EXPERIMENTAL ANALYSIS OF THE COUPLING MECHANISM BETWEEN FLOW AND ACOUSTIC FIELDS IN A DUCT WITH RESONATORS

Thiago C. Souza, e-mail: thiagosp@lva.ufsc.br
Arcanjo Lenzi, e-mail: arcanjo@lva.ufsc.br
LVA - Laboratório de Vibrações e Acústica
Departamento de Engenharia Mecânica
Universidade Federal de Santa Catarina
88040-900, Florianópolis, SC, Brazil

Cesar J. Deschamps, e-mail: deschamps@polo.ufsc.br
POLO - Laboratórios de Pesquisa em Refrigeração e Termofísica
Departamento de Engenharia Mecânica
Universidade Federal de Santa Catarina
88040-900, Florianópolis, SC, Brazil

Abstract. *The present paper reports an experimental analysis of phenomena associated with the coupling mechanism between flow and acoustic fields in ducts provided with resonators. Such a coupling can be explained by three effects. The first one is associated with the dissipation of acoustic modes by the flow in the presence of external acoustic sources. The second aspect is linked to self-sustained oscillations originated by the interaction between the flow and the acoustic field in the resonator. Finally, the irradiated noise from the jet exiting the duct is also relevant and has to be considered in this situation. The main motivation for the present study is related to the design of mufflers of reciprocating compressors, in which it is fundamental to understand how the flow field affects the performance of resonators.*

Keywords*: Aeroacoustics, Self-sustained flow oscillations, Mufflers, T-shaped Resonators, Linear Acoustics*

1. INTRODUCTION

Acoustics filters are devices widely employed in reciprocating compressors, but in some situations their application may not be appropriate due to resonances of the acoustic system. In such cases, the performance of these devices can be improved by applying T-shaped resonators to suppress critical frequencies. These side branch resonators are acoustic reactive elements of sound absorption that are tunable for multiple or specific frequencies, and can be applied with the purpose of noise control in acoustic filters, such as compressor mufflers. The sound attenuation of resonators is usually provided by abrupt changes in acoustic impedance that are created with their coupling to acoustic geometries.

In most cases, the influence of the flow is not considered and the resonator acoustic characteristics are determined through a linear acoustic approach. It has been shown that for certain situations this methodology provides a robust and convenient method to determine such characteristics (Mareze, 2009). However, some results (Souza, 2009; Radavich *et al.*, 2001) demonstrate that the presence of fluid flow can make the linear acoustic approach inappropriate in some situations, especially if the turbulent flow regime prevails. In fact, when feedback resonances of the closed side branch are assessed, it can be shown that the flow effects may turn a T shaped resonator into a potential noise source.

The main goal of this paper is to experimentally investigate noise attenuation systems of reciprocating compressors that employ a side branch acoustics resonator. To allow a detailed analysis of the flow effects on the acoustic system, measurements of the flow were carried out in the main duct with and without an acoustic resonator.

2. EXPERIMENTAL SET-UP

Three experimental setups were adopted in order to analyze flow effects on resonances associated with a simple duct and a coupled acoustic system. These setups can be systematized in the following purposes:

1. Measurement of the acoustic transfer function for an acoustic system made of a simple duct, with and without a resonator;
2. Measurement of the global external noise system;
3. Measurement of acoustic pressure level at the closed extremity of the T shaped resonator.

In all experiments, a compressor was used to generate the flow in the system, with air being admitted from atmospheric condition. The mean flow velocity was determined from the volumetric flow rate measured by means of a calibrated Micro-Motion, D6 model, Coriolis flow meter, resulting in an uncertainty of about 0.75 m/s for velocity measurements (Souza, 2009).

A schematic view of the test facility used in all experiments is shown in Fig. 1.

