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DYNAMICS OF A HYDROELASTIC OSCILLATING CYLINDER WITH ADDED VISCOELASTIC DAMPING FOR PASSIVE CONTROL OF VIBRATIONS INDUCED BY VORTEX

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Abstract. *In the scope of flow around cylindrical structures, the vortex shedding phenomena are those that have attracted considerable attention from many mechanical and civil engineering domains. Since, these phenomena – dependent on a number of factors – may have disastrous effects in engineering practice, such as economic loss, damage of installations, loss of power-generating time, etc. This is a reason for which in the last decades, a great deal of effort has been devoted to the development of numerical and computational procedures for dealing with the problem of vortex shedding phenomena. Among the various new developments for dealing with the problem of vortex-induced vibration, the use of viscoelastic materials is considered as an interesting strategy, since they present great efficiency in mitigating vibrations levels at moderate application and maintenance costs. In this paper, the Immersed Boundary Method combined with the Virtual Physical Model is used to investigate the forces and response associated with the vortex-induced vibration of a rigid oscillating cylinder incorporating viscoelastic damping. In the context of fluid-structure interactions, an important issue addressed herein is the modeling procedure of the viscoelastic behavior in the time domain, which is performed by using a fractional-order constitutive model. After the presentation of the underlying theoretical foundations related to the various aspects of the numerical modeling procedure, numerical simulations with a viscoelastically damped oscillating cylinder performed in a wide range of oscillating amplitude and forcing frequency are presented and discussed aiming at demonstrating the influence of viscoelastic damping on the flow structure and fluid forces.*

Keywords: *Immersed boundary method, virtual physical method, oscillating cylinder, vortex shedding modes, viscoelastic damping.*

1. INTRODUCTION

Incompressible viscous flows over circular cylinders have been studied by many researchers (Da Silva et al, 2009), (Lima and Silva et al, 2007), (Bao et al, 2010) and (Da Silva et al, 2011). However, nowadays it is also important to appreciate the numerical studies involving vortex-induced vibrations (VIV) of a circular cylinder due to its great importance both academic and industrial domains. It is especially true if the main interest is to investigate control strategies of the undesirable induced vibrations with the aim of increasing the fatigue life of engineering systems subjected to the VIV problem.

In practice, the VIV problem can occurs in many engineering fields, such as bridges, transmission lines, maritime structures, drilling and petroleum production, hydrodynamic and hydro-acoustic applications. Thus, several methods have been developed to overcome this class of flow problem to increase the understanding, the prevention and the prediction of the VIV phenomenon and to quantify the relation between the structure response and the most important parameters related to it. As an example, the behavior of long cylindrical structures immersed in water has been extensively investigated due to development of the hydrocarbons sources in depth greater than 1,000 meters (Sarpkaya, 2004). Within this context, experimental and numerical efforts have been made to characterize the vortex shedding frequency, the response frequency, the induced forces, the damping effects (Zhou, 1999) , the time histories of the displacement, the acceleration and the total force acting on elastically mounted circular cylinders in uniform flows (Vikestad et al, 2000). The wake vortices (AI-Jamal et al, 2004) and the response amplitude of the cylinder have been also addressed by many authors (Klamo et al, 2006), (Singh et al, 2005), (Khalak et al, 1997) ans (Khalak et al, 1999).

Thus, several works have been proposed for performing fluid-structure interaction analyses of systems, as reported in the references. However, few studies have addressed the problem of estimating the VIV problem of systems incorporating viscoelastic damping devices, which motivates the study reported in this paper.

Several approaches have been developed for performing dynamic responses of simple and more complex engineering structures containing viscoelastic damping devices, as reported in (Da Silva et al, 2009) , (Lima and Silva et al, 2007) and (Anagnostopoulos, 1994). Among the widely used mathematical representations accounting for the typical dependence of the viscoelastic properties with respect to frequency and temperature, in this paper the four-parameter fractional derivative model (FDM) originally proposed by (Bagley and Torvik, 1983) and modified by (Galucio et al, 2004) for the transient analysis of viscoelastically damped systems, has been retained to be combined with the VIV problem in order to the mitigate undesired levels of vibration. In this context, it is used the Immersed Boundary Methodology (IBM) combined with the Virtual Physical Model (VPM) to simulate the flow around the viscoelastically mounted circular cylinder composed by one degree of freedom in an uniform flow.

Therefore, the motivation intended for the present study is to provide a detailed analysis of the flow features over a two-dimensional viscoelastically mounted circular cylinder with one degree of freedom based on the vortex shedding mechanism visualized by the vorticity contours, the drag and lift coefficients aiming to investigate the influence of the viscoelastic damping on the cylinder response.

