteristics can be modeled either by (a) the flow pulsation source with a source impedance in parallel at the pump exit as shown in Fig. 1 (a) (hereafter called a standard model) or (b) the flow pulsation source located at the inner end of the discharge passageway in a pump casing as shown in Fig. 1 (b) (hereafter called a revised model) (Edge and Johnston, 1990; Weddfelt, 1992; Kojima, 1992).

First, the standard model will be used as the pump pulsation source. Assuming the flow in the upstream line to be the two-dimensional, laminar, viscous and compressible, the relationship between the variables, pressures and flows, at each end of the line can be described using a Laplace transfer matrix [a] as follows.

$$\begin{bmatrix} P_0 \\ Q_0 \end{bmatrix} = \begin{bmatrix} \cosh(\beta\, l_1) & Z_C \sinh(\beta\, l_1) \\ \sinh(\beta\, l_1)/Z_C & \cosh(\beta\, l_1) \end{bmatrix} \begin{bmatrix} P_1 \\ Q_1 \end{bmatrix} \quad (2)$$

where wave propagation coefficient, β, and characteristics impedance, Z_C, are given using a complex coefficient representing the two-dimensional laminar resistance effects, ξ, by

$$\beta = \frac{\xi}{c} s \quad (3)$$

$$\beta\ l_1\ Z_C = \frac{\rho\ c\ \xi}{\pi\ r^2} \quad (4)$$

$$\xi \cong 1 + \sqrt{\frac{\nu}{r^2 s}} + \frac{\nu}{r^2 s} \quad (5)$$

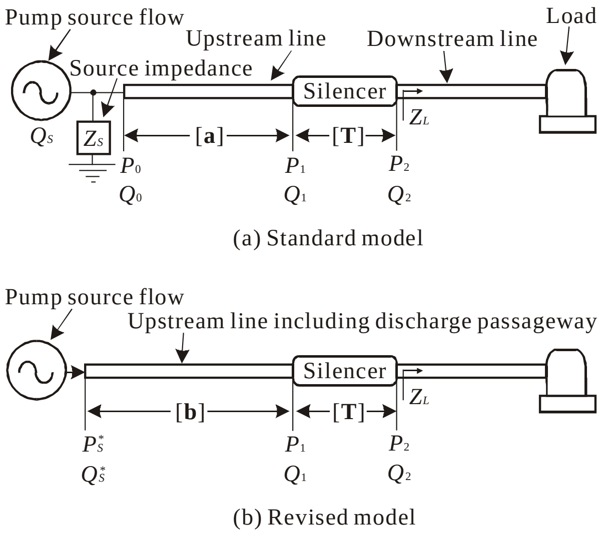


Fig. 1: *Hydraulic circuits with a silencer inserted and typical models for pump pulsation source*

The boundary condition at the pump exit ($x = 0$) relates to the source flow pulsation, Q_S, and the loading effect of the source impedance, Z_S, on the flow pulsation entering the upstream line, Q_0, thus,

$$Q_0 = Q_S - \frac{P_0}{Z_S} \quad (6)$$

The relationship between the variables at the inlet and outlet of a silencer also can be described using a Laplace transfer matrix.

$$\begin{bmatrix} P_1 \\ Q_1 \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} P_2 \\ Q_2 \end{bmatrix} \quad (7)$$

where the transfer matrix parameters, T_{11}-T_{22}, are functions of only the silencer and independent of the circuit.

The boundary condition at the outlet of a silencer (i.e. at the inlet of downstream line) is defined by the load impedance Z_L, which can be found analytically as follows if the termination impedance Z_T and the transfer matrix parameters c_{11}-c_{22} of the downstream circuit including various system components are known.

$$Z_L = \frac{P_2}{Q_2} = \frac{Z_T\ c_{11} + c_{12}}{Z_T\ c_{21} + c_{22}} \quad (8)$$

Thus, the pressure pulsation at the inlet of the downstream line due to the pump source flow pulsation can be derived as follows

$$P_2 = Z_S\ Z_L\ Q_S / \left\{ \left(a_{11} + a_{21} Z_S\right)\left(T_{11} Z_L + T_{12}\right) + \left(a_{12} + a_{22} Z_S\right)\left(T_{21} Z_L + T_{22}\right) \right\} \quad (9)$$

If the silencer is not in the circuit, Eq. (9) is reduced to, since $T_{11} = T_{22} = 1$ and $T_{12} = T_{21} = 0$,

$$P_2' = \frac{Z_S\ Z_L}{\left(a_{11} + a_{21} Z_S\right) Z_L + \left(a_{12} + a_{22} Z_S\right)} Q_S \quad (10)$$

Hence, the expression of the insertion loss defined by Eq. (1) can be derived from the last two equations as follows

$$IL = 20\log_{10}\left|\left\{\left(a_{11}+a_{21}Z_S\right)\left(T_{11}Z_L+T_{12}\right)+\left(a_{12}+a_{22}Z_S\right)\times\left(T_{21}Z_L+T_{22}\right)\right\}/\left\{\left(a_{11}+a_{21}Z_S\right)Z_L+\left(a_{12}+a_{22}Z_S\right)\right\}\right| \quad (11)$$

