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Flow-induced pulsations in pipe systems with closed side branches: study of the effectiveness of detuning as remedial measure

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ABSTRACT

Flow-induced pulsations in resonant pipe systems with two closed side branches in cross configuration are considered. These pulsations, commonly observed in many technical applications, are self-sustained aeroacoustic oscillations driven by the instability of the flow along the closed branches. Detuning of the acoustic resonator is often considered as a possible remedial measure. Although this countermeasure appears to be very effective for double side branch systems in cross configuration with anechoic boundary conditions of the main pipe, its effectiveness has not been assessed for different boundary conditions. The significance of the acoustic boundary conditions of the main pipe has been studied by means of experiments conduced on double side branch systems presenting two acoustically reflecting boundaries of the main pipe. While pulsations are often a nuisance, the double side branch system can also be used as a powerful sound source.

INTRODUCTION

Flow-induced pulsations in pipe systems with closed branches have been observed in many technical applications [1-12]. These pulsations can appear in new pipe systems or in existing systems when the operational conditions are modified. Pulsations in pipe networks are undesirable because of the high levels of noise produced, the possibility of fatigue failure and the additional pressure losses.

The main source of oscillations in a gas transport network is the unstable shear layer that forms when the flow passes a pipe discontinuity, such as a closed side branch opening. Self-sustained oscillations of an unstable shear layer are due to a feedback mechanism between the shear layer instability, that acts as amplifier, and an acoustic mode, that acts as filter.

Acoustic resonance of a pipe network, or a part of it, occurs when the acoustic energy accumulates into a standing wave which is called an acoustic mode of the system. The resonance behavior of a pipe system depends on its geometry and on the acoustic boundary conditions at its terminations. The resonance modes involving the whole pipe network, called global modes [13, 14], depend strongly on the acoustic boundary conditions at the system terminations. Differently, the resonance modes involving only a part of the network, called trapped (or localized) modes [13, 14], do not depend on these boundary conditions.

Self-sustained oscillations involving a trapped mode are able to produce very high levels of pulsations, since the trapped mode does not present radiation losses. This condition, frequently encountered in technical applications, is particularly problematic because of the severe vibration levels. Systems with multiple closed side branches of equal length are likely to display trapped modes [15].

A common design rule for avoiding flow-induced pulsations in pipe networks containing closed side branches is to detune the length of the branches, which should avoid trapped modes. Even though this countermeasure has been proved to be very successful in double side branch systems with anechoic boundary conditions of the main pipe, its effectiveness has not been assessed for different acoustic boundary conditions. In the present paper we study the effectiveness of the length-detuning in a double side branch system in cross configuration with reflective boundary conditions at its terminations.

Flow-induced pulsations have been mainly studied because of the technical problems related to them. Even if these pulsations are often undesirable, a double side branch system in cross configuration can also be used as a powerful sound source. In the present paper we consider also this aspect of the phenomena.

EXPERIMENTAL SETUP

General setup

The experimental setup is sketched in Figure 1. The main pipe of the side branch system has an internal diameter $D_{mp} = 33$ mm that is equal to the internal diameter of the side branches $D_{sb} = 33$ mm. The other geometrical characteristics depend on the configuration tested and they are presented in Table 1. In the calculation of the Helmholtz number (dimensionless frequency) the length of the upstream main pipe L_{in} and of the downstream main pipe L_{out} will be corrected by including the end corrections $\delta_{in} = \delta_{out} = D_{mp}/3 + D_{sb}/2$ [16]. The junctions between the side branches and the main pipe present all rounded edges whose radius of curvature is $r_{edge} = 0.1D_{sb}$. The closed terminations of the side branches are equipped with two flush mounted microphones (PT1 and PT2). For some configurations, other two microphones are flush mounted in the wall of the main pipe (PT3 and PT4). The location of these four microphones is specified in Table 1.

