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EFFECT OF VARYING TRIP ROD POSITION ON RESONATOR TUBE OSCILLATIONS

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ABSTRACT

An investigation of the coupling between the vortex shedding from a trip rod with triangular cross-section and a resonant mode of a tube is presented. The physical experiments were conducted using a shallow water channel; this system represents an hydraulic analogy of the acoustic resonator tube in air flow. The resonator tube differs from that used in previous physical experiments as a result of rounding the tips and increasing the size of the trip rod. Rounding the tips is found to reduce the level of vortex shedding from the tips. This increases the amplitude of the tube resonance as result of less damping of the oscillations. The spacing between the trip rod and the resonator tube leading to maximum resonance amplitude increases with increasing flow velocity. Numerical experiments using aeroacoustic theory to explain the transfer of energy between the resonance field and the mean flow field predict well this observed trend, in addition to predicting the source of wave energy.

INTRODUCTION

In this paper, discrete vortex models of the flow have been used in conjunction with a theory of aerodynamic sound due to Howe (1975,1984) to gain an understanding of the mechanisms involved in the energy transfer between the flow and acoustic fields for the specific case of flow past a resonator tube.

Since the experiments of Kawahashi *et al.* (1988), further investigations using the hydraulic analogy have been undertaken. Some advantages of the method are that: (1) it allows easy flow visualisation using the shadowgraph technique, and; (2) the water wave speed is typically several hundred times smaller than the sound speed in air which means that the resonant frequency is much reduced and the flow structures can be observed by eye. The effect of the trip rod position and tube tip shape on the amplitude of the tube resonance is investigated and reported here.

The present paper investigates the relationship between vortex shedding from a wedge-shaped trip rod and the tube resonance in a flow past a resonator tube (figure 1) in terms of Howe's (1974) theory of aerodynamic sound. In particular, an understanding of the phase relationship between the vortex shedding and the resonant acoustic field, and spatial and temporal details of the acoustic source regions, are sought for low Mach number flows. The present application of the theory cannot be used estimate the level of the resonance since it does not consider the transient growth phase or the damping mechanisms; however, it can predict for what system parameters resonance is likely to occur.

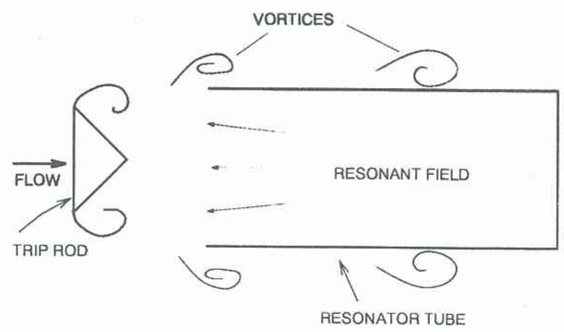

Figure 1: Schematic of the resonator tube and trip rod.

EXPERIMENTAL PROCEDURE: THE HYDRAULIC ANALOGY

A motivation for the present study is the understanding of the excitation of acoustic resonances in resonator tubes in air flow. Results of such measurements in air have been reported by Brocher and Duport (1988). The difficulties associated with flow visualisation in air led Kawahashi *et al.* (1988) to investigate the coupling of the fluid flow with the tube resonance using an hydraulic analogy.

For shallow water flows that are dominated by convection and for which bottom shear can be ignored, the depth-averaged equations are analogous to the compressible Navier-Stokes equations (Zienkiewicz, 1991). The analagous variables are: T (temperature) $\rightarrow$ h (water height); u_i (velocity) $\rightarrow$ U_i (depth-averaged velocity); p (pressure) $\rightarrow 1/2g(h^2 - H_0^2)$; M (Mach number) $\rightarrow$ Fr (Froude number), where H_0 is the mean water depth,

In the present study, the flows are restricted to low Froude number which allows the use of the aeroacoustic theory of Howe (1974), valid for low Mach number. The open channel has width 500 mm and length 2 m. The wedge-shaped trip had a width (D) of 94.5mm, a width to height ratio (D/B) of 1.29 and a width to tube width ratio (D/W) of 0.21. The Reynolds number is defined as $Re = uD/\nu$ and the Froude number is $Fr = u/\sqrt{gH_0}$, where D is the width of the wedge, u_∞ is the mean velocity of the incident flow. Although difficult to completely eliminate, an attempt to restrict tip shedding was made by placing cylinders of circular cross-section at the tube tips.