If using the revised model shown in Fig. 1 (b) as a pump pulsation source and assuming the discharge passageway being able to be modeled as a uniform pipe with same characteristic impedance as a pump discharge line (i.e. a silencer upstream line) and with a length estimated from the frequency characteristics of the experimentally-determined source impedance Z_S, the equivalent equations to Eq. (9), (10) and (11) can be obtained as follows.

$$P_2 = \frac{Z_L}{b_{21}\left(T_{11}Z_L+T_{12}\right)+b_{22}\left(T_{21}Z_L+T_{22}\right)}Q_S^* \quad (12)$$

$$P_2' = \frac{Z_L}{b_{21}Z_L+b_{22}}Q_S^* \quad (13)$$

$$IL = 20\log_{10}\left|\frac{b_{21}\left(T_{11}Z_L+T_{12}\right)+b_{22}\left(T_{21}Z_L+T_{22}\right)}{b_{21}Z_L+b_{22}}\right| \quad (14)$$

where b_{11}-b_{22} are the transfer matrix parameters of the upstream line including the discharge passageway in a pump casing.

Note: It should be noted that the insertion loss representing the amplitude ratio of pressure pulsations with and without a silencer is independent of the position because the standing wave ratio is the same in both cases.

Further, for such silencers inserted in the main line in a branch connection as a Helmholz resonator, transfer matrix parameters, T_{11}-T_{22}, in Eq. (7) can be expressed using the entry impedance of a silencer, Z_r, as follows

$$[\mathrm{T}] = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 1/Z_r & 1 \end{bmatrix} \quad (15)$$

Hence, Eq. (11) and Eq. (14) are transformed, respectively, into

$$IL = 20\log_{10}\left|\frac{\left(a_{11}+a_{21}Z_S\right)Z_rZ_L+\left(a_{12}+a_{22}Z_S\right)\left(Z_r+Z_L\right)}{\left(a_{11}+a_{21}Z_S\right)Z_rZ_L+\left(a_{12}+a_{22}Z_S\right)Z_r}\right| \quad (16)$$

$$IL = 20\log_{10}\left|\frac{b_{21}Z_LZ_r+b_{22}\left(Z_L+Z_r\right)}{Z_r\left(b_{21}Z_L+b_{22}\right)}\right| \quad (17)$$

As can be seen from Eq. (16) or Eq. (17), the insertion loss characteristics in a practical fluid power system involve a complex interaction between the wave propagation characteristics of the silencer and the entire circuits. For that reason, it is very difficult to understand even the qualitative dependence of system parameters on its properties from the expression of Eq. (16) or Eq. (17) as it is.

However, by transforming the above equation into the following form, it becomes possible to explain the insertion loss properties uniformly for any combinations of branch type silencers and circuits. The following shows the derivation process of the new expression for the insertion loss.

Seen from the side branch junction, the main line is considered to be the two branch lines acting in parallel as shown in Fig. 2. Hence, the resultant (parallel) impedance of the main line to both the upstream and downstream lines at the side branch junction, Z_e, is given by

$$\frac{1}{Z_e} = \frac{1}{Z_p} + \frac{1}{Z_L} \quad (18)$$

where Z_p is the impedance of the main line to the upstream line (to the source) at the side branch junction and given by Eq. (19) and Eq. (20) for the standard model and the revised model, respectively.

$$Z_p = \frac{a_{12}+a_{22}Z_S}{a_{11}+a_{21}Z_S} \quad (19)$$

$$Z_p = \frac{b_{22}}{b_{21}} \quad (20)$$

Hence, the alternative expression of the insertion loss for the branch-connection type silencers can be obtained, by substituting Eq. (18) and (19) to Eq. (16) or (18) and Eq. (20) to (17), for both the models as follows.

$$IL = 20\log_{10}\left|1+\frac{Z_e}{Z_r}\right| \quad (21)$$

$$\equiv 20\log_{10}\left|1+\bar{K}e^{j\theta}\right|$$

Newly proposed Eq. (21) indicates that the insertion loss can be expressed by the function of only the complex ratio ($\bar{K}$: amplitude ratio, θ: phase difference) of the resultant impedance of the main line at the side-branch junction, Z_e, to the entry impedance of the branch type silencer, Z_r, independent of the combinations of silencers and circuits.

Figure 3 is the graphical representation of Eq. (21) showing the insertion loss characteristics with the relations of $\bar{K}$ and θ. This figure is of a general character applicable to any combinations of all branch type silencers and all circuits. From this figure it can be seen that the insertion loss increases as the amplitude ratio $\bar{K}$ increases (i.e., as $|Z_r|$ decreases and $|Z_e|$ increases) and the phase difference $|\theta|$ decreases. Moreover, it can be seen that the insertion loss becomes a negative quantity, that is, the amplitude of pressure pulsation may be rather amplified for the insertion of a silencer due to the impedance mismatching when $|\theta|$ is larger than $\pi/2$ and $\bar{K}$ is smaller than 2.