The outlet of the side branch system is open to the laboratory (a large room of 15 m $\times$ 4 m $\times$ 4 m) and presents sharp edges; it is, acoustically, an unflanged open pipe termination. The inlet is connected to a high pressure air supply system. It consists of

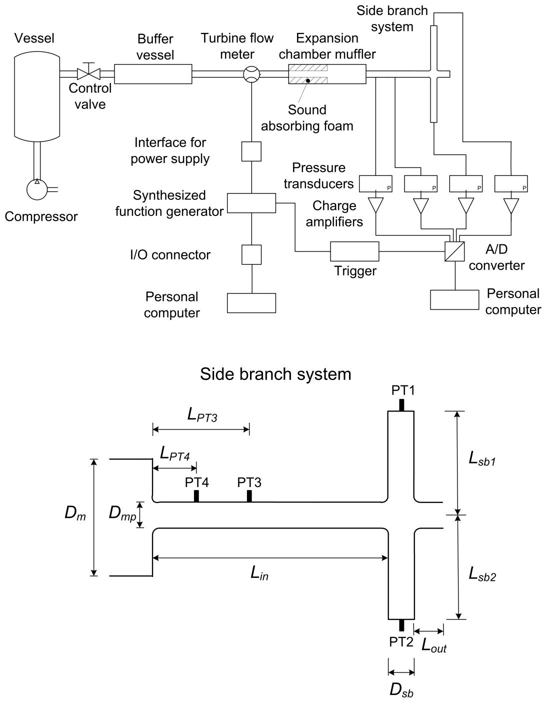

Figure 1: Experimental setup.

a compressor, a 3 m^3 vessel filled with air at 15 bar, a control valve to regulate the flow through the system, an intermediate cylindrical buffer vessel of 1.23 m length and 200 mm diameter, a 2.3 m long pipe of 100 mm diameter, a turbine flow meter, a 1.1 m long pipe of 100 mm diameter and an expansion chamber muffler with diameter $D_m = 150$ mm and length $L_m = 930$ mm. Half of the muffler is internally covered with sound absorbing (open cell) foam in order to avoid cavity resonances. The side branch system is connected to the muffler by a smooth contraction with a radius of curvature of $D_{mp}/3$, which avoids flow separation at the main pipe inlet.

Table 1: Geometrical characteristics of the side branch configurations tested. All the dimensions are in mm.

Configuration	L_{in}	L_{out}	L_{sb1}	L_{sb2}	L_{PT3}	L_{PT4}
Conf-c1	77	66	412	412	-	-
Conf-c2	77	66	339	339	-	-
Conf-c3	77	66	309	309	-	-
Conf-c4	77	66	236	236	-	-
Conf-c5	77	66	221	221	-	-
Conf-c6	77	66	150	150	-	-
Conf-c7	77	66	133	133	-	-
Conf-c8	77	66	119	119	-	-
Conf-da1	592	66	153	153	197	94
Conf-da2	592	66	143	153	197	94
Conf-da3	592	66	133	153	197	94
Conf-da4	592	66	123	153	197	94
Conf-da5	592	66	113	153	197	94
Conf-db1	489	66	153	153	197	94

Instrumentation

The microphones used are piezo-electric pressure transducers *PCB 116A*. They are connected to charge amplifiers *Kistler 5011*, which are, in turn, connected to a personal computer via an A/D converter acquisition board *National Instruments NI SCXI-1314*. The turbine flow meter *Instromet SM-RI-X-K G250* is used to measure the main flow velocity. It is connected to a personal computer via an interface designed in our group for the power supply of the flow meter sensor, a synthesizer function generator *Yokogawa FG120* and a I/O connector *National Instruments NI SCB-68*. The synthesizer function generator and the A/D converter acquisition board are linked via a trigger board *National Instruments NI SCXI-1180*, so that the acquisition of the pressure and the flow velocity are synchronized. The air temperature is measured within $0.1°C$ by means of a digital thermometer *Eurotherm 91e*. Its sensor is positioned inside the expansion chamber muffler.