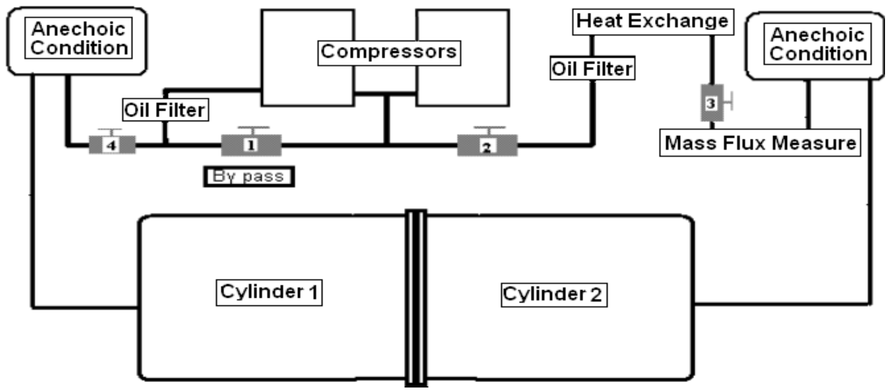


Figure 1. Schematic of the test facility

For the assessment of the transfer function, the experiments were carried out with an external sound source. The idea was to excite the internal resonances of the system through an acoustic field, initially in the absence of fluid flow. Then, tests were conducted for different flow conditions, by adjusting the mass flow rate. In this way, the effect of the flow regime, laminar and turbulent, on the internal acoustic field could be analyzed, with a very prominent effect being observed on the sound energy associated with the internal resonance. The related problem of acoustic noise generated by the flow in the duct was also investigated, but in this case the measurement of the global external noise was performed without using an external sound source.

Experiments in the test facility were performed with air driven to a 70 mm pipe. In some cases, a T-shaped resonator (L_r = 35mm) was applied at the middle of the main duct. As can be seen in Fig. 2, two setups were used for the following goals: a) analysis of the flow effect on the duct resonance with and without a resonator; b) analysis of the noise generated by the flow in the duct with and without a resonator. The purpose of the first analysis was to determine the shifting of the acoustical characteristics originated by the flow, through the assessment of the transfer function and the associated experimental coherence.

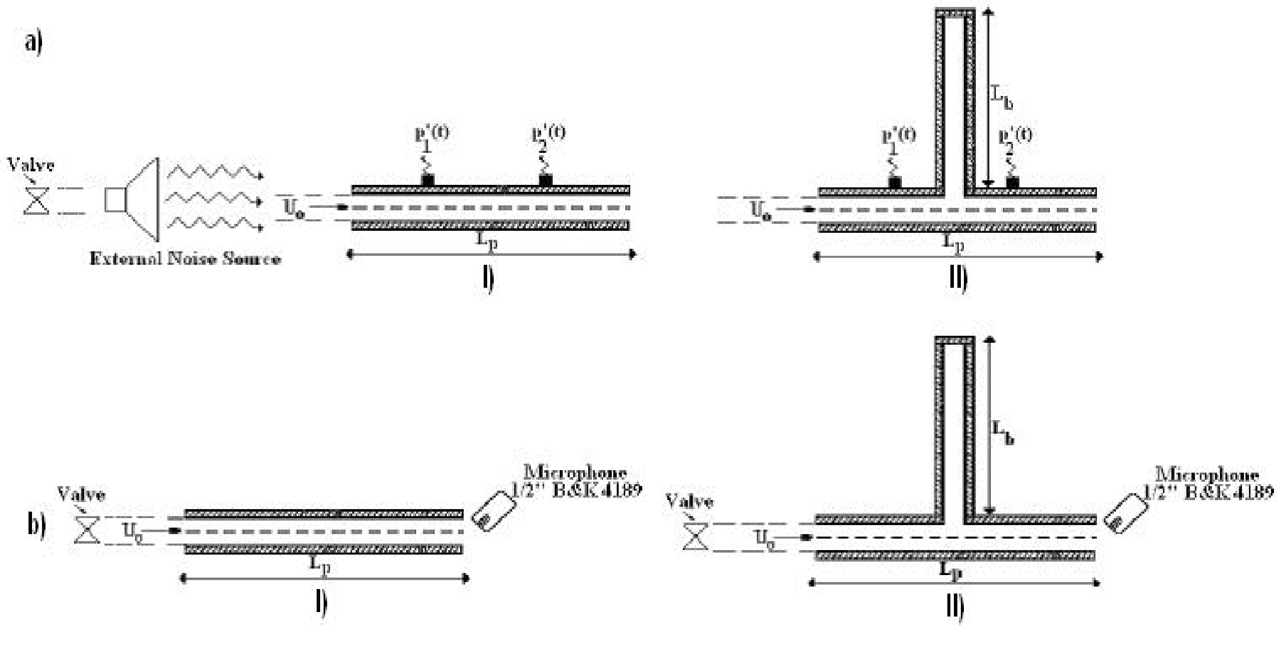


Figure 2. Experimental setups: a) Analysis of the flow effect on the resonance associated with a single duct (I) and a coupled system (II), b) Analysis of flow noise in a single duct (I) and with a coupled system (II).

