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► **To cite this version:**

Thomas Humbert, Yves Aurégan. Loss compensation by flow in a corrugated resonant duct. e-Forum Acusticum 2020, Dec 2020, Lyon, France. pp.777-778, 10.48465/fa.2020.0242 . hal-03229437

HAL Id: hal-03229437

<https://hal.science/hal-03229437>

Submitted on 21 May 2021

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Loss compensation by flow in a corrugated resonant duct

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This oral presentation describes the aeroacoustic behaviour of a corrugated tube placed between two resonators. The sound-flow interactions in the corrugated tube allows controlling the resonant duct damping: loss compensation can be achieved and the onset of whistling is observed and studied.

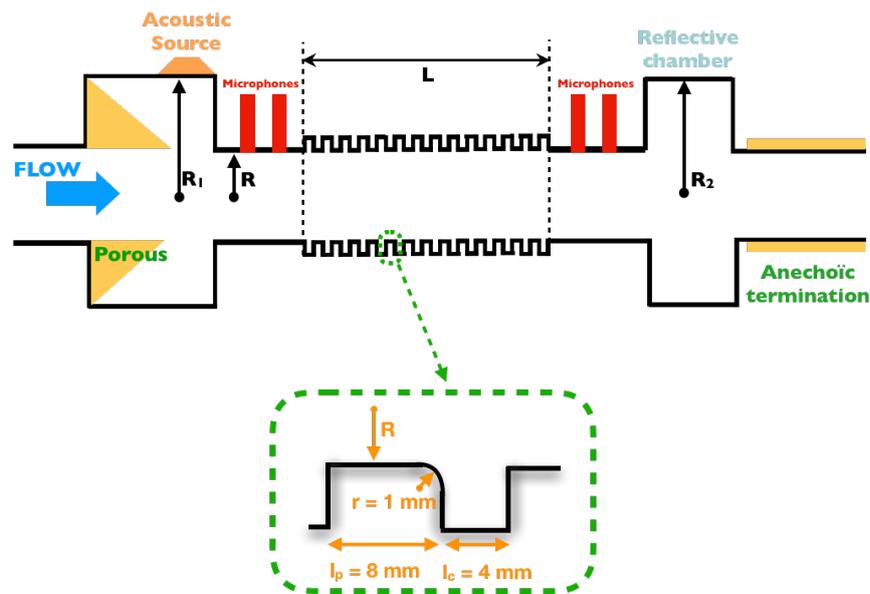


Figure 1: (Top) Sketch of the experimental set-up. A corrugated tube composed of 103 elements is located between two hard-wall tubes and two expansion chambers. The upstream chamber is equipped with an acoustic source and some gradually mounted porous material. (Bottom) Sketch of the corrugated tube unit cell.

The experimental set-up, depicted in Figure 1, consists in a corrugated tube surrounded by two resonators and mounted on a flow rig ended by an anechoic termination. The flow is provided by a centrifugal fan. The corrugated tube is composed of 103 cells and has already been studied in [1]: depending on the mean flow Mach number, some frequencies are amplified and others are damped. Downstream the corrugated tube, a cylindrical resonator of radius $R_2 = 4.7 \text{ cm}$ and length $L_2 = 10.96 \text{ cm}$ is mounted. Upstream the corrugated tube, another cylindrical element of radius $R_1 = 7.5 \text{ cm}$ and length $L_1 = 66.5 \text{ cm}$ is mounted. It is equipped with an acoustic source and some gradually mounted porous material drilled to let the air flow inside. More than 90% of the wave energy coming from the corrugated tube is reflected by the chambers. The measurement sections consist in two microphones separated by a distance $\Delta x = 3 \text{ cm}$. There is one measurement section between the corrugated tube and each reflective chamber. From the upstream pressure ratio, an impedance is computed for the resonant duct.

Figure 2 (left) displays this impedance, as a function of the frequency. Without flow, all the impedance peaks look similar. For $M = 0.016$, the three first impedance peaks are narrower than without flow. For $M = 0.025$, low frequency peaks seem wider, but the other resonances are very narrow.

Figure 2 (right) plots the damping rates $\eta = \delta f / 2f_0$ associated with each resonance, where f_0 is the peak frequency and δf its width at -3dB. Without flow, damping rates are nearly constant, recalling that damping, mainly caused by viscothermal losses, increases linearly with frequency. The presence of the flow changes completely the picture. At low frequencies, damping is reduced for the slowest flow, whereas it is increased for the fastest flow. The most interesting result is that the modal damping reaches nearly 0 for the modes at 762 Hz and 837 Hz when $M = 0.025$. It means that the amplification induced by sound-flow interactions compensates the losses coming from propagation effects and non-perfect reflection at the boundaries.

