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INFLUENCE OF THE RESONATOR DAMPING ON ITS COUPLING WITH A JET-SLOT OSCILLATOR

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Abstract

In ventilation systems, disturbing whistlings can occur at outlets. This whistling phenomenon belongs to the self-sustained tones class which can be produced when a low velocity flow impinges on a downstream obstacle. The coupling of the tones with acoustic resonances (so-called indirect feedback path) can promote high-level harmonic noises. This experimental study deals with the coupling of a jet-slot oscillator, modelling a ventilation outlet, with the flow supply duct resonances. The flow supply duct is damped by using various amounts of acoustic damping foam, and the effect of this damping on the sound production is studied.

It is shown that for a jet exit/obstacle distance higher than 2.5 jet height ($L/H>2.5$), the tone's frequency matches satisfactorily with frequencies predicted by a model describing favourable phase relationships for the occurrence of tones along the feedback loop. When the flow supply duct is resonant, it is noticeable that the tone's frequency occurs at a duct resonance frequency which is close to the frequency described by this model. For lower values of L/H (slot closer to the jet exit), a better knowledge of the convection velocity behaviour along the vortices path could improve the tone's frequency prediction. The use of damping foam within the flow supply duct prevents the occurrence of the tones' coupling with a duct resonance, especially with high amounts of damping material, and dramatically reduces the emitted acoustic levels. For moderate amounts of damping material, the suppression of coupling occurs only for higher flow speeds, as the tone's frequencies are higher and the damping material is then more efficient.

INTRODUCTION

A ventilation system can be a source of discomfort when whistlings are produced. This type of noise which is called tones is especially irritating for human ears due to the distribution of its acoustic energy only concentrated on a few frequency peaks. These whistlings are the result of self-sustained oscillations which occur when a sheared flow impinges on an obstacle. Such tones can be produced by a wide variety of configurations [1, 2, 3].

Their excitation mechanism is as follows: the narrow ambient perturbations at the jet exit, called primary perturbations, are amplified during their convection by the sheared flow until forming vortices. Their impingement on an obstacle produces a secondary perturbation which is fed back to the flow separation and then controls the vortex formation. If a constructive relationship between these two perturbations is met, a self-sustained loop is created. By modelling each step and summing the phase around this loop for one cycle, the frequency f_N leading to a constructive feedback can be determined. for the N^{th} mode of hydrodynamic oscillation (*i.e.* for the presence of N vortices between the jet exit and the obstacle at the same instant):

$$f_N = \frac{\bar{U}_c}{L}\left(N + \frac{1}{4}\right), \quad N = 1, 2, 3, \ldots \tag{1}$$

where $\bar{U}_c$ is the mean convection velocity of the vortices and L the jet exit/obstacle distance [4].

When the secondary perturbation is fed back directly to the jet exit, the feedback is called *direct*. Conversely, if an acoustic resonator placed in the vicinity of the aeroacoustic sound source is excited, controlling the frequency and reinforcing the emitted level, the feedback is called *indirect* [5].

For a given hydrodynamic mode, the system behaviour depends on the type of feedback path. In dominant direct feedback, the tones' frequency (defined as the frequency with highest energy in the emitted sound frequency spectrum) decreases *continuously* when the jet exit/obstacle distance increases and when the jet speed decreases (see Eq. 1). In dominant indirect feedback, the same behaviour is observed but the frequency decrease is stepwise, each step corresponding to a resonance frequency of the resonator [6].

The present work deals with the jet-slot oscillator where a free plane subsonic jet (Mach number $M < 0{,}1$) impinges on a slotted plate [7, 8], modelling a ventilation outlet. In the experimental apparatus used for this study, the flow supply duct can resonate and thus creates an indirect feedback path which increases the acoustic disturbance as it increases the emitted sound level. The influence of this duct frequency response on the occurrence of an indirect feedback path is studied. For this purpose, the duct frequency response is gradually modified by adding different amounts of damping foam. The overall goal of this study is to achieve a better understanding of the coupling between the tones and the flow supply duct resonances, which would make it possible to diagnose or prevent some whistling problems.