The setup (a) was used to determine the interaction of resonance with the flow within the duct, considering the attenuation of resonances due to turbulence in the duct for arrangements I and II, as illustrated in Fig. 2a. For this analysis, an external sound source and two ½" B&K 4189 pre-polarized free field microphones were used, as indicated in Fig. 2a. One of the microphones was located upstream and the other downstream of the acoustic system, so as to obtain the transfer function and the coherence associated with the quality measurement. As already pointed out, the effect on the flow noise was determined through setup shown in Fig. 2b, without using an external noise source.

The acoustic pressure fluctuation was also monitored at the closed extremity of the resonator with the objective of observing any *feedback* mechanism of the acoustic field in the cavity. This is a non-linear mechanism (Ziada, 1994) produced by the coupling between the flow at the open extremity and the resonance originated in the resonator.

Pressure oscillations are induced by the flow when the resonator is assembled to the main duct (Souza, 2009). For certain conditions, the resonator may become even a noise source, due to vortex shedding at the leading edge of the cavity, with a subsequent transfer of flow energy to the acoustic field in the cavity. Figure 3 gives a schematic view of experimental setup used to investigate this effect.

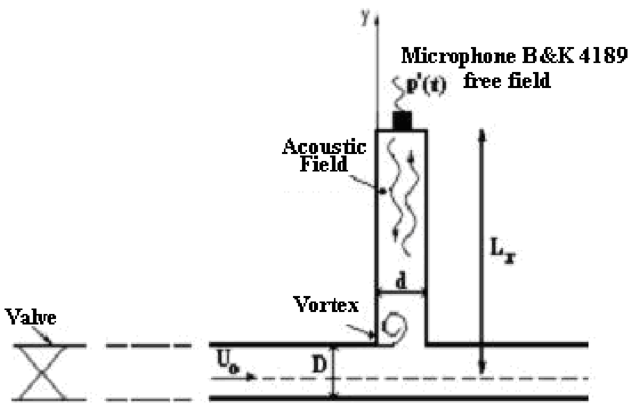


Figure 3. Schematic of the experiment setup for measuring the acoustic field inside the resonator in the presence of flow.

3. RESULTS

Initially, results are presented for a discussion of flow effects on the resonance, considering the experimental setup with an external noise source, but without the resonator. Figure 4 shows a comparison of results for the acoustic transfer function resulting from three situations: duct without flow, duct with laminar flow (Re = 1,335), duct with turbulent flow (Re = 18,371). Additionally, an analytical result based on the linear acoustic theory, not considering the flow, is also presented.

