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Numerical study of the fluid-dynamic loading on pipes conveying fluid with a laminar velocity profile

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ABSTRACT

The paper deals with the fluid-dynamic forces of laminar fluid flow in long vibrating pipes as well as in long curved pipes at rest. It focuses on the contribution of the fluid-dynamic force related to the centrifugal acceleration. The study is based on the results of a CFD numerical model of an incompressible laminar fluid flow in the pipe, which is deflected in a bending, beam-type mode. The results are presented in terms of the centrifugal correction factor, which is defined as the ratio between the actual centrifugal-related term of the observed fluid-dynamic force and the solution of the one-dimensional model. The estimated values of the respective correction factor are presented for a range of vibration frequencies, lengths of the pipe and amplitudes of the pipe deflection. The results show that all the listed parameters affect the centrifugal correction factor. The resulting values are also compared and assessed with respect to some accessible analytical solutions. The influence of the predicted values of the centrifugal correction factor on the onset of static and dynamic instabilities of pipes conveying fluid is also discussed.

1. Introduction

The fluid-conveying pipe is a fundamental dynamical problem in the field of fluid–structure interactions (Païdoussis, 2014, 2016). This topic has several engineering applications and also serves as a model for understanding more complex systems and for searching out new dynamic features and phenomena. The modelling of fluid-conveying pipes is typically focused on an analysis of the dynamic behaviour and stability issues. In the field of Coriolis flow metering, which represents the application of fluid-conveying pipes for direct measurements of the mass flow rate and the density of the fluids, the aim of the modelling is to examine the measurement characteristics and different influential parameters (Baker, 2016; Wang and Baker, 2014).

The use of analytical modelling for fluid-conveying pipes is dependent on the availability of suitable analytical mathematical models for the fluid-dynamic loading acting on the pipe. The currently available analytical models have different ranges of application, because they are based on different assumptions regarding the pipe and the fluid flow through the pipe. Regarding their treatment of the fluid flow, these models can be divided into one-dimensional models and wave models. See, e.g., some early contributions to the evolution of the one-dimensional models from Housner (1952), Benjamin (1961) and Gregory and Païdoussis (1966), etc., and of the wave models from Païdoussis and Denise (1972), Weaver and Unny (1973), Weaver and Myklatun (1973) and Shayo and Ellen (1974), etc. More exhaustive and recent review of this topic can be found in Païdoussis (2014, 2016).

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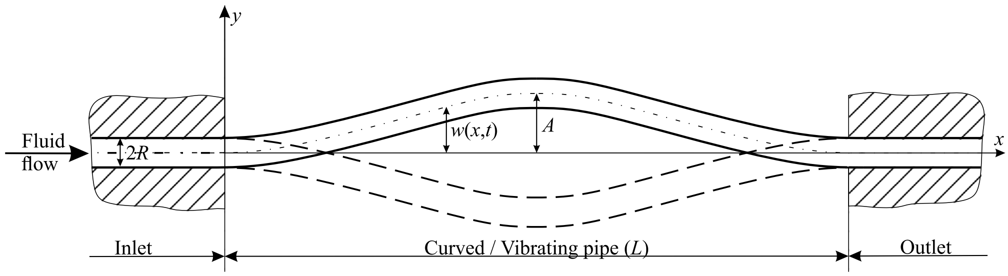

Fig. 1. Sketch of a pipe conveying fluid deflected in a beam-type mode.

Here, the fluid flow can be divided into the mean flow, which refers to the flow in the undeflected pipe, and the perturbed flow, which considers all the variations relative to the mean flow resulting from the pipe deflections. As will be presented in the continuation of this section, there are some open questions about the proper application ranges of particular analytical models for the fluid-dynamic loading, as well as some discrepancies between the results of different authors. In this respect, the primary aim of the paper is to develop a numerical method that will allow the analytical models to be validated for the fluid-dynamic loading.

