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On whistling of pipes with a corrugated segment: Experiment and theory

Oleksii Rudenko*, Güneş Nakiboğlu, Ad Holten, Avraham Hirschberg

Department of Applied Physics, Eindhoven University of Technology, Eindhoven 5600 MB, The Netherlands

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ABSTRACT

Corrugated pipes are commonly used because of their local rigidity combined with global flexibility. The flow through such a pipe can induce strong whistling tones, which is an environmental nuisance and can be a threat to the mechanical integrity of the system. This paper considers the use of a composite pipe: a shorter corrugated pipe segment embedded between smooth pipe segments. Such a pipe retains some flexibility, while the acoustical damping in the smooth pipe reduces whistling tones. Whistling is the result of coherent vortex shedding at the cavities in the wall. This vortex shedding is synchronized by longitudinal acoustic waves traveling along the pipe. The acoustic waves trigger the vortex shedding, which reinforces the acoustic field for a critical range of the Strouhal number values. A linear theory for plane wave propagation and the sound production is proposed, which allows a prediction of the Mach number at the threshold of whistling in such pipes. A semi-empirical approach is chosen to determine the sound source in this model. This source corresponds to a fluctuating force acting on the fluid as a consequence of the vortex shedding. The functional form of the Strouhal number dependency of the dimensionless sound source amplitude is based on numerical simulations. The magnitude of the source and the Strouhal number range in which it can drive whistling are determined by matching the model to results for a specific corrugated pipe segment length. This semi-empirical source model is then applied to composite pipes with different corrugated segment lengths. In addition, the effect of inlet acoustical convective losses due to flow separation is considered. The Mach number at the threshold of whistling is predicted within a factor 2.

1. Introduction

A corrugated pipe is a tube with a periodically modulated diameter. This undulatory shape makes thin-walled corrugated pipes locally rigid, while they retain global flexibility. Various industrial applications benefit from corrugated pipes' utilization ranging from vacuum cleaners to offshore natural gas production [1]. A drawback of corrugated pipes is the production of strong whistling sounds induced by the flow through the pipe. This whistling is an environmental nuisance and associated vibration can lead to a mechanical failure [2].

In this paper we consider a generic experiment in which a corrugated pipe segment is placed between two smooth pipe segments, which we call a composite pipe. This configuration, inspired by the work of Elliott [3], is based on the idea that for a sufficiently long smooth pipe the whistling can be suppressed by thermo-viscous damping, while the global pipe flexibility

* Corresponding author. Tel.: +31 40 247 2163 (Office); fax: +31 402438272.
E-mail addresses: o.rudenko@tue.nl, oleksii.rudenko@gmail.com (O. Rudenko).

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(and mechanical adjustability) still remains. We will describe a theoretical model allowing to design such composite pipes on the basis of a limited amount of experimental data on the whistling of a corrugated pipe segment. The aim of the model is to determine the Mach number range for which whistling can occur for specific acoustical modes. The model will not predict the whistling amplitudes.

The whistling of a corrugated pipe is the result of a coupling between local vortex shedding at the cavities formed by the corrugations and acoustic waves traveling along the pipe [2–17]. The hydrodynamic instability of the shear layer separating the main flow from the fluid in the cavity acts as an amplifier transferring energy from the main flow to the acoustic waves. These acoustic waves induce hydrodynamic perturbations of the shear layer by vorticity shedding at the upstream edge of the cavity resulting in a feedback loop. Whistling is a self-sustained oscillation driven by this feedback loop.

For relatively short whistling pipes, involving strong acoustical reflections at the pipe ends, the vorticity shedding at the upstream edge of the corrugations is triggered by the grazing oscillating velocity, u', associated with resonant acoustic longitudinal standing waves in the pipe.[1] The unsteadiness of the flow at each corrugation results into a fluctuating hydrodynamic force on the pipe walls with a component in the direction of the pipe axis. As a consequence of Newton's third law, this force of the fluid on the wall is associated with an equal and opposite reaction force of the wall of the fluid. It has been suggested by Gutin [21], demonstrated by Curle [22] and verified by Powell [23] that the fluctuating reaction force from the walls on the fluid is a dipolar source of sound. In a low frequency model, which assumes plane acoustic waves propagation, such a dipole sound source is represented by a fluctuating "discontinuity" $\Delta p'$ in the acoustic pressure. This pressure source $\Delta p'$ is the magnitude of the reaction force divided by the minimal corrugated pipe cross-section area. The dipole nature of the sound source has been confirmed in multiple side branches' experiments [10,11,19]: plugging the branches near nodes of the acoustic velocity did not affect the whistling amplitude, while plugging the branches near the acoustic velocity anti-nodes (acoustic pressure nodes) dramatically reduced the whistling amplitude.