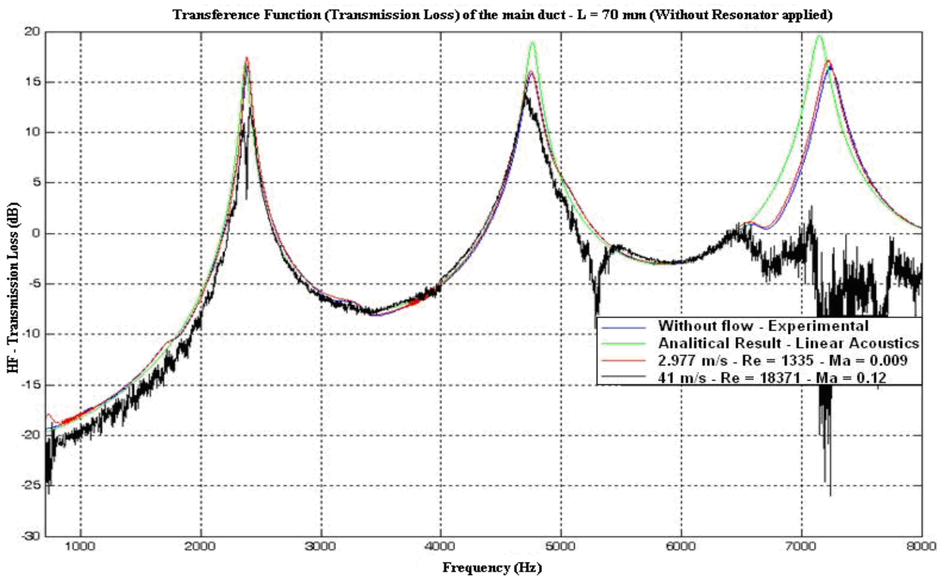


Figure 4. Experimental results for Transfer Function in the main duct without flow, with laminar flow (Re = 1,335) and with turbulent flow (Re = 18,371), compared to the analytical solution without flow.

The modes present in the duct, considering a correction for the open end can be determined by the following expression (Pierce, 1989):

$$f_m = \frac{c_0}{\lambda_m} = m\frac{c_0}{2L'} \tag{1}$$

where m is the mode number related with the natural frequencies of the duct, c_0 is the speed of the sound, and L' is the corrected tube length.

The excitation of resonance modes, m, is very clear even in the presence of flow in the duct. The amplitude of the transfer function for laminar flow, corresponding to U_0=3.0m/s, is very similar to results obtained for the system without flow. On the other hand, it can be seen that for higher mass flow rate, associated with the turbulent regime, an asymmetric distribution and subsequent dissipation of resonances modes appear.

As indicated by Ingard (1975), the additional damping of modes by the turbulent flow is due to the vortex shedding generated by the jet at the duct exit. A hypothesis that could explain the damping of higher modes is the proximity between their wave lengths and the characteristic length of the vortex shed at the duct exit.

The modes of a closed side branch can be determined by the following expression (Pierce, 1989):

$$f_m = \left(\frac{1}{4} + \frac{m}{2_m}\right)\frac{c_0}{L_r'} \tag{2}$$

where m is the mode number related with the corresponding natural frequencies, c_0 is the speed of sound, and L_r' is the corrected length of the resonator tube. Results for the flow effect on the acoustic response of a duct with a resonator are show in Fig. 5.

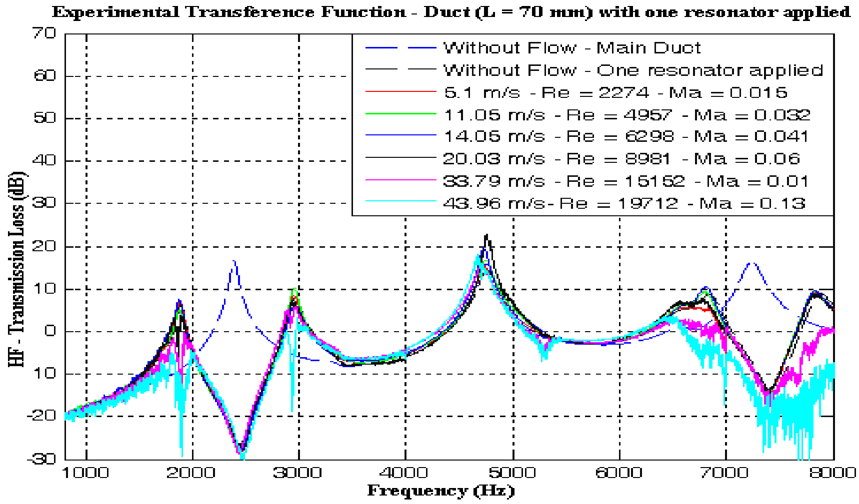


Figure 5. Experimental data for transfer function in the duct with resonator.

The experimental data show a similar behavior for all curves, with a dissipative trend for those situations of higher velocity magnitude, which is a consequence of the intrinsic turbulent flow regime. Transfer functions of tonal frequency components, represented by resonances in Fig. 5, show the dissipative behavior is very significant for certain velocity levels. For instance, in the frequency range 7kHz < f_m < 8kHz the modes are clearly damped when compared with other modes, with the resonance peaks becoming asymmetric due to convection and turbulence dissipation.