FLOW-INDUCED PULSATIONS

Moderate amplitude flow-induced pulsations are observed in the configuration *Conf-c1* when air is blown through this pipe system. In Figure 2 the measured dimensionless pulsation frequency $He_L = fL_{sb}/c_0$ and dimensionless pulsation amplitude $|p'|/(\rho_0 c_0 U)$ are presented as function of the Mach number $M = U/c_0$ of the main flow. These measurements have been conducted by monotonically increasing the flow velocity U. The dimensionless frequency corresponds to the Helmholtz number based on the pulsation frequency *f*, the length of the side branches L_{sb} and the speed of sound c_0. The dimensionless pulsation amplitude corresponds to the pressure amplitude $|p'|$ divided by the product $\rho_0 c_0 U$ of the characteristic impedance $\rho_0 c_0$ of the fluid (ρ_0 is the fluid density) with the main flow velocity U. In the particular case of a resonant closed branch, the dimensionless pulsation amplitude at the closed branch termination (i.e. a pressure antinode in the closed branch resonator) corresponds to the ratio $|\vec{u}'_{jun}|/U$ of the acoustic velocity amplitude $|\vec{u}'_{jun}|$ at the sound source with the steady main flow velocity U.

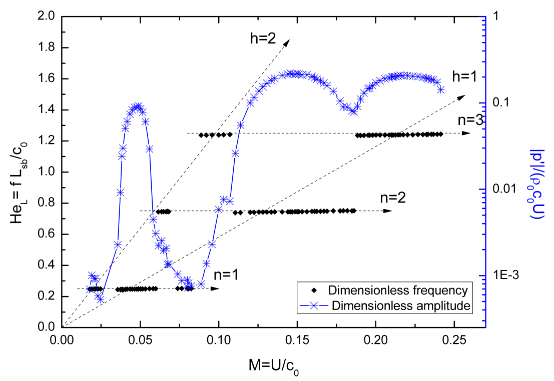

Figure 2: Dimensionless pulsation frequency $He_L = fL_{sb}/c_0$ and dimensionless pulsation amplitude $|p'|/(\rho_0 c_0 U)$ as function of the Mach number $M = U/c_0$ of the main flow. The frequency *f* and the amplitude $|p'|$ are measured by means of the pressure transducer *PT1* (Figure 1). Configuration *Conf-c1*.

By varying the main flow velocity U, different resonant modes are excited, corresponding to standing waves in the side branches with frequencies $f_n \approx (2n-1)c_0/(4L_{sb})$, $n = 1, 2, 3, \ldots$. All these modes present a pressure node at the junction, so that they do not generate plane waves in the main pipe. For frequencies below the cut-off frequency $f_{cut} \approx 6$ kHz for propagation of non-planar modes in the pipe system, these acoustic modes do not radiate into the main pipe, they are trapped (or localized) modes.

The instability of the shear layer separating the main flow in the main pipe to the stagnant fluid in the closed side branch is the source of pulsations in the side branch systems. The time-averaged acoustic source power, which is the power transferred from the main flow to the acoustic flow, can be estimated using the low Mach number $M << 1$ approximation as proposed by Howe [17]:

$$\langle P_{source} \rangle = -\rho_0 \left\langle \int_V (\vec{\omega} \times \vec{u}) \cdot \vec{u}' dV \right\rangle \qquad (1)$$

where $\vec{u}$ is the local fluid velocity, $\vec{\omega} = \nabla \times \vec{u}$ is the vorticity, $\vec{u}'$ is the acoustic velocity, V is the volume in which $\vec{\omega}$ is not vanishing and the brackets $\langle \ldots \rangle$ indicate the time averaging.

The maximum acoustic velocity of the modes observed in double side branch systems in cross configuration, like *Conf-c1/c8*, occurs at the junction between the closed side branch and the main pipe. Since the shear layer instabilities (vortices) form at the junction and are also convected by the main flow in a direction which is almost normal to the acoustic velocity, the acoustic source power, according to Eq. (1), is at its maximum for double side branch systems in cross configuration. The peculiar characteristics of negligible radiation losses and efficient sound production make the well-tuned double side branch system very liable to strong flow-excited acoustic resonances.