EXPERIMENTAL SETUP

An air flow, produced by a blower, passes through an expansion volume and a duct (190 x 90 mm) ended by a slotted plate creating a H=10mm high and 190mm large free jet. U is the maximum value of jet velocity at the exit. The jet impinges on a 4mm thick slotted aluminium plate. The plate slot, bevelled with an 45° angle, is aligned with the jet exit and has the same dimensions (Figure 1.a, 1.b). The jet exit/obstacle distance normalised by the jet height is noted L/H.

Three configurations are studied, with different repartitions of acoustic absorbant: (i) undamped, (ii) half-damped and (iii) damped (Figure 1.c, 1.d, 1.e). The input characteristic of the undamped resonator is evaluated by exciting the duct with a loudspeaker and using microphones to measure the response (Figure 2). These measurements are not input impedance measurements (or admittance in the case of an open end duct). Nevertheless, they make possible the identification the duct resonance frequencies.

The obstacle is moved using a computer-controlled moving system and the pressure measurements are carried out by using a 4944-A B&K ¼” microphone placed 60mm behind the obstacle and 100mm above the jet. Measurements are made

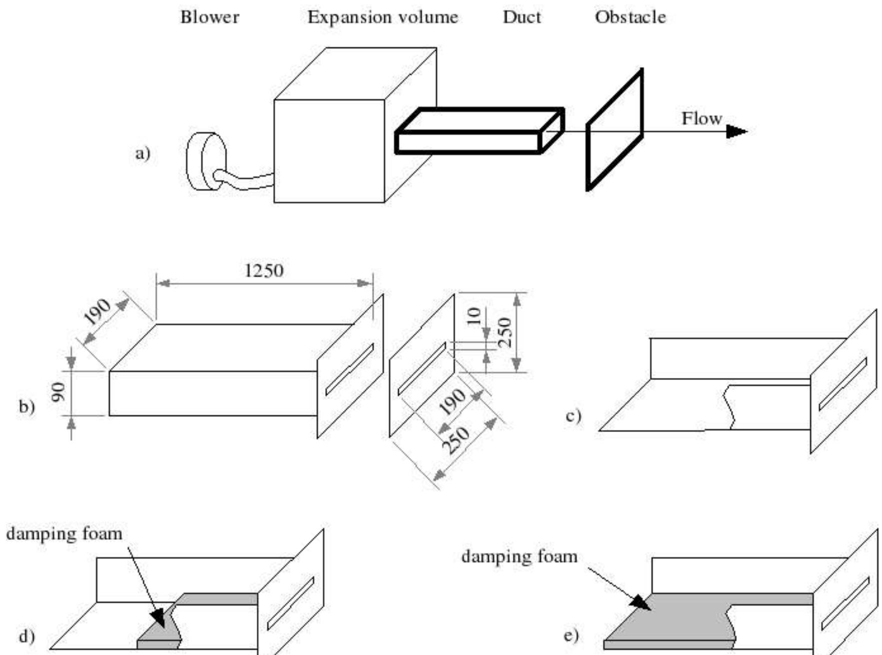


Figure 1 – Scheme of the experimental setup, a) overall view, b) duct and obstacle geometry, in duct view of c) undamped, d) half-damped and e) damped configurations