With the purpose of examining whether the dissipative effect is caused by the turbulent flow in the main duct, experimental data for coherence was also obtained, since this is a powerful parameter to determine the quality of a given measurement. For the present situation, it is expected that the loss of coherence is a result of turbulence dissipation in the flow. Souza (2009) carried out experiments without flow and observed no loss of coherence, confirming the aforementioned phenomenon.

The next step was to analyze results of noise generated by the flow in the system with and without the resonator. For this situation, no external noise source was applied. An investigation of flow noise in ducts (Ingard *et al.*, 1975) was used as a reference for the present experimental approach. It should be noted that it is difficult to differentiate between the flow noise within the duct and the overall noise at the exit of the duct. Results for global external noise generated by the flow in the main duct, with and without a resonator, are shown in Fig. 6.

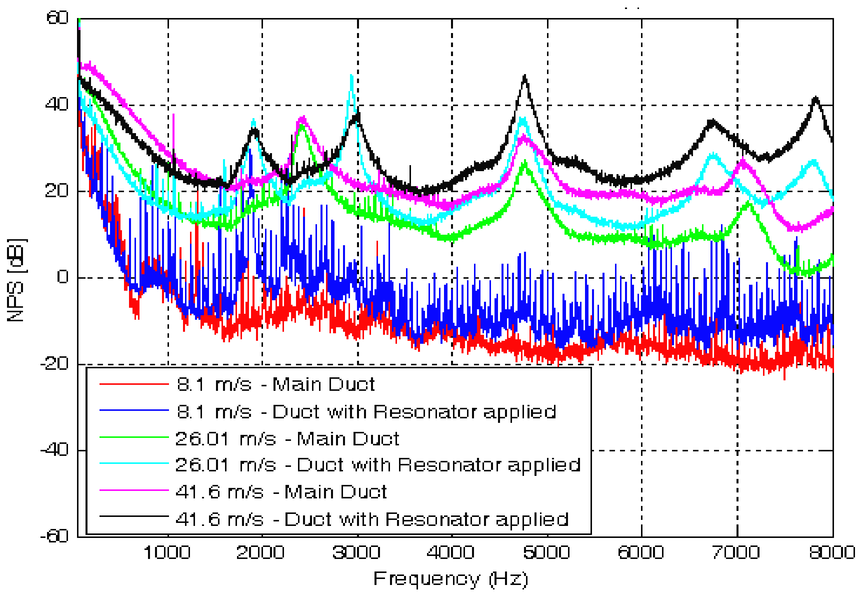


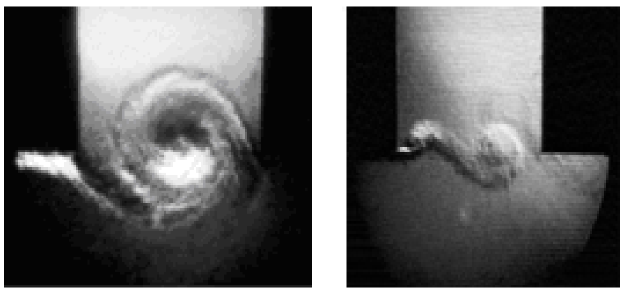
Figure 6. Experimental results for SPL at the duct exit, with and without the presence of a resonator.

Low-frequency noise is generated by the turbulent jet at the duct exit, while contributions at high-frequency are fully due to resonances modes. For the duct without the resonator, a turbulent shear layer occurs only at the duct exit and, therefore, energy loss in low frequency is less significant. However, when a resonator is used the flow separation in the intersection region between the resonator and the main duct causes an additional energy loss. Therefore, this aspect entails a greater loss of energy in the low frequency range when compared with the duct without a resonator. The evidence of this non-linear effect is exposed in Fig. 6. Firstly, the figure shows that significant excitation can only occur for flows with high velocity magnitude. For instance, low velocity levels (U_0=8.1m/s) are seen not to excite internal modes. However, for the resonance frequency close to 3 kHz the sound pressure level (NPS) for U_0=26.0m/s is greater than that for U_0= 41.6m/s, which is result opposite to the one that could be expected.