The self-sustained oscillations of configuration *Conf-c1* occur at each acoustic mode $n = 1, 2, 3, \ldots$ within certain ranges of flow velocity, which correspond to the hydrodynamic modes of the shear layers $h = 1, 2, \ldots$. The hydrodynamic mode number h indicates the number of vortices, formed by the shear layer, present at the same time in each side branch opening. The ranges of flow velocity are related to ranges of Strouhal number $Sr_{Weff} = fW_{eff}/U$ based on the effective cavity width $W_{eff} = \frac{\pi}{4} D_{sb} + r_{up}$, where r_{up} is the radius of curvature of the upstream edge of the junction [18]. Within each of these intervals, the Strouhal number at which the dimensionless pulsation amplitude displays a maximum is referred to as the optimal Strouhal number $Sr_{Weff,opt}$.

The source of sound is most effective when it operates at the first hydrodynamic mode $h = 1$ (Figure 2). Increasing gradually the flow velocity, higher order hydrodynamic modes $h > 1$ are observed before the first hydrodynamic mode $h = 1$. This feature is particularly clear in Figure 2: each acoustic mode n is first excited by the second hydrodynamic mode $h = 2$ and then by the first hydrodynamic mode $h = 1$.

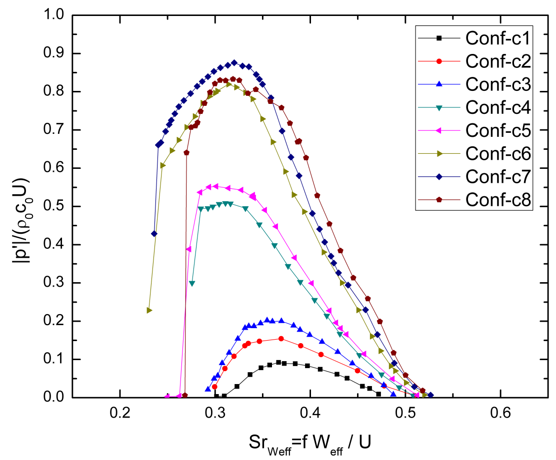

Figure 3: Influence of visco-thermal damping on the dimensionless pulsation amplitude $|p'|/(\rho_0 c_0 U)$ of the quarter wave length resonance $He_L = fL_{sb}/c_0 \approx 0.25$. The amplitude $|p'|$ is measured by means of the pressure transducer *PT1* (Figure 1). Configurations *Conf-c1/c8*.

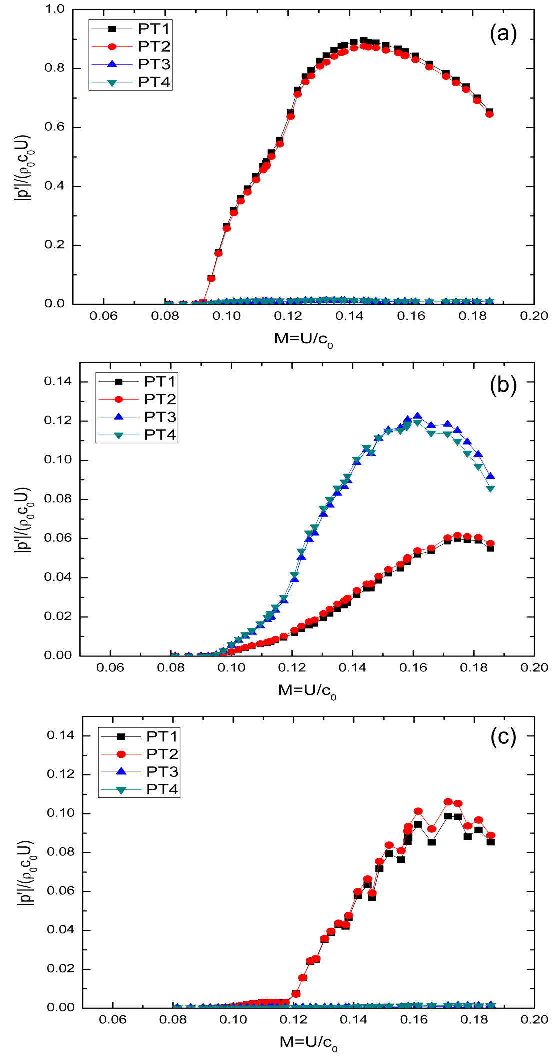

Figure 4: Dimensionless pulsation amplitude $|p'|/(\rho_0 c_0 U)$ as function of the Mach number $M = U/c_0$ of the main flow of: (a) the fundamental frequency $He_L = fL_{sb}/c_0 \approx 0.25$, (b) the second higher (non-trapped) harmonic $He_L \approx 0.5$ and (c) the third higher (trapped) harmonic $He_L \approx 0.75$. Configuration *Conf-da1*. The acoustic pressure $|p'|$ is measured by means of the pressure transducers at the side branch terminations (*PT1* and *PT2*) and along the main pipe (*PT3* and *PT4*).