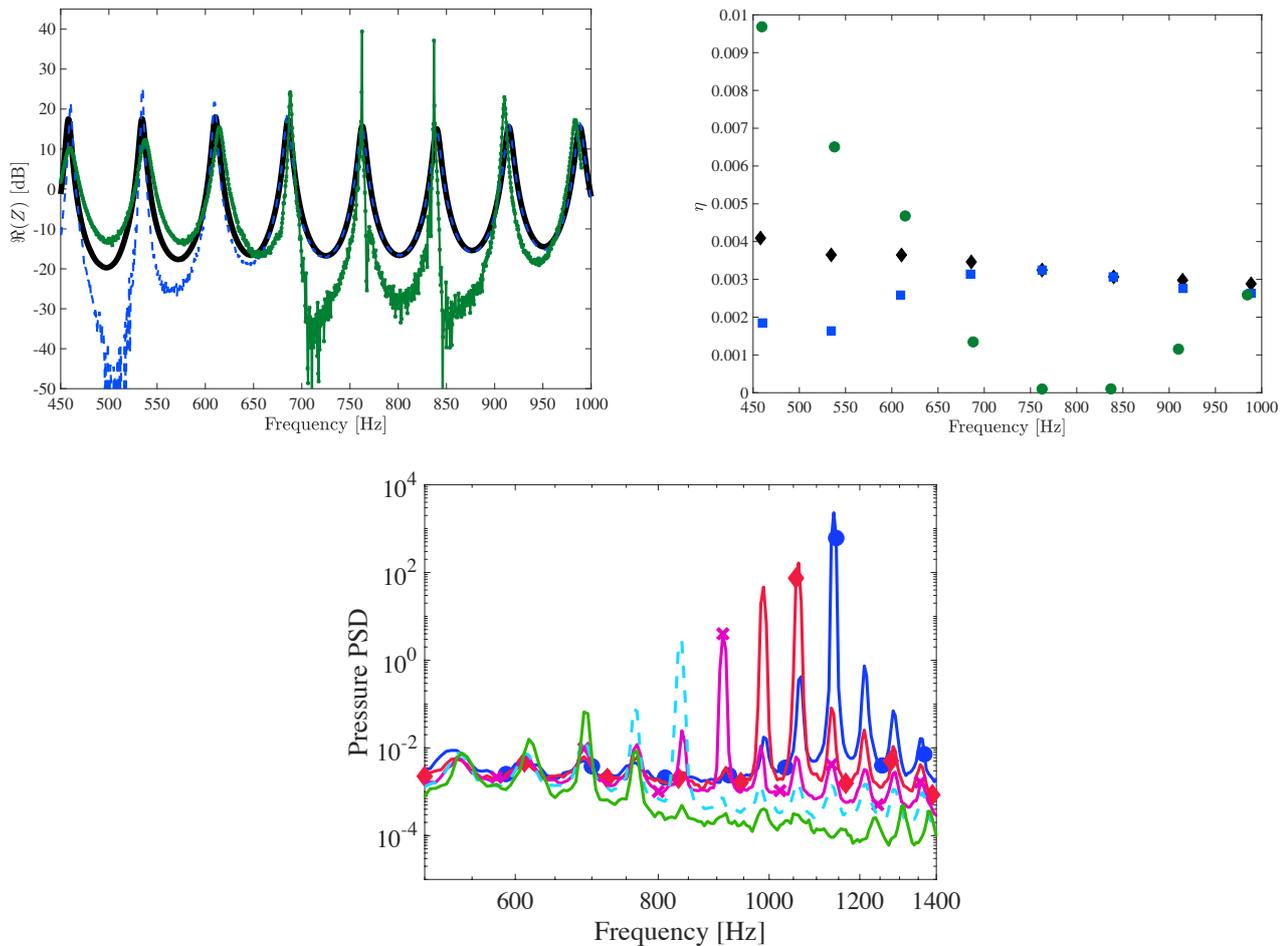


Figure 2: (Left) Real part of the tube impedance as a function of frequency. (Right) Damping rate η of the system resonances as a function of frequency. Black: $M=0$. Blue: $M=0.016$. Green: $M=0.025$. (Bottom) Power Spectral Density of the pressure signal recorded by the first downstream microphone. Green solid line: $M=0.023$. Cyan dashed line: $M=0.027$. Magenta solid line with crosses: $M=0.03$. Red solid line with diamonds: $M=0.033$. Blue solid line with circles: $M=0.037$.

Figure 2 (bottom) displays, for multiple flow velocities, the power spectral density (PSD) of the pressure recorded by the first downstream microphone in absence of any external acoustic excitation. When $M=0.023$, the energy decreases with frequency, except around the duct modes that are linearly amplified by the flow. For faster flows, a whistling is heard outside the resonant duct, and peaks at frequencies corresponding to the ones heard out of the tube appear on the PSD. Just after the onset of whistling, the energy is concentrated on two narrow frequency bands that match the weakly damped duct modes revealed by Figure 2 (right).

The duct mode located at 837 Hz is a good example of the transition from linear to non linear regime. Before $M=0.026$, gain is brought by sound-flow interactions and this mode damping is decreased until it reaches 0 for $M=0.025$. Contrary to lower frequency modes at weaker flow velocities, when M is increased above $M=0.025$, the flow gain at this frequency keeps growing. System losses thus become weaker than the gain, damping rate is negative, and the mode is amplified. To expel the energy that accumulates at this mode, the system starts a nonlinear saturation process: a whistling is produced at the frequency of the mode. For increasing flow velocities, this particular process occur for duct modes at higher frequencies.

[1] Golliard, J., Aurégan, Y., & Humbert, T. (2020). Experimental study of plane wave propagation in a corrugated pipe: Linear regime of acoustic-flow interaction. *Journal of Sound and Vibration*, 472, 115158.