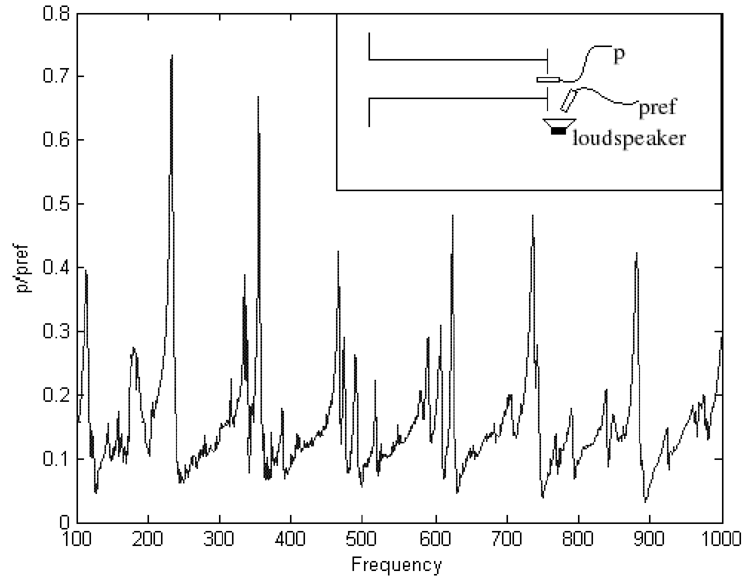


Figure 2 – Input characteristic of the undamped duct

at two flow speeds: U=9m/s and U=20m/s (respectively Re=6000 and Re=13000, Re being the Reynolds number based on the jet height).

RESULTS

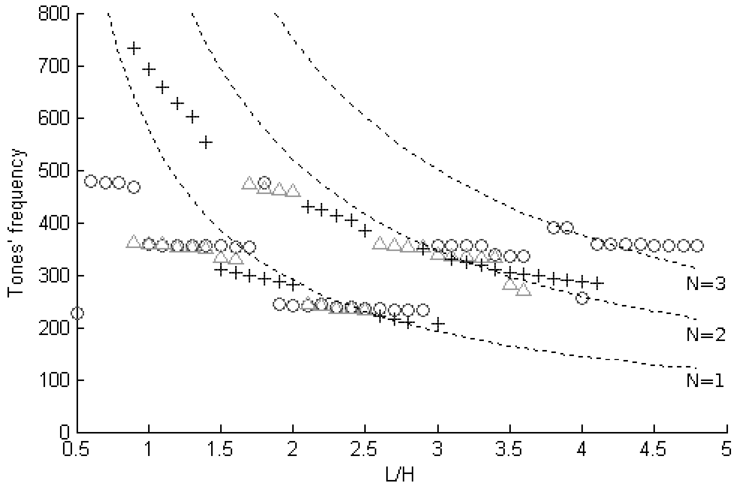
Re=6000

For the three configurations, the evolution of the tones' frequency and the emitted sound level (in dB_{SPL} ref. 2.10^{-5} Pa) when the distance *L/H* is increased is plotted (Figure 3). The hydrodynamic modes frequency predicted using Eq. 1 are also plotted in dotted lines (with N=1,2,3), the vortices mean convection velocity being assumed equal to 0.5*U*, which is an approximation generally admitted.

The overall behaviour is roughly similar for the three configurations. For *L/H>2.5* the tones' frequency follows alternately the dotted lines (Figure 3.a). This permits to identify the operating hydrodynamics modes for a given plate location. For *L/H<2.5*, the tones' frequency get away from the frequency predicted but the different hydrodynamic modes are still identifiable. The frequency jumps corresponding to the transition from one mode to another are characteristic to self-sustained tones production.

For each hydrodynamic mode, a closer consideration permits to identify two behaviours. On one hand, the tones' frequency of the undamped and the half-damped configurations depicts steps at almost constant frequency. These steps frequencies match well with the resonance frequencies of the flow supply duct (Figure 2). In both configurations the dominant feedback is very likely to be indirect. On the other hand, the tones' frequency of the damped configuration decreases continuously, following for large L/H values an hyperbole which is very roughly predicted by Eq. 1. This behaviour is characteristic of direct feedback path.

As stated in the introduction, the coupling with the resonator involves an increase of the emitted sound level. An examination of this level for the three configurations (Figure 3.b) allows one to confirm that the feedback path for the half-damped and the undamped configurations is indirect (sound levels between 80 and 110 dB_{SPL}) whereas the feedback path occurring for the damped configuration is direct (sound levels between 60 and 75 dB_{SPL}).