EFFECT OF VISCO-THERMAL LOSSES AND NON-LINEARITIES ON THE PULSATION BEHAVIOR

The effect of the length of the side branches on the pulsation behavior of the double side branch system in cross configuration has been studied by decreasing the length of the side branches of configuration *Conf-c1* and focusing on the pulsations occurring at the first hydrodynamic and first acoustic mode $h = n = 1$. This condition, corresponding to one vortex in the side branch opening $h = 1$ and to a quarter wave length resonance $f \approx c_0/(4L_{sb})$ of each side branch, has been observed to lead to the highest pulsations in configuration *Conf-c1*. The experiments with different length of the side branches L_{sb} (*Conf-c1/c8*), summarized in Figure 3, show an overall increase of the pulsation amplitude with decreasing the side branch length. This increase in amplitude is due to the decrease of visco-thermal losses in the side branches with decreasing the side branch length. The inverse proportionality between the pulsation amplitude and the side branch length is not observed for the shortest configurations

Conf-c7/c8. This is due to the saturation of the sources of sound and to the radiation losses caused by the generation of higher harmonics [18-20] that appear at very-high amplitudes.

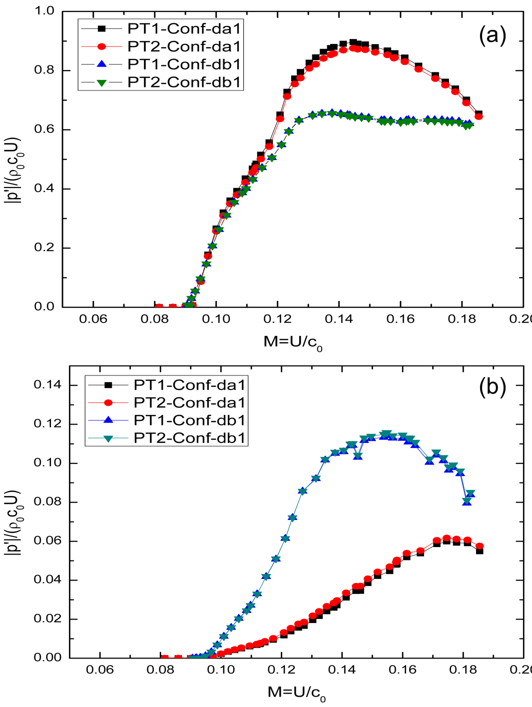

Figure 5: Dimensionless pulsation amplitude $|p'|/(\rho_0 c_0 U)$ as function of the Mach number $M = U/c_0$ of the main flow of: (a) the fundamental frequency $He_L = fL_{sb}/c_0 \approx 0.25$ and (b) the second higher (non-trapped) harmonic $He_L \approx 0.5$. Configurations *Conf-da1* and *Conf-db1*. The acoustic pressure $|p'|$ is measured by means of the pressure transducers at the side branch terminations (*PT1* and *PT2*) and along the main pipe (*PT3* and *PT4*).

Measurements of the acoustic pressure in the resonator (microphones *PT1* and *PT2*) and in the main pipe (microphones *PT3* and *PT4*) carried out in configuration *Conf-da1* show the effect of the non-linearities on the aeroacoustic behavior of a double side branch system in cross configuration (Figure 4). When the system oscillates at the fundamental frequency f_0, that is the quarter wavelength resonance $He_L \approx 0.25$, the non-linearities of the system generates higher order harmonics. These are integer multiples of the fundamental frequency f_0. The odd multiples are resonating harmonics that are, together with the fundamental frequency, trapped in the resonator. The even multiples are non-resonating harmonics that are radiated into the main pipe. The causes for the generation of higher harmonics are the non-linear aeroacoustic sources and the non-linear wave steepening of the high amplitude fundamental frequency f_0 in the side branches [19, 20]. At very-high amplitudes, the generation of higher harmonics by non-linearities of the system is an important mechanism of acoustic energy loss.