2. MATHEMATICAL METHODOLOGY

The Immersed Boudary Method (Peskin, 1977) with the Virtual Physical Model (Lima and Silva et al, 2007) is used to simulate two-dimensional, incompressible, isothermal and viscous flow. This methodology is based on the Navier-Stokes equations added a force term, which acts upon the fluid so that it notice the existence of interface, thus making the exchange of information between fluid and solid. In this method, there are two meshes, one called Eulerian, which represents the calculation domain and other called Lagrangean that represents the immersed body (Lima and Silva, 2014).

These meshes are geometrically independent and coupled through the force term. This methodology characteristic enables the study of flows over simple, complex and even mobile and deformable geometry without remeshing process. We can observe the Fig. 1 that shows an illustrative scheme of the both meshes for a two-dimensional domain, with the presence of an arbitrary interface (Lima and Silva, 2014).

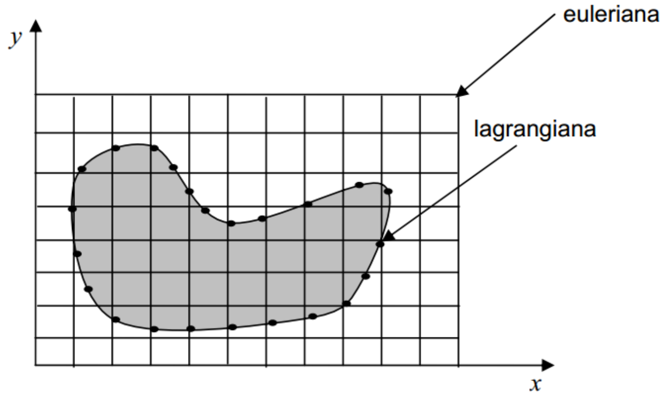


Figure 1- Representation of the Eulerian and Lagrangean meshes for an arbitrary interface.

2.1 Mathematical Formulation for the Fluid

The Navier-Stokes can be expressed in the tensor form, as follow:

$$\frac{\partial u_i}{\partial t} + \frac{\partial (u_i u_j)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\nu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + f_i \tag{1}$$

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{2}$$

where ρ is the specific mass of fluid, ν kinematic viscosity of fluid, u_i are the velocity vector components, p is the pressure and f_i are the components of the Eulerian force vector components.

The source term of Eulerian force only exists in the Eulerian points coincident or near the Lagrangean grid, being null for the order points of the calculation domain. This term is calculated through the distribution of the components of the Lagrangean interfacial force vector (Peskin and McQueen, 1994).

2.2 Virtual Physical Model (VPM)

The Virtual Physical Model (VPM) allows the calculation of the Lagrangean force based on physical interaction of the fluid and the solid surface immersed structure in the flow. The mathematical calculation is based on the application of the balance of the momentum equation of the fluid particles located in the Lagrangean points by observing the following expression (Lima and Silva, 2014):

$$\vec{F}\left(\vec{x}_k, t\right) = \vec{F}_a + \vec{F}_i - \vec{F}_v + \vec{F}_p \tag{3}$$

where $\vec{F}_a$ is the acceleration force, $\vec{F}_i$ is the inertial force, $\vec{F}_v$ is the viscous force and $\vec{F}_p$ is the pressure force.

2.3 Fractional Step Model

The Fractional Step Method (FSM) (Chorin, 1968) is used here with the aim of coupling the pressure-velocity fields. The FSM is a non-iterative method that form the velocity components, pressure and force of the previous iteration in which the velocity components fields are estimated. After obtained, the pressure correction is then calculated through the solution of the linear system. Next, the Poisson equation for the pressure correction is that it makes the connection between the momentum and continuity equations. Therefore, it has values of pressure that allow to calculated the components of velocities (u^{n+1} and v^{n+1}), obtained from Navier-Stokes equations and lastly it needs to test the mass conservation in the time n+1.

2.4. Description of the problem

The present study is focused on the flow over a circular viscoelastically damped cylinder subjected to one degree of freedom. The computational domain is chosen to be $40D \times 30D$ with the cylinder located at $16.5D$ from the inlet boundary. Numerical investigations (Chen et al, 2005) have shown that the downstream boundary located at least $20D$ away from the cylinder center do not influence the numerical solution. The results reported by (Prasanth et al, 2006) showed that the outflow boundary has no significant effect on the flow and on the cylinder response if it is located at least $25.5D$ away from the cylinder center. In this work this distance is assumed to $23.5D$. The lateral boundaries are located at $15D$