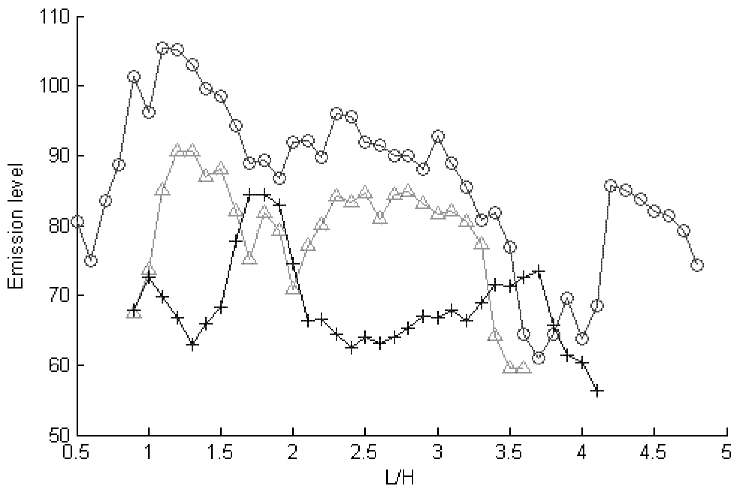


Figure 3 – a) Tones' frequency (Hz) and b) emitted sound level versus jet exit / obstacle distance at Re=6000 for (o) non damped configuration, (△) half-damped configuration and (+) damped configuration; dotted lines corresponds to the predicted hydrodynamic modes (Eq. 1)

Re=13000

The same results as in Figure 3 are depicted in Figure 4 for a higher Reynolds number (13000). Figure 4 shows that the system operates roughly at the same hydrodynamic modes, but the increase of the jet speed involves higher tones' frequencies. The argumentation developed in the previous part permits to identify the dominant feedback path in each configuration.

Only the undamped system presents an indirect feedback loop, with a stepwise evolution of the tones' frequency (Figure 4.a). The sound level emitted reaches $115dB_{SPL}$ for this configuration (Figure 4.b). Conversely, both configuration with