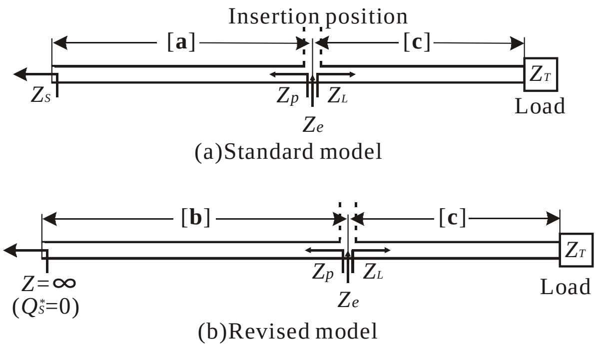


Fig. 2: *Impedance representation of circuit and resultant impedance Z_e of the main line at the side-branch junction*

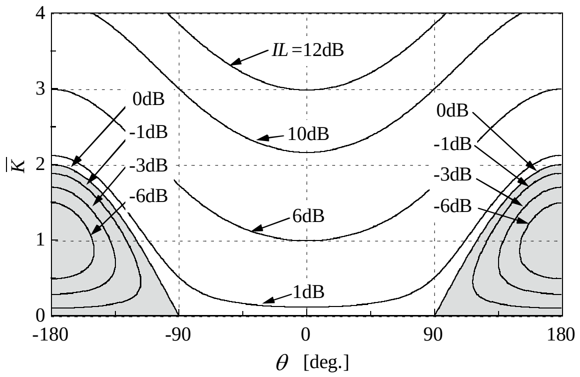


Fig. 3: *Insertion loss with relation of $\overline{K}$ and θ*

3 Experimental Analysis of Insertion Loss Characteristics

3.1 Test Silencers and their Wave Propagation Properties

A conventional quarter-wavelength side-branch resonator and a Kojima's devised new resonator called a "variable-resonance mode type side-branch resonator" shown in Fig. 4 (a) and (b), respectively, were selected as the test silencers. Of these, the variable - resonance mode type side-branch resonator has a feature that the resonance can be generated at the desired frequencies of plural number, with a single closed-end tube configuration, by adjusting the diameters and lengths of the choke orifices and divided parts of the tube (Kojima et al 1998, Kojima and Ichiyanagi, 1998). The present new resonator used in this test has been designed so as to have the resonance frequencies of a successive integer (1, 2 and 3) multiple of the designated lowest resonance frequency (e.g. fundamental frequency of the pump flow pulsation). Both of the test silencers used in chapter 3 were made of steel tube and sized so that the lowest resonance frequency coincides with 250 Hz of the fundamental frequency of pump-induced pressure pulsation. Figure 5 shows the measured and calculated (designed) frequency characteristics of their entry impedance, Z_r, and transmission loss, *TL*. Of these, the measured values were predicted from Eq. (15) using the matrix parameters T_{11}-T_{22} experimentally determined by the "4 pressures/2 systems" method (Kojima and Edge, 1994, Kojima et al 1996).

3.2 Test Hydraulic Circuit and Test Method

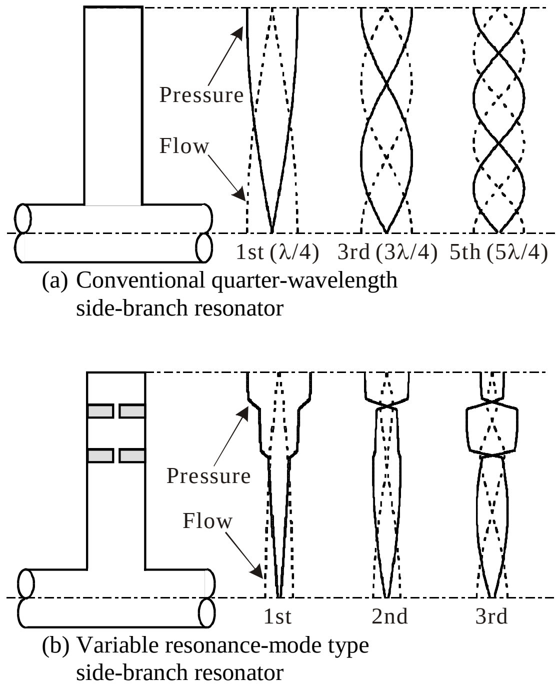


Fig. 4: *Schematic diagram of test silencers and their resonance shapes*

The test hydraulic circuit for measuring the insertion loss characteristics is composed of an axial piston pump, an upstream line (from pump exit to silencer inlet), a test silencer, a downstream line (from silencer outlet to load) and a load (resistive impedance load or capacitive impedance load) as shown in Fig. 6, which is a relatively simple but a realistic hydraulic system. The total length of the main line from the pump exit to the load is always maintained at a fixed value of 2.05 m for the resistive impedance load (valve load) and 2.10 m for the capacitive impedance load (volume load) independent of the insertion position of a silencer, following the practical application in a real system. Hence, the lengths of the upstream and downstream lines were altered in accordance with the insertion location of a silencer. Figure 7 shows the source impedance, Z_s, of the axial piston pump used in this test, which is determined experimentally using the "2 pressures / 2 systems" method devised by Kojima (Kojima, 1992). An anti-resonance corresponding to the quarter-wavelength mode of the discharge passageway, at which the amplitude of the source impedance reaches a minimum and the phase angle switches from -90 to +90, is apparent at around 2.1 kHz. Hence, assuming the speed of sound in fluid to be 1370 m/s of the same value as one in the reference pipe used in the test of Fig. 7, the equivalent length of discharge passageway in a pump casing can be estimated to be around 163 mm from the equation of $L = c/(4\,f)$ (= 1370/(4·2100)).