A slight modification of the length of the upstream main pipe, from $L_{in} = 592$ mm of *Conf-da1* to $L_{in} = 489$ mm of *Conf-db1*, leads to a surprising result. Both configurations *Conf-da1* and *Conf-db1* display a trapped mode, localized in the double side branch resonator. Since the geometrical dimensions of the resonator are identical in the two configurations, we expect to find the same aeroacoustic behavior. However, configuration *Conf-db1* shows lower amplitudes of the fundamental oscillation frequency than configuration *Conf-da1* (Figure 5-a). This surprising result can be understood by observing the pulsation behavior of the second higher (non-trapped) harmonic $He_L \approx 0.5$, presented in Figure 5-b. In the case of configuration *Conf-da1* the amplitude of this second harmonic is 7% of the amplitude of the fundamental, while in configuration *Conf-db1* it is 20% of the amplitude of the fundamental. The radiation losses due to the non-linear wave steepening are higher for configuration *Conf-db1* than for configuration *Conf-da1* and this results in a significant decrease of pulsation amplitude. Since the only distinction between the two configurations is the difference in the length of the upstream main pipe, the acoustic properties of this section are responsible for the increased transfer of energy by non-linear radiation losses. In particular we expect that in *Conf-db1* this is promoted by the fact that the second higher harmonic $He_L \approx 0.5$ has a frequency that matches the resonance condition of the system composed by the two side branches and the inlet main pipe: $(L_{sb} + L_{in} + \delta_{in})/\lambda \approx 2.16$.

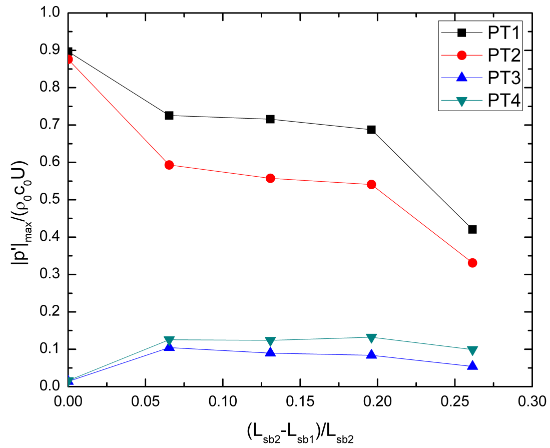

Figure 6: Maximum dimensionless pulsation amplitude $|p'|_{max}/(\rho_0 c_0 U)$ as function of the length-detuning $(L_{sb2} - L_{sb1})/L_{sb2}$ of the fundamental frequency, that is the quarter wave length resonance $He_L = fL_{sb}/c_0 \approx 0.25$. Configurations *Conf-da1/da5*. The acoustic pressure $|p'|$ is measured by means of the pressure transducers at the side branch terminations (*PT1* and *PT2*) and along the main pipe (*PT3* and *PT4*).

Table 2: Helmholtz number $He_L = fL/c_0$ of the fundamental pulsation frequency at optimal whistling (maximum of the dimensionless pulsation amplitude) of configurations *Conf-da1/da5* based on different lengths L. These are the length of each side branch L_{sb1} and L_{sb2}, half of the length of the entire double side branch resonator $(L_{sb2} - L_{sb1})/2$ and the length of the upstream section of the main pipe L_{in}.

	$He_{L_{sb1}}$	$He_{L_{sb2}}$	$He_{L_{sb1-sb2}}$	$He_{L_{in}}$
Conf-da1	0.245	0.245	0.245	0.99
Conf-da2	0.239	0.255	0.247	1.03
Conf-da3	0.232	0.267	0.250	1.08
Conf-da4	0.225	0.279	0.252	1.13
Conf-da5	0.214	0.289	0.251	1.17

DETUNING THE SIDE BRANCH LENGTH

A proposed [21, 22] remedial measure for the reduction of the pulsation intensity in double side branch systems in cross configuration consists of detuning the length of the side branches. The introduction of an asymmetry in the side branch geometry leads to a pressure node of the resonating acoustic modes that shifts from the position along the main pipe axis. This promotes the radiation of acoustic power by generation of plane waves in the main pipe. Although this countermeasure can be very effective for a double side branch system in cross configuration with anechoic boundary conditions of the main pipe [21, 22], its effectiveness has not been assessed for different boundary conditions. There are examples [16] in which the acoustic properties of the pipes upstream and downstream of a closed branch system strongly influence the aeroacoustic behavior of the system.