Considering a mean flow velocity U_0 = 26.0m/s and the resonator diameter d = 3.2mm, the first axial mode of the resonator (f_m = 2400 Hz) corresponds to a Strouhal number S_r = 0.32. This hydrodynamic mode is associated with a single vortex shed in the intersection region. When the time residence of the vortex is less than the acoustic period related with the frequency mode of the closed side branch, two vortices are shed in that region. This situation corresponds to the second hydrodynamic mode (Kriesels *et al.*, 1995).

The decay of the shear layer instabilities in coherent structures of elevated vorticity only depends of the excitation amplitude of the acoustic field in the resonator, as pointed out by Bruggeman (1991). Figure 7, extracted from Dequand *et al.* (2001), provides an insight about such situations.

Results for pressure fluctuation at the closed extremity of the resonator associated with the first and second hydrodynamic modes of the acoustic system are presented in Fig. 8, as a function of the mean flow velocity.



(a) (b)

Figure 7. Schlieren visualization of the flow in the intersection region: (a) First hydrodynamic mode;
(b) Second hydrodynamic mode (Dequand *et al.*, 2001).

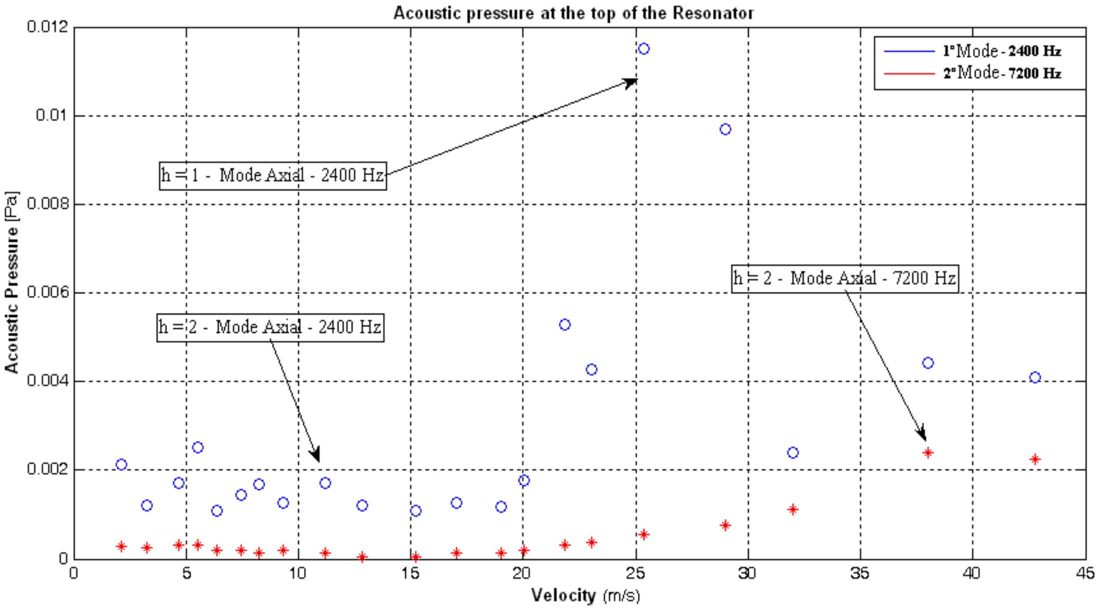


Figure 8. Experimental data for acoustic pressure at the top of the closed side branch.

The acoustic pressure measured at the closed end of the resonator has its highest levels for flow velocities between 20m/s and 30m/s. For the first axial mode, the Strouhal number is found to be in the range $0.3 < S_r < 0.5$, whereas for the second axial mode, 7.2 kHz, the range is changed to $0.5 < S_r < 1$. It can be observed that energy transfer will be smaller for the second mode due to the presence of two vortices in the junction between the duct and the resonator.

In line with the Bruggeman criteria, the results show that the self-sustained amplitude regime corresponds to a low feedback regime (Bruggeman et al., 1991). Therefore, in the present case, the energy transfer by the flow to the acoustic field in the resonator cavity is compromised due to the vorticity diffusion at the junction.

The amplitude of the feedback mechanism found in the present work is smaller than the results provided by Bruggeman *et al.* (1991) and Jungowski *et al.* (1989). This is a consequence of the laboratory test facility, combined with the filter geometric parameters, which was not able to supply Reynolds numbers for the flow as high as those employed in the aforementioned references.