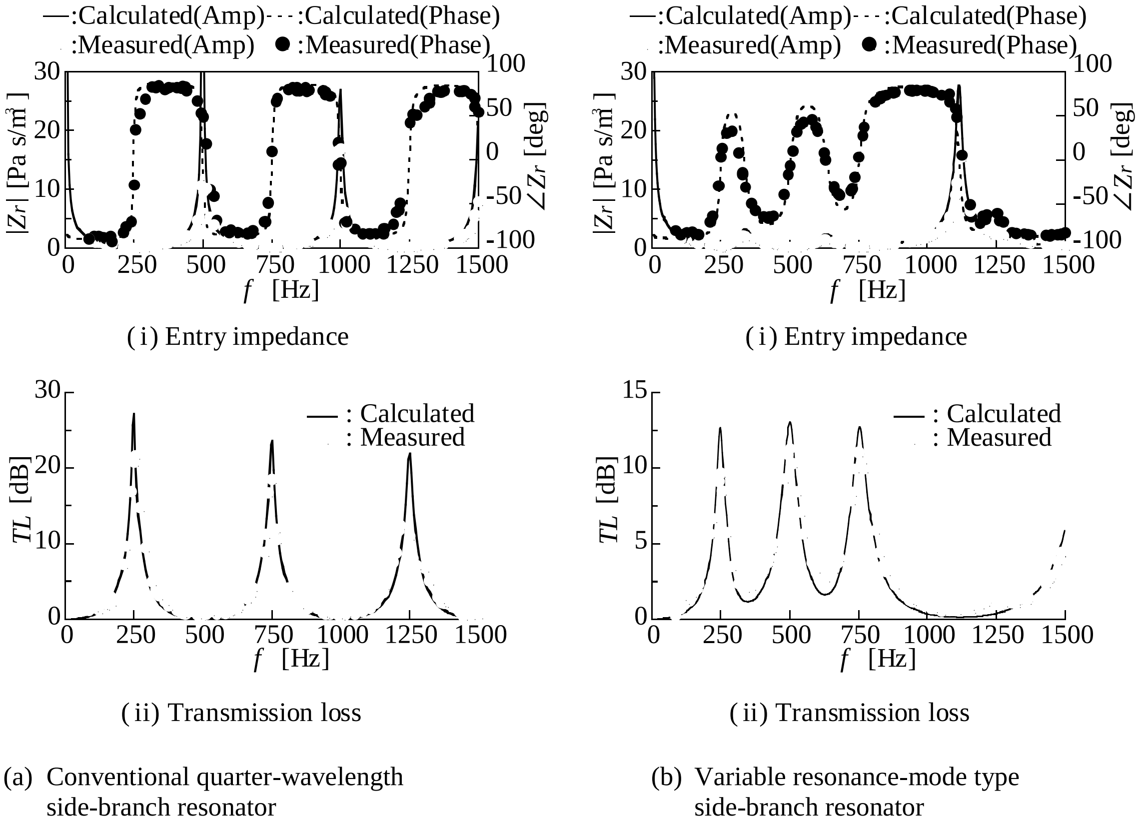


Fig. 5: *Measured and designed (calculated) entry impedance Z_r and transmission loss TL of the test silencers*

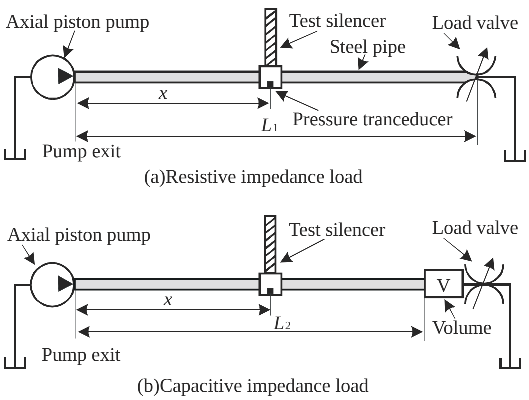


Fig. 6: *Test hydraulic circuits for insertion loss*

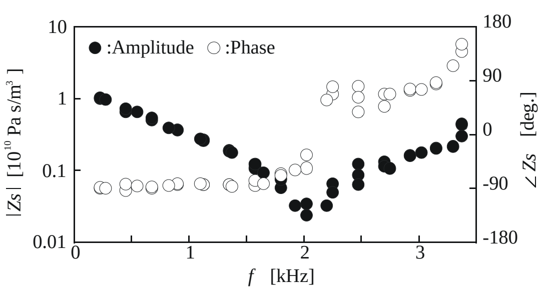


Fig. 7: *Experimentally-determined source impedance of the axial piston pump used in this test*

The test method is as follows. Running a nine-piston pump at 1667 rpm so that the fundamental frequency of the pump-induced pressure pulsation becomes 250 Hz, the pressure pulsation at the inlet of the downstream line with and without the test silencer, P_2, P_2', is measured by a piezo-resistive pressure transducer mounted in the connecting block. Then, the spectral characteristics of the pressure pulsations are analyzed in order to get the insertion loss of each harmonic component.