The significance of the acoustic boundary conditions of the main pipe on the effectiveness of detuning is assessed in our work by means of experiments (Figure 6) on configurations presenting two acoustically reflecting boundaries of the main pipe, *Conf-da1/da5*. In presence of anechoic boundaries, a 10% of detuning of the side branch length was sufficient to decrease by an order of magnitude the pulsation amplitude [21, 22]. In our system with reflective boundaries, even a change of 30% in length of one of the side branches is not sufficient to reduce by more than a factor 3 the magnitude of the pulsations in the side branches.

The asymmetry in the side branch geometry, as described above, promotes the radiation of acoustic power by generation of plane waves in the main pipe. In a system with anechoic boundary conditions these acoustic waves are radiated away, so that the radiation of sound in the main pipe results in a considerable damping. If the system is bounded by two acoustically reflective boundaries, the energy radiated from the side branch resonator into the main pipe is not entirely radiated away. If the acoustic pulsations have a frequency matching a global resonance condition of the pipe system, detuning the length of the side branches does not reduce the pulsation amplitude. As can be observed in Figure 6 and Table 2, up to a length-detuning of $(L_{sb2} - L_{sb1})/L_{sb2} \approx 0.2$, a reduction of the pulsation amplitude is avoided by the occurrence of a resonance involving the side branch resonator and the upstream main pipe segment.

DOUBLE SIDE BRANCH SYSTEM AS SOURCE OF SOUND

Experimental setups for the testing of the acoustic response of structures usually implement sirens and loudspeakers. An alternative is to use a double side branch system in cross configuration. This system can be used for the generation and radiation of its second higher harmonic, in the symmetric configuration, and of its fundamental frequency, in the asymmetric configuration. By using two adjustable pistons as side branch terminations, in order to vary independently the length of the side branches, will produce a sound generator that is able to produce and radiate a wide range of frequencies with a wide range of acoustic amplitudes.

CONCLUSIONS

Double side branch systems in cross configuration display trapped acoustic modes that can lead to high levels of self-sustained pulsations. The amplitude of these pulsations increases as the length of the side branches decreases, due to the decrease in visco-thermal losses.

At very high pulsation levels $|\vec{u}'_{jun}|/U \approx 1$, the pulsation amplitude saturates. In trapped modes, non-linear generation of higher harmonics results into the radiation of sound.

Detuning a double side branch systems in cross configuration is an efficient countermeasure to avoid pulsations for systems with anechoic boundary conditions of the main pipe. However, it is a less effective remedial measure for systems with acoustically reflective boundaries of the main pipe. Since the acoustic properties of the pipe systems play a significant role, scale models aiming at the prediction of pulsations should have realistic acoustic boundary conditions.

A double side branch system in cross configuration can be used as a powerful sound source, able to produce and radiate a wide range of frequencies with a wide range of acoustic amplitudes.

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REFERENCES

[1] Y.N. Chen and R. Stürchler, Flow-induced vibrations and noise in the pipe system with blind branches due to coupling of vortex shedding, *Internoise 77*, Zurich, Switzerland, 1977.

[2] J.T. Coffman and M.D. Bernstein, Failure of safety valves due to flow induced vibration, *Journal of Pressure Vessel Technology* 102 (1980) 112-118.

[3] R.M. Baldwin and H.R. Simmons, Flow-induced vibration in safety relief valves, *Journal of Pressure Vessel Technology* 108 (1986) 267-272.

[4] J. Gorter, A. Hirschberg, A. Wijnands and J.C. Bruggeman, Flow induced pulsations in gas transport systems, *International Gas Research Conference*, Tokyo, Japan, 1989.