The synchronization of the sound source and the acoustic oscillations is characterized by the ratio of convection time of vorticity perturbations across the cavity and oscillation period of the acoustic field. This ratio corresponds to the Strouhal number,

$$Sr \equiv fW_{\text{eff}}/U_{\text{cp}}, \tag{1}$$

where f is the oscillation frequency and U_{cp} is the steady flow velocity in the corrugated pipe averaged over the inner minimal cross-section of the corrugated pipe. The effective cavity width is defined in accordance with Ref. [11]: $W_{\text{eff}} = W + R$, i.e. it is a sum of the cavity width W and the upstream edge radius R, see Fig. 1. The whistling is the strongest when the sound source is in phase with the acoustic velocity oscillation. This corresponds to a specific value of the Strouhal number, which is further referred to as the peak Strouhal number Sr_{peak}.

The oscillation frequency will commonly be close to the acoustic passive resonance frequency of the pipe. This corresponds to an acoustic pipe mode. Such modes are standing waves due to strong reflections at the pipe terminations. However, as will be discussed later, whistling and associated synchronization of vorticity shedding can also be induced when the pipe has non-reflecting (anechoic) terminations. This is an unexpected prediction of the model that has been developed.

At very low acoustic amplitudes, the oscillation amplitude can grow or decay exponentially as an acoustic wave travels along a corrugated pipe. In this case the sound source amplitude is proportional to the local amplitude $|u'|$ of fluid velocity perturbation u' induced by the acoustical waves. We call this regime a linear regime. As the amplitude increases, vorticity perturbations of shear layers result into the formation of discreet vortices. This is a nonlinear saturation mechanism. When this nonlinear regime is reached, a steady whistling amplitude can be established (limit cycle) [10,12].

The steady whistling amplitude is a result of balance between the power produced by the sound sources and the power losses due to thermo-viscous damping and radiation. If damping is sufficiently high, whistling does not occur.

The acoustic power generated by a *single* corrugation for a given imposed acoustical velocity perturbation has been studied by means of numerical simulations in [12–14]. The method combines incompressible flow simulations with Vortex Sound Theory to estimate the strength of an acoustic source of a single cavity in a pipe flow at high Reynolds numbers with a low Helmholtz number acoustic field. The incompressible flow assumption is justified by the fact that the cavity width is very small compared to the acoustic wavelength. At the inlet, a velocity, $\mathcal{U}(r,t) = \overline{\mathcal{U}}(r) + |u'|\sin(2\pi f t)$, is imposed. This velocity consists of the time-averaged mean velocity profile, $\overline{\mathcal{U}}(r)$, with r being the radial distance from the pipe axis, and a uniform acoustic velocity oscillating in the axial direction with a frequency, f, and an amplitude, $|u'|$.

The time averaged acoustic source power, $\langle P_{\text{src}} \rangle$, generated in a control volume is calculated via the surface integral of the product of fluctuating total enthalpy and mass flux through the boundary of the control volume [24]. The simulations of the pipe with the cavity are corrected by subtracting the reference state simulations. These reference calculations are performed in a uniform (without the cavity) pipe with the same boundary conditions. In such a way the high Reynolds numbers limit is approximated, in which the solution becomes Reynolds number independent.

[1] While due to the flow separation the vorticity is continuously shed at the upstream edge of each cavity, the instability of the resulting shear layers leads to the formation of coherent vortical structures. Acoustic perturbations driving oscillations in the grazing flow modulate the vorticity distribution along the shear layer. When acoustic velocity perturbations reach an amplitude above 0.1 percent of the mean flow velocity, the formation of the coherent structures is determined by the longitudinal acoustic velocity perturbation, u'. A new vortex is formed each time the acoustic velocity turns from flow upwards to flow downwards with respect to the main flow [18–20].