3.3 Experimental Results and Discussions

Figure 8 and 9 show the experimental results of the insertion loss characteristics curves of the conventional quarter-wavelength side-branch and the variable-resonance mode type side-branch for the two different load conditions, respectively, together with the simulation results calculated from Eq. (17) based on the revised model for the pump pulsation source, which represents the effects of the insertion location on the attenuation performance of each harmonic of the pressure pulsation. Figure 10 shows an example of simulation result of insertion loss of the variable-resonance mode type side-branch calculated from Eq. (16) based on the standard model. As can be seen from Eq. (8) and (11), the insertion loss characteristics is also affected by the termination impedance Z_T, i.e. by the mean delivery pressure P_0 and delivery flow Q_0 (for, $Z_T \cong 2P_0/Q_0$ in the case of valve termination for instance). In this study, however, all tests were carried out under the fixed condition of P_0 = 14.0 MPa and $Q_0 \cong 0.6 \cdot 10^{-3}$ m^3/s. From these results, the following points are noteworthy:

- Difference in the two simulation results of the insertion loss characteristics due to the difference of the pump source model used (i.e. the standard model and the revised model) is negligibly small for practical use (this means that the characteristic values of the revised model can be estimated accurately from the experimentally determined source impedance Z_S and source flow Q_S).

- Experimental results agree with the simulation results with a sufficient accuracy for practical use (this means that the present computer simulation program is a very useful design tool for reducing the system fluid-borne noise level by inserting a reactive silencer in the pump discharge line).
- Insertion loss characteristics are strongly dependent on both the frequency (i.e. harmonic component of the pressure pulsation) and the insertion location of a silencer.

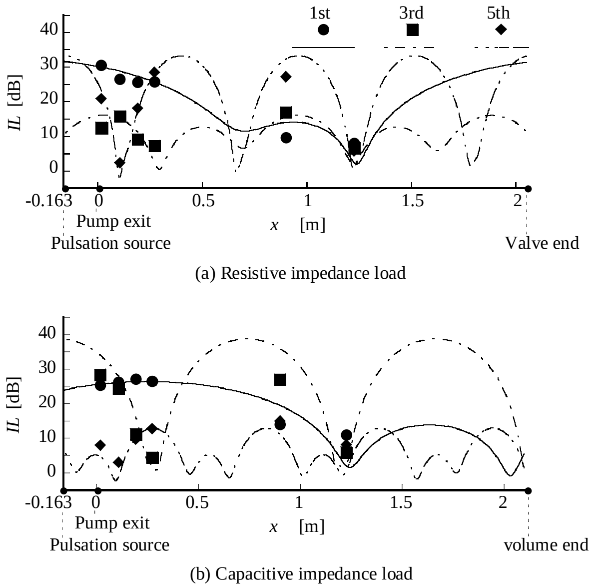


Fig. 8: *Experimental and simulated (based on the "revised model") results of the insertion loss of quarter-wavelength side-branch for (a) resistive and (b) capacitive impedance load*

- Insertion loss becomes minimum at the places where the standing wave of the harmonic pressure pulsation in either upstream line and/or downstream line is close to the nodes. For instance in the case of valve load having the approximately closed-end condition, these are the distances of odd multiples of $c/(4f)$ from the pulsation end of the upstream line and the load end of the downstream line respectively (e.g. for the 3rd harmonic at 750 Hz, odd multiples of $1440/(4 \cdot 750) = 0.48$m).
- Satisfactory attenuation performance can not be attained for the harmonics in the high frequency range (above around 700-800 Hz for the ordinary size of pump) even if a silencer is placed at the position close to the pump exit, because the length of discharge passageway in a pump casing becomes close to the quarter wavelength corresponding to the frequency in question. As mentioned later, some harmonics may be rather amplified by the insertion of a silencer.
- Insertion loss becomes maximum at the positions where both the standing waves of the harmonic pressure pulsation in the upstream and downstream lines become close to the anti-nodes (to be exact, at the positions where the resultant impedance Z_r becomes maximum).
- Silencer should be inserted as close to the pump exit (if possible, as close to the inner end of the discharge passageway in a pump casing) as possible, and moreover the length of the downstream line should be sized so that the silencer insertion position comes to the vicinity of anti-nodes or becomes to be at least sufficiently away from the nodes in order to bring out the latent attenuation capabilities of a silencer maximally.

Next, the phenomenon that the pressure pulsations may be rather amplified for the silencer insertion will be explained physically using Eq. (21) and Fig. 3. The following is an example of analysis elucidating this phenomenon.

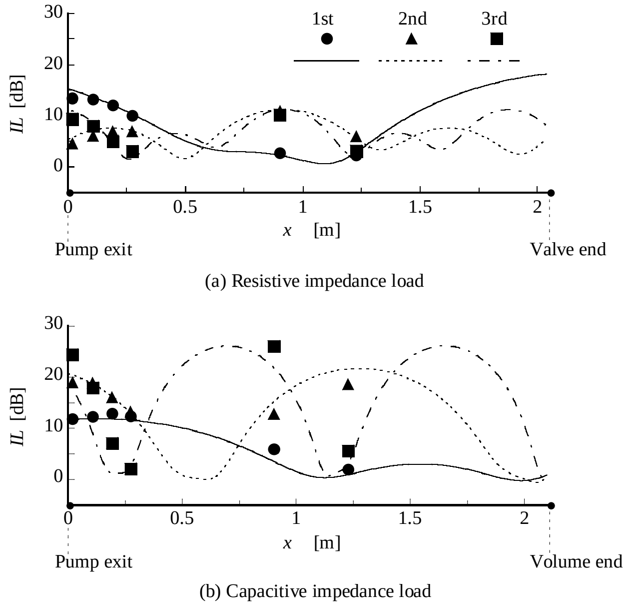


Fig. 9: *Experimental and simulated (based on the "revised model") results of the insertion loss of variable resonance-mode side-branch for (a) resistive and (b) capacitive impedance load*