MODELLING OF THE ACOUSTIC FEEDBACK PROCESS

In this paper, there is no attempt to model the transient approach to resonance or the balance between damping and acoustic energy generation once equilibrium is attained. The present approach is to assume that resonance occurs and to use an aerodynamic theory of sound to investigate whether, for the corresponding calculated flow field, the direction of energy transfer between the flow and the acoustic field enables the resonance to be maintained. The method also allows spatial and temporal details of acoustic sources to be identified and, to some extent, can be used to predict optimal system parameters to maximise or minimise sound generation. Each of these components are only described briefly here. Further details of both the discrete vortex modelling and the application of the aerodynamic theory are given in Hourigan *et al.* (1990) and Thompson *et al.* (1992).

The Acoustic Field

The wavelength of the acoustic field is much longer than the distance between the wedge trip and tube opening, where it is assumed that most of the acoustic energy is generated. Using dimensional arguments it can be shown that in this region, the acoustic particle velocity can be approximated by a sinusoidally-oscillating potential flow solution. The potential solution is obtained using a Schwarz-Christoffel transformation from the physical plane (z plane) for the resonator tube and outer channel system to the transformed upper half-plane (λ plane). The acoustic field in the tube is modelled by placing an oscillating potential source at the origin in the λ plane. The uniform flow towards the tube is modelled by placing potential sinks at $\quad k$. The transformation is given by

$$z = \quad \frac{W}{2\pi}\left(2\log(\lambda) + (k^2 \quad 1)\log(\frac{k^2 \quad \lambda^2}{k^2 \quad 1})\right) + i\frac{W}{2}, \quad (1)$$

where D is the channel width, $k = \sqrt{D \quad W}$, W is the tube width and $i = \sqrt{\quad 1}$. The potential flow velocity field, which includes the uniform inflow and the oscillating flow modelling the resonant acoustic field is given by

$$v_{pot} = u_\infty \left(1 + \frac{1}{(\lambda^2 \quad 1)^*}(1 + \alpha \sin 2\pi f t)\right), \qquad (2)$$

where * denotes complex conjugate, α and f are the amplitude and frequency, respectively, of the oscillation. The first part of the second term in parentheses accounts for the blockage effect of the tube, resulting in zero time-mean flow into the tube.

Discrete-Vortex Flow Model

The separating flow from the trip rod is simulated by a discrete-vortex model. In particular, the surface-vorticity approach of Lewis (1981) is used. In this method, the surface of the wedge is represented by a number of discrete vortex sheet segments so that the no-slip condition is satisfied on the inside of the contour. At each timestep, these elemental surface vortex sheet segments are replaced by potential line vortices and released into the flow. Once released they are convected by the background potential flow, the simulated acoustic field and the self-induced field due to these and previously-released vortices.

Typically, the surface vorticity of the wedge is split into sixty discrete segments with increased concentration towards the shedding points at the wedge vertices. Each timestep these vortices are released into the flow and new surface vortex-sheet segment strengths calculated to again satisfy the no-slip condition.

The elemental vortices are potential vortices with smoothed cores (of Rankine profile) with radius $\Delta s \quad 3$, where Δs is the mean numerical surface vortex sheet segment length. The results are insensitive to the exact value used. Once released from the surface of the wedge, the vortices are convected using a second-order Adams-Bashford timestepping scheme. Generally, runs are performed using one hundred timesteps per acoustic period but again the results are insensitive to the exact value. (This corresponds to a timestep of approximately $0.1((W \quad 2) \quad v_\infty)$. Typically, approximately 700 vortices are used. A scheme is employed to limit the number of vortices by merging close vortices of like circulation once they are sufficiently far from the wedge and tube opening. Computer runs with up to 3000 vortices verify that the results are not sensitive to the number of vortices.

Interaction of Flow and Sound

The generation of sound by the growth and acceleration of vortex structures is predicted using the theory of aerodynamic sound due to Howe (1975, 1985), which states that the acoustic power P generated in a volume V is given by:

$$P = \quad \rho_0 \int (\omega \quad \mathbf{v}) \cdot \mathbf{u}\, dV = \quad \rho_0 \int \omega \cdot (\mathbf{v} \quad \mathbf{u})\, dV; \qquad (3)$$

where ω is the vorticity, $\mathbf{u}$ is the acoustic particle velocity and ρ_0 the mean density of the fluid. Furthermore, when the vorticity is compact, *i.e.*, it extends over a region which is small relative to the acoustic wavelength, the acoustic power per unit length of vortex tube is given by

$$P = \quad \rho_0 \Gamma \mathbf{k} \cdot (\mathbf{v} \quad \mathbf{u}_o) \sin(2\pi f t) \,. \qquad (4)$$

Here, Γ is the circulation of the vortex structure, f is the acoustic frequency, and $\mathbf{u}_o$ the acoustic particle velocity amplitude vector.