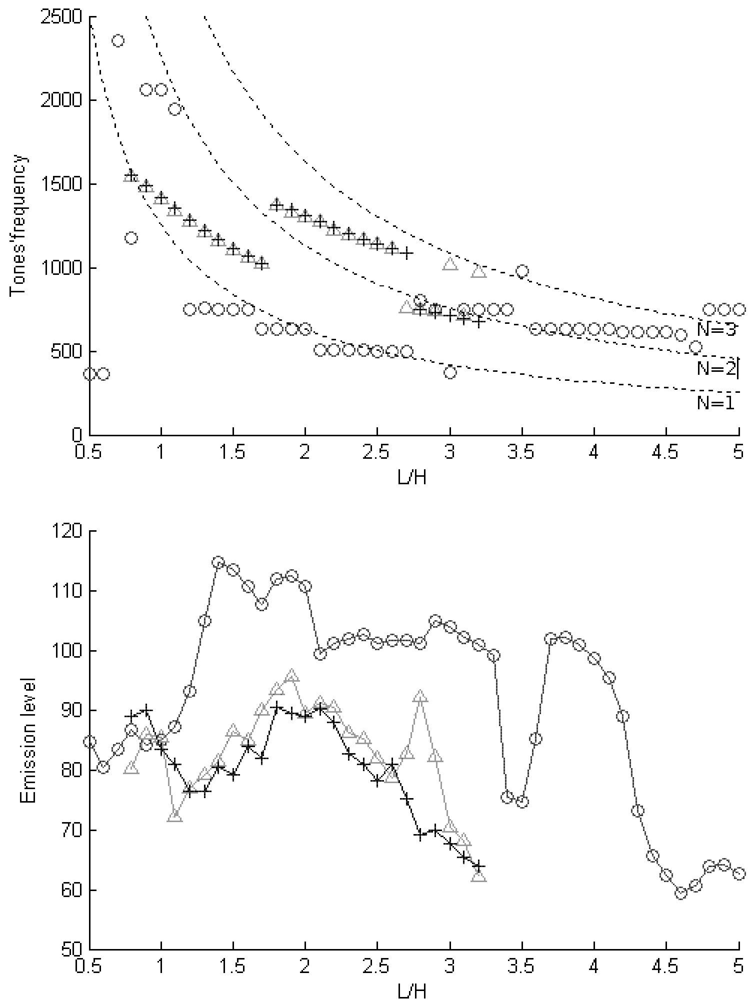


Figure 4 – a) Tones' frequency (Hz) and b) emitted sound level versus jet exit / obstacle distance at Re=13000 for (o) non damped configuration, (△) half-damped configuration and (+) damped configuration; dotted lines corresponds to the predicted hydrodynamic modes (Eq. 1)

acoustic foam involve a direct feedback path: the tones' frequency evolves continuously and the sound level emitted is substantially lowered (between 70 and 90 dB_{SPL}). The domain of existence of the self-sustained tones is reduced in this case since no tones are emitted above *L/H*=3.5.

DISCUSSION

The discrepancy between the measured and predicted (Eq. 1) tone's frequency was already noticed by Powell [4, 9] and was justified by the use of the approximation $\bar{U}_c \propto U$ for different obstacle locations. It was pointed out that the convection velocity may be dependent on both frequency and jet exit/obstacle distance. Thus Eq. 1 only permits a rough prediction of the hydrodynamic modes frequency. It is plausible that the approximation $U_c \approx 0.5$ used is particularly erroneous for small values of *L/H*, for which the discrepancy between predicted and measured frequency is the highest.

In the damped configuration, the acoustic foam damps the resonance modes of the duct; no coupling is thus possible and a direct feedback is imposed. Conversely, in the undamped case, the duct resonances are effective in the sense that they have a sufficiently high quality factor to be excited by external solicitation. The modes density must be sufficient to allow the tones' frequency to follow quite closely the hydrodynamic mode frequency. An indirect feedback is thus possible and the system will work at the resonance frequency the closest to the frequency corresponding to a favourable phase relationship, given by the hyperbole in Figures 3.a and 4.a.

For the half-damped configuration the feedback is indirect at the lowest Re while it becomes direct for the highest Re. This behaviour can be explained by the nature of the acoustic foam which absorbs preferentially high frequencies, and then makes high-frequency resonances less efficient in the coupling process. Indeed, at the low jet speed, the frequency imposed by the hydrodynamic system is quite low. The volume of acoustic foam is not sufficient to damp efficiently the acoustic resonances within the duct, and the feedback is indirect. The frequency evolution is thus similar to the one of the undamped configuration (Figure 3.a), but with a lower sound level (about 10dB). At higher jet speeds (Re=13000), the tones' frequencies increase and the absorption is sufficient to prevent the occurrence of an indirect feedback. The behaviour (frequency and sound level) is thus comparable to the damped configuration.

CONCLUSION

The coupling of self-sustained tones with acoustic resonances (so-called indirect feedback path) can promote disturbing high-level harmonic noises. In this experimental study, the coupling is created by the resonant flow supply duct. To prevent such coupling, the flow supply duct is damped using various amounts of acoustic damping foam. The effect of this damping on the coupling and the tone's frequency is studied.

It is shown that for *L/H>2.5*, the tone's frequency matches satisfactorily with frequencies predicted by a model describing favourable phase relationships along the feedback loop for the tones to occur. When the flow supply duct is resonant, the tone's frequency occurs at a resonance frequency of the duct which is the closest to the frequency described by this model. For lower values of *L/H* (shorter obstacle distance), a better knowledge of the convection velocity behaviour along the vortices path could improve the tone's frequency prediction.

The use of damping foam within the flow supply duct prevents the occurrence of the tones' coupling with a duct resonance, especially with high amounts of damping material, and dramatically reduces the emitted acoustic levels. For moderate amounts of damping material, the suppression of coupling occurs only for higher flow speeds, as the tone's frequencies become higher and the damping material is then more efficient.

This work should be extended with quantitative measurements of the duct input admittance, which could make possible to determinate some quantitative criteria for the occurrence of an indirect feedback.

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