Fig. 11 (a) shows the insertion loss characteristic curve of the test variable-resonance mode side-branch for the 4th harmonic (in this study, the harmonic of 1000 Hz) of the pump-induced pressure pulsation, and Fig. 11 (b), (c) and (d) show the characteristics of Z_e ($|Z_e|$ and $\angle Z_e$), Z_r ($|Z_r|$ and $\angle Z_r$) and K ($\overline{K}$ and θ) relating to the insertion location, x, for this harmonic, respectively. The insertion loss for the 4th harmonic indicates a negative quantity when the silencer is placed at x = 0.02m and x = 0.05m in this test circuit. This is for the reason that $\overline{K}$ becomes smaller than 1 and θ close to $-\pi$ at these locations and hence the circuit conditions fall into the region of IL coming to negative value shown by the shade in Fig. 3. Phenomenon that the even harmonics of pressure pulsation may be rather amplified or the higher odd harmonic can not be attenuated satisfactorily, which is often encountered with the use of a conventional quarter-wavelength side-branch, are also based on the same reason as the above and can be explained uniformly using Eq. (21) or Fig. 3.

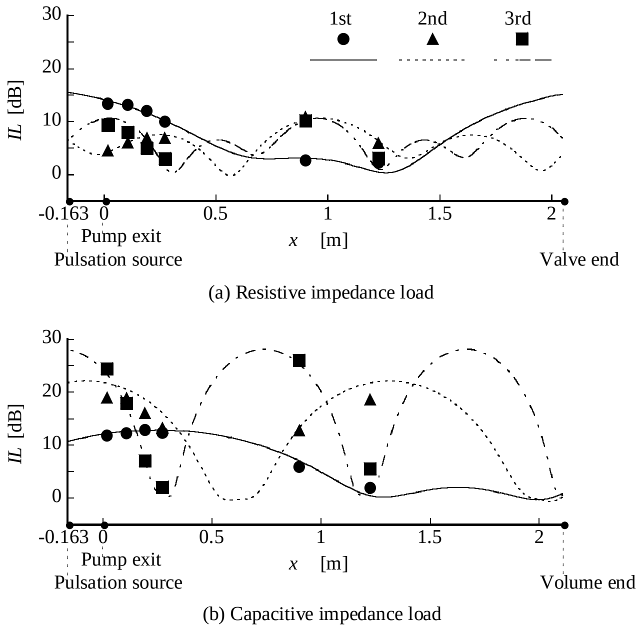


Fig. 10: *Experimental and simulated (based on the "standard model") results of the insertion loss of variable-resonance mode type side-branch for (a) resistive and (b) capacitive impedance load*

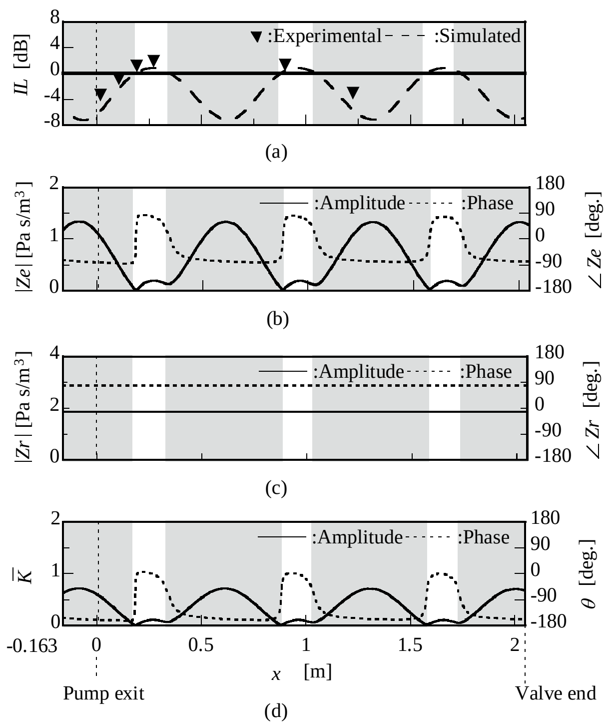


Fig. 11: *IL, Z_e, Z_r and K of the variable-resonance mode type side-branch for 4th harmonic of pressure pulsation relating to the insertion position x*

4 Application of the "Variable-Resonance Mode Type Side-Branch Resonator" to a Real Hydraulic Excavator

In this section, a case example will be shown that our devised "variable resonance-mode type side-branch resonator", which is made of flexible hose as a tube and sized on the basis of the mathematical model of insertion loss characteristics described in the previous chapter, was found to be very useful also for the reduction of the system audible noise radiated by a real hydraulic excavator.