[5] W.M. Jungowski, K.K. Botros and W. Studzinski, Cylindrical side-branch as tone generator, *Journal of Sound and Vibration* 131 (1989) 265-285.

[6] M.C.A.M. Peters and H.J. Riezebos, Analysis of the occurrence of flow-induced pulsations in a gas control station, *International Gas Research Conference*, Amsterdam, The Netherlands, 2001.

[7] P. Lafon, S. Caillaud, J.P. Devos and C. Lambert, Aeroacoustical coupling in a ducted shallow cavity and fluid/structure effects on a steam line, *Journal of Fluids and Structures* 18 (2003) 695-713.

[8] S.A. Hambric, T.M. Mulcahy, V.N. Shah, T. Scarbrough and J. Wu, Acoustic loading on BWR steam dryers caused by valve singing, *9th NRC/ASME Symposium on Valves, Pumps and Inservice Testing*, Washington, D.C., United States, 2006.

[9] G. DeBoo, K. Ramsden, R. Gesior and B. Strub, Identification of Quad Cities main steam line acoustic sources and vibration reduction, *ASME Pressure Vessels and Piping Division Conference*, San Antonio, Texas, United States, 2007.

[10] S. Takahashi, M. Ohtsuka, K. Okuyama, T. Ito and K. Yoshikawa, Experimental study of acoustic and flow-induced vibrations in BWR main steam lines and steam dryers, *ASME Pressure Vessels and Piping Division Conference*, Chicago, Illinois, United States, 2008.

[11] S. Ziada, Flow-excited acoustic resonance in industry, *Journal of Pressure Vessel Technology* 132 (2010) 1-9.

[12] NRC, Additional flow-induced vibration failures after a recent power uprate, *NRC Information Notice 2002-26, Supplement 2, January 9, 2004, US Nuclear Regulatory Commission*, Washington, D.C., United States, 2004.

[13] Y. Duan, W. Koch, C.M. Linton and M. McIver, Complex resonances and trapped modes in ducted domains, *Journal of Fluid Mechanics* 571 (2007) 119-147.

[14] S. Hein and W. Koch, Acoustic resonances and trapped modes in pipes and tunnels, *Journal of Fluid Mechanics* 605 (2008) 401-428.

[15] D. Tonon, G. Nakiboglu, J.F.H. Willems, A. Hirschberg, R.E. Leandro, W. Polifke and H.J. Riezebos, Self-sustained aeroacoustic oscillations in multiple side branch pipe systems, *15th AIAA/CEAS Aeroacoustics Conference*, Miami, Florida, United States, 2009.

[16] D. Tonon, A. Hirschberg, J. Golliard and S. Ziada, Aeroacoustics of pipe systems with closed branches, *Submitted to the International Journal of Aeroacoustics*, 2010.

[17] M.S. Howe, The dissipation of sound at an edge, *Journal of Sound and Vibration* 70 (1980) 407-411.

[18] J.C. Bruggeman, A. Hirschberg, M.E.H. van Dongen, A.P.J. Wijnands and J. Gorter, Self-sustained aero-acoustic pulsations in gas transport systems: experimental study of the influence of closed side branches, *Journal of Sound and Vibration* 150 (1991) 371-393.

[19] P.C. Kriesels, M.C.A.M. Peters, A. Hirschberg, A.P.J. Wijnands, A. Iafrati, G. Riccardi, R. Piva and J.C. Bruggeman, High amplitude vortex-induced pulsations in a gas transport system, *Journal of Sound and Vibration* 184 (1995) 343-368.

[20] S. Dequand, S.J. Hulshoff and A. Hirschberg, Self-sustained oscillations in a closed side branch system, *Journal of Sound and Vibration* 265 (2003) 359-386.

[21] H.R. Graf and S. Ziada, Flow induced acoustic resonance in closed side branches: an experimental determination of the excitation source, *International symposium on Flow-Induced Vibration and Noise*, Anaheim, California, United states, 1992.

[22] H.R. Graf and S. Ziada, Excitation source of a side-branch shear layer, *Submitted to the Journal of Sound and Vibration*, 2010.