The study presented here is focused on straight fluid-conveying pipes that deflect in a bending, beam-type mode (see Fig. 1). The pipe deflections $w(x,t)$ and the corresponding perturbed flow field are assumed to be small enough so that the non-linear effects can be neglected. The fluid-dynamic force per unit length can be written as the sum of three components that are related to the translational acceleration $\partial^2 w/\partial t^2$, the Coriolis acceleration $2\overline{V}\partial^2 w/\partial t\partial x$ and the centrifugal acceleration $\overline{V}^2\partial^2 w/\partial x^2$ of the fluid:

$$f_d(x,t) = -\alpha_0 M_f \frac{\partial^2 w}{\partial t^2} - \alpha_1 2 M_f \overline{V} \frac{\partial^2 w}{\partial t \partial x} - \alpha_2 M_f \overline{V}^2 \frac{\partial^2 w}{\partial x^2}, \tag{1}$$

where M_f is the fluid mass per unit length, $\overline{V}$ is the average axial velocity of the mean flow and α_j, $j = 0$, 1 and 2 are the corrections factors that depend on the assumptions of the model. First, we define some dimensionless numbers that will serve in discussions of the model assumptions:

$$\mathrm{Re} = \frac{2\overline{V}R}{\nu}, \quad \mathrm{Re}_\omega = \frac{\omega R^2}{\nu}, \quad \varepsilon_L = \frac{R}{L}, \quad \varepsilon_C = \frac{\omega R}{c}, \tag{2}$$

where ν is the fluid kinematic viscosity, c is the fluid speed of sound, ω is the angular frequency of the pipe, R is the pipe internal radius and L is the pipe length. The Reynolds number Re represents the ratio of the inertial forces to the viscous forces within the mean flow field; the velocity profile approaches the uniform distribution for turbulent flow at high Reynolds numbers ($\mathrm{Re} \gg 10^4$), but the parabolic velocity profile is achieved for a fully developed flow in a straight circular pipe under laminar flow conditions ($\mathrm{Re} < 2000$). The vibrational Reynolds number Re_ω represents the ratio of the inertial forces to the viscous forces within the perturbed flow field; the viscous effects become less significant at higher vibrational Reynolds numbers. The dimensionless numbers ε_L and ε_C represent the magnitudes of the pipe length effects and the fluid compressibility effects, respectively, on the perturbed flow field.

Kutin and Bajsić (2014) estimated the correction factors α_j of the uniform and laminar mean-flow velocity profiles for different circumferential mode shapes of the straight pipe. This study is based on a mathematical model that assumes inviscid perturbed flow, so it is valid for sufficiently high vibrational Reynolds numbers Re_ω. The results for the beam-type pipe, for the case of the uniform mean-flow velocity profile, can be written as:

$$\begin{aligned} \alpha_0 &= 1 + \frac{1}{4}R^2\frac{\partial^2}{\partial x^2} + \frac{1}{4}\left(\frac{\omega R}{c}\right)^2, \quad \alpha_1 = 1 + \frac{1}{4}R^2\frac{\partial^2}{\partial x^2} + \frac{1}{2}\left(\frac{\omega R}{c}\right)^2, \\ \alpha_2 &= 1 + \frac{1}{4}R^2\frac{\partial^2}{\partial x^2} + \frac{3}{2}\left(\frac{\omega R}{c}\right)^2, \end{aligned} \tag{3}$$

and, similarly, for the case of the laminar mean-flow velocity profile:

$$\begin{aligned} \alpha_0 &= 1 + \frac{1}{4}R^2\frac{\partial^2}{\partial x^2} + \frac{1}{4}\left(\frac{\omega R}{c}\right)^2, \quad \alpha_1 = 1 + \frac{5}{12}R^2\frac{\partial^2}{\partial x^2} + \frac{7}{12}\left(\frac{\omega R}{c}\right)^2, \\ \alpha_2 &= \frac{2}{3} + \frac{5}{8}R^2\frac{\partial^2}{\partial x^2} + \frac{43}{24}\left(\frac{\omega R}{c}\right)^2, \end{aligned} \tag{4}$$