For this model it is assumed that the resonant acoustic field is in equilibrium. No energy exchange actually takes place between the acoustic and flow components in the model. In reality, the amount transferred during any cycle is small, so it is expected that the effect on the flow of neglecting this back reaction would be small.

Equation 3 indicates that energy transfer from or to the acoustic field can only occur when the vortices cut across acoustic field lines. Furthermore, for aggregate non-zero energy transfer to occur over an acoustic cycle, there must be variation in $\mathbf{u}$ or $\mathbf{v}$ over this period. In the present study it will be seen that the transfer of energy to the acoustic field occurs when the large-scale vortex structures traverse the gap between the wedge and tube tips.

RESULTS AND DISCUSSION

Modification of the leading tips of the resonator tube by the positioning of cirular rods led to significantly larger resonant oscillations (typically 50% larger) than for the original sharp tips. Figure 2 shows the maximum oscillation amplitude ($\Delta h \quad h_o$) as a function of the inflow velocity (v_∞) when

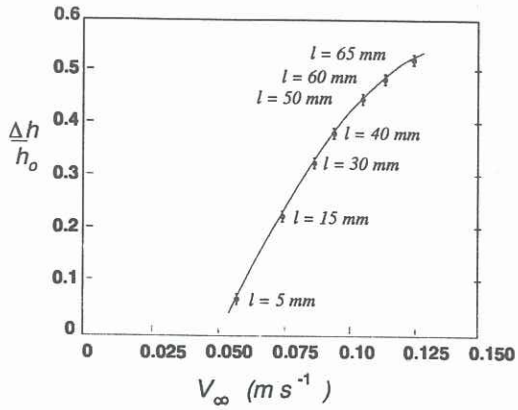

Figure 2: Observed variation of the maximum amplitude of the resonant oscillation versus the inflow velocity. Also shown is the distance (l) between the wedge and tube opening leading to the maximum amplitude.

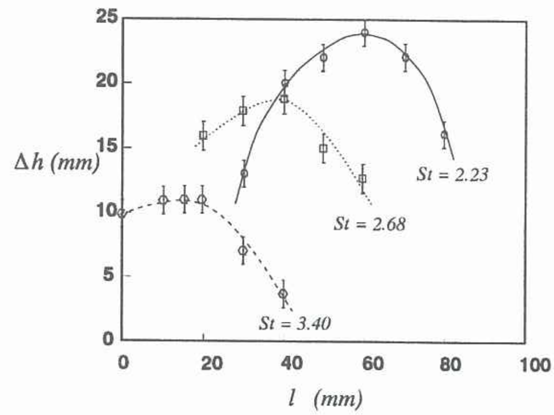

Figure 4: Observed variation of amplitude Δh versus wedge-to-tube distance (l) for three forcing Strouhal numbers St.

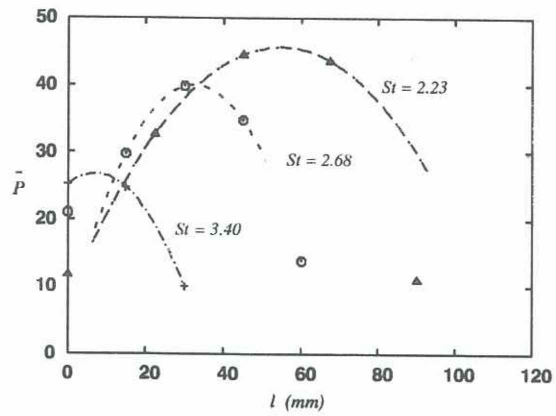

Figure 3: Predicted variation of energy transferred to the acoustic field per cycle ($\bar{P}$) (scaled units) as a function of wedge to tube distance l.