4. CONCLUSION

The present paper considered an experimental analysis of the coupling mechanism between flow and acoustic fields in ducts provided with resonators, which are of great importance in mufflers of reciprocating compressors. The results show that excitation of axial modes by the flow is greater for higher Reynolds number flows. The experimental results also show a dissipation of axial modes promoted by turbulence in the flow field. The effect of an interaction between the turbulent flow and the acoustic field can negatively alter the characteristics of the filter. Hence, the common procedure of adopting a linear acoustic approach to analyze resonators must be revised, if the acoustic system is to operate in the presence of fluid flow. The results obtained in this study show that the flow can excite a third mode in the duct that can not be attenuated by the resonator. Moreover, it has been found that the presence of a resonator can even generate higher sound pressure levels than would occur if no resonator was applied. Another effect related with the flow field is the self-sustained oscillations associated with the coupling between the flow and the acoustic field in the closed side branch acting as a resonator. Previous results show that such interactions can make the acoustic resonator to behave like a noise generator.

5. ACKNOWLEDGEMENTS

The authors thank CAPES and EMBRACO for the financial support, and also Paulo H. Mareze, whose assistance was fundamental for the experimental setup and technical discussion.

6. REFERENCES

Bruggeman, J.C, Hirschberg, A., Van Dongen, M.E.H., Wijnands, A.P.J. Gorter, J., 1991, “Flow Induced Pulsations in Gas Transport Systems: Analysis of the Influence of Closed Side Branches”, J. Fluids Engineering, v. 111, pp. 484-491.

Dequand, S., 2001, "Duct Aeroacoustics: from Technological Applications to the Flute", Ph.D.Thesis, Technische Universiteit Eindhoven (Nl) & Universite du Maine (Fr).

Dequand, S., Hulshoff, S.J., Kuijk, Van H., Willems, J., Hirschberg, 2003, "A Self-Sustained Oscillations in a Closed Side Branch System", J. Sound and Vibration, v. 265, pp. 359-386.

Hirschberg, A., 1997, "Self-Sustained Aeroacoustic Oscillations in Gas Transport Systems: A Prediction Method for Pulsations Induced by Closed Pipe Segments", Technical Report, Technische Universiteit Eindhoven, Gasdynamics Group, Laboratory for Fluid Dynamics, Report R-1428-D.

Howe, M. S., 1998, "Acoustics of Fluid-Structure Interactions". Cambridge University Press.

Ingard, K. U., Singhal, V.K., 1975, "Effect of Flow on the Acoustic Resonance of an Open-Ended Duct", J. Acoustical Society of America, v. 58, pp. 788.

Jungowski, W.M., Botros, K.K., Studzinski, W., 1989, "Cylindrical Side-Branchas Tone Generator", J. Sound and Vibration, v. 131, pp. 265-285.

Kriesels, P.C., Peters, M.C.A.M., Hirschberg, A., Wijnands, A.P.J., Iafrati, A., Riccaradi, G., Piva, R., Bruggeman, J.C., 1995, "High Amplitude Vortex-Induced Pulsations in a Gas Transport System". J. Sound and Vibration, v. 184, pp.343–368.

Mareze, P.H., 2009, "Application of Resonators for Acoustic Attenuation of Cavities and Mufflers", M.Sc. Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianopolis, SC, Brazil (in Portuguese).

Pierce, A.D., 1989, "Acoustics: An Introduction to Its Physical Principles and Applications", Acoustical Society of America, Woodburg, NY.

Rienstra, S.W., Hirschberg, A., 2006, "An Introduction to Acoustics". Eindhoven University of Technology, Eindhoven, Netherlands.

Souza, T.C, Lenzi, A., Deschamps, C.J., 2008, "Dynamics and Acoustics of Flow through Resonator Tubes", Proc. VI Congreso Iberoamericano de Acústica - FIA2008, Buenos Aires (in Portuguese).

Souza, T.C., 2009, "Induced Aeroacoustic Oscillations in Tubes with Resonators", M.Sc. Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianopolis, SC, Brazil.

Ziada, S., 1994, "A Flow Visualization Study of Flow-Acoustic Coupling at the Mouth of a Resonant Side-Branch". J. Fluids and Structures, v. 8, pp. 391-416.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.