Figure 12 (a) shows the spectra of near-by noise at some point on the arm surface at the time of bucket relief operation of a real hydraulic excavator under test. As can be seen from this figure, when any silencer is not in circuit the noise of fundamental (230 Hz), 2nd (460 Hz) and 3rd (690 Hz) of pumping frequency is radiated strongly. These three components of the noise affect dominantly the system audible noise level and tone quality in practice. Therefore, an optimum design of the present variable-mode type side-branch resonator here is to determine the unknown lengths of choke orifices and divided tube parts (Kojima et al, 1998 or Kojima and Ichiyanagi, 1998) so that the resonance frequency $f_{r.i}$ and the insertion loss $IL(f_{r.i})$ of the present resonator coincide with the desired value $f_{r.i}^*$(i.e., 230 Hz, 460 Hz and 690 Hz in this study) and $IL^*(f_{r.i})$, respectively. The "Powell's conjugate method" (Powell, 1964) was adopted in searching for the design variables. The following equations represent the design variables, the objective function and the constraint conditions.

<u>Design variables:</u>

$$X = \{L_1, \quad L_2, \quad L_3, \quad l_1, \quad l_2\}^T \tag{22}$$

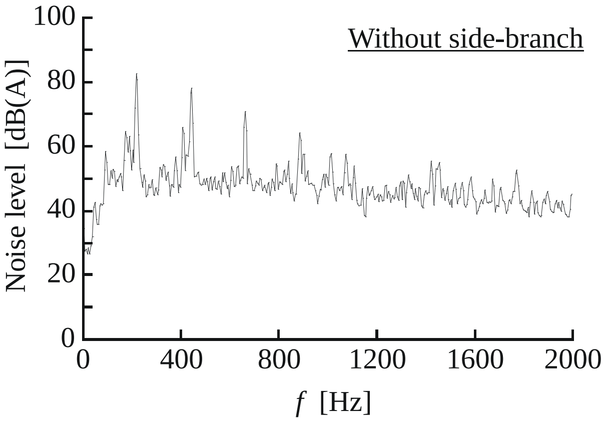
<u>Objective function:</u>

$$P(X) = \left|f_{r.1} - f^*_{r.1}\right| + \left|f_{r.2} - f^*_{r.2}\right| + \left|f_{r.3} - f^*_{r.3}\right| \tag{23}$$

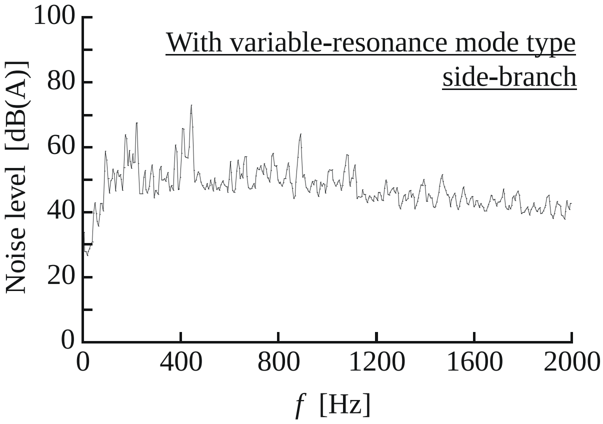
<u>Constraint conditions:</u>

$$IL\left(f_{r.1}\right) = IL\left(f_{r.2}\right) = IL\left(f_{r.3}\right) \tag{24}$$

Unknown parameters Z_S and Z_L in Eq. (11) used for the constraint conditions, pump source impedance and load impedance, were determined experimentally. In the design process of flexible hose made resonator, in other words, in the theoretical estimation of transfer matrix parameters T_{11}-T_{22}, the rigid pipe model given by Eq. (2) was used, approximately changing only the speed of sound in fluid from 1440 m/s (for the rigid pipe) to 1090 m/s. In a flexible rubber hose, it has been known well that a visco-elasticity of a hose wall has a significant effect on wave propagation characteristics in the fluid. But, since the most concern in this work is to conform the resonance frequencies with the desired frequencies, the above approximation may be used.



(a) Without silencer



(b) With silencer

Fig. 12: *Spectra of near-by noise of the arm surface at the time of bucket relief of hydraulic excavator*

Frequency characteristics of source impedance Z_S of the pump installed in test excavator was measured in the bench test stand by the “2 pressures/2 systems” method. Figure 13 shows the measured load impedance Z_L at the pump exit (i.e., at the resonator insertion position in this study) at the bucket relief operation, which was obtained in-situ by the “2 pressures method” using the reference pipe.

Figure 14 shows the experimental measurement of the insertion loss characteristics of the present test resonator, which was obtained directly from Eq. (1) using the measured pressure pulsations at the entrance of bucket cylinder in the downstream line with and without resonator, *P* and *P’*. Provided that the insertion loss of 13.2 dB can be obtained in theory for all harmonics frequencies from 1st to 3rd.