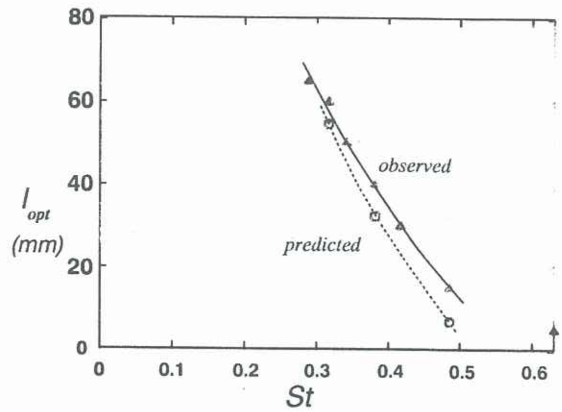

Figure 5: Predicted ( ) and observed ( ) variation of the optimal spacing l_{opt} between the wedge and the resonator tube.

the tip modification was employed. The tube length was fixed at 350 mm. The distance between the wedge and the tube opening (l) is shown for each reading. This represents the optimal distance to within 5 mm. The highest ratio of the (velocity) amplitude of the oscillating field to the inflow velocity, $\epsilon=\Delta h\ (2Fr\ h_o)\ (\cdot\ 100)$ where $Fr=v_\infty\ \sqrt{gh_o}$ is the Froude number, in this case is approximately 150% which occurs at v_∞=0.113 m·s^{-1} and l = 60 mm.

Although the numerical model cannot be used to predict the amplitude of the resonant field, it can predict whether resonance is likely to occur for any particular set of parameters l, f (and α), by calculating the direction of energy transfer between the flow and acoustic fields. Figure 3 shows the predicted energy transfer , P (arbitrary units), per oscillation period as a function of l for the three Strouhal numbers plotted in figure 4. These results are obtained by integrating the predicted power over time for ten oscillation periods and using equation 3 to calculate the energy transfer to the acoustic field. The average energy transfer per cycle ($\bar{P}$) is estimated from the last six cycles.

The value of α chosen for each velocity corresponds to the optimum experimental value (figure 2). Parabolas are drawn through the three points closest to each maximum in order to estimate the optimal l. Figure 5 shows the variation with Strouhal number for both the experimental results and numerical predictions. The numerical predictions are within 10 mm of the observed values for the three Strouhal numbers used for the calculations.

Figure 6 depicts the vortex shedding pattern from the wedge over an acoustic cycle for St=2.68 and l=45 mm. The large-scale vortex structures form during the inflow phase of the cycle. During this part of the cycle, the mean velocity of the vortex structures is small and little energy transfer between the flow and acoustic fields occurs. This initial formation period is associated with a small temporal acoustic sink. When the acoustic field reverses direction the vortex structures traverse the gap between the wedge and tube tips. During this time, the vortex structures move quickly, cutting across the acoustic field lines. According to equation 3 this leads to significant transfer of energy from the flow field to the acoustic field. This period is associated with a temporal acoustic source. Once the vortices pass the tube tips they travel approximately parallel to the acoustic field lines and consequently little further energy transfer takes place.

The spatial distribution of sound sources and sinks is depicted in figure 7. The variation of time-integrated acoustic power is displayed as a greyscale plot. Clearly, the region close to the wedge tips is a time-mean acoustic sink (shown

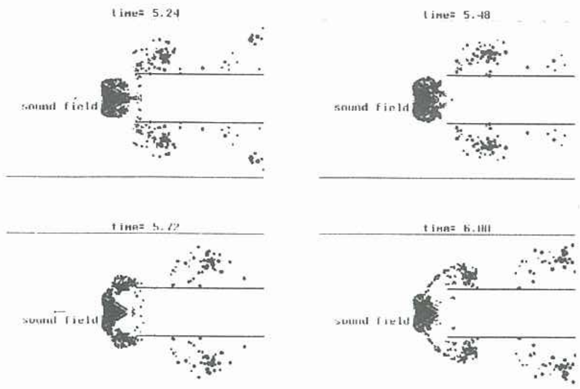

Figure 6: Predicted vortex shedding over an acoustic cycle. The dots represent the elemental vortices shed from the wedge.

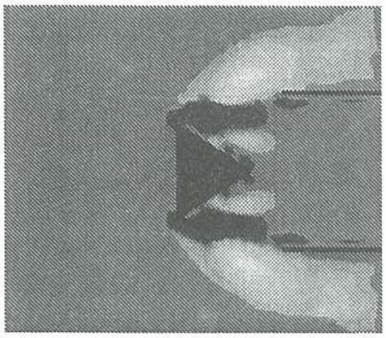

Figure 7: Greyscale plot of predicted time-integrated acoustic power showing the spatial distribution of acoustic sources and sinks. Black represents mean energy transfer from the sound field and white represents transfer to the sound field.

as black) and the regions between the wedge and tube tips (and beyond) are time-mean acoustic sources (shown as white).

ACKNOWLEDGEMENTS

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