Figure 12 (b) shows the spectra of the near-by noise at the same position as Fig. 12 (a) when the present resonator is inserted just behind the pump discharge port. From the comparison between Fig. 12 (a) and (b), it can be seen that the near-by noise reduction of 15.1 dB for the 1st, 5.1 dB for the 2nd and 13.7 dB for the 3rd harmonic have been achieved by the insertion of the present resonator and hence that our developed “variable resonance-mode type side-branch resonator” is very useful for practical use. The result of Fig. 12 (b) is not necessarily satisfactory yet, because though the optimally designed present resonator has a high insertion loss characteristics of 8.3 dB for pressure pulsation and thereby an excellent audible noise reduction capabilities of 11.3 dB on the frequency average, the second harmonic of pressure pulsation dose not reduce as fully as predicted. But, this disagreement of 2nd harmonic is not due to the fundamentals of the present resonator but supposedly mainly due to the difference in speed of sound in each divided tube part or the insufficiency of mathematical model of flow resistance in a choke orifice neglecting both the inlet and outlet flow loss.

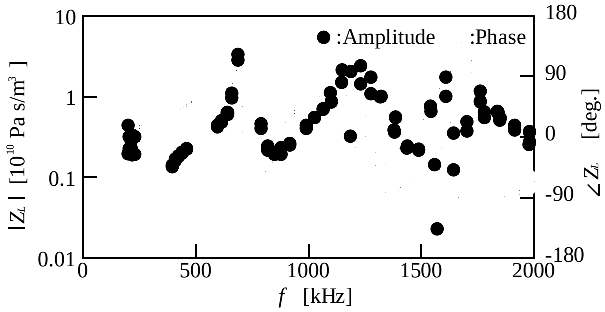


Fig. 13: *Experimentally-determined load impedance Z_L at the pump exit (silencer insertion location)*

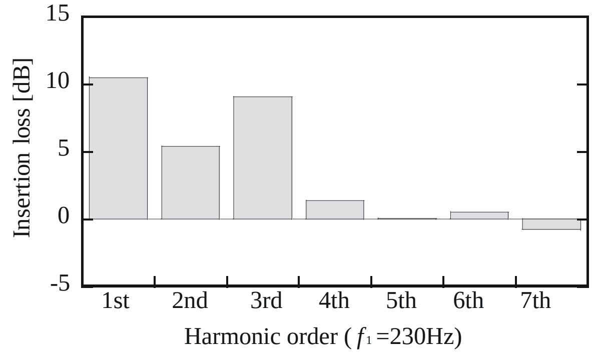


Fig. 14: *Experimental measurement of insertion loss of test silencer*

5 Conclusions

The insertion loss characteristics of a reactive silencer was investigated theoretically and experimentally taking all the key circuit parameters of real hydraulic systems including the pump pulsation source and the load into consideration, for the main purpose of developing the computer simulation program as a useful design tool for reducing the system fluid-borne noise level by inserting a silencer into the pump discharge line. The insertion loss was found to be strongly dependent on the wave propagation characteristics of both the upstream and downstream lines including the discharge passageway in a pump casing (i.e., pump source impedance) and the load, respectively, besides the silencer itself, as well as on the harmonic frequencies in question. A new expression of the insertion loss was also proposed which can describe the attenuation characteristics uniformly in any combination of all branch type silencers and all circuits, using only the complex ratio of the entry impedance of silencer to the resultant (parallel) impedance of main line at the silencer junction. Further, it was also confirmed that the specific resonator called a "variable resonance-mode type side-branch resonator" developed by one of the authors (Kojima), which was sized by the optimum design method so as to reduce the successive 1st to 3rd harmonics of pump-induced pressure pulsation on the basis of the present mathematical model of insertion loss characteristics, could have a satisfactory performance for the reduction of audible noise also in the application to a real hydraulic excavator as well as in the test bench circuit .

Nomenclature

[a]	transfer matrix of upstream line (see Eq. (2))	
[b]	transfer matrix of upstream line including discharge passageway in pump	
[**c**]	transfer matrix of downstream circuit	
c	speed of sound in fluid	[m/s]
f	frequency of pulsation	[1/s]
IL	insertion loss (see Eq. (1))	[-]
K	complex ratio of resultant impedance of main line Z_e to entry impedance of silencer Z_r (see Eq. (21))	[-]
$\overline{K}$	amplitude ratio of K	[-]
L	total length of main line (from pump exit to termination)	[m]
l_1	length of upstream line (from pump exit to silencer)	[m]
l_2	length of downstream line (from silencer to termination)	[m]
P	pressure pulsation	[Pa]
Q	flow pulsation	[m^3/s]
Q_S	pump source flow pulsation	[m^3/s]
r	inner radius of tube	[m]
s	Laplace operator	[1/s]
[**T**]	transfer matrix of silencer	
TL	transmission loss	[-]
x	insertion position of silencer from pump exit (= l_1)	[m]
Z_C	characteristic impedance of tube	[Ns/m^5]
Z_e	resultant (parallel) impedance of main line at junction of silencer	[Ns/m^5]
Z_L	impedance of downstream line to load at silencer outlet	[Ns/m^5]
Z_p	impedance of upstream line to source at silencer inlet	[Ns/m^5]
Z_r	entry impedance of branch type silencer	[Ns/m^5]
Z_S	pump source impedance	[Ns/m^5]
Z_T	termination impedance	[Ns/m^5]
β	wave propagation coefficient	[rad/m]
ρ	density of fluid	[kg/m^3]
θ	phase difference of K	[rad]
ν	kinematic viscosity of fluid	[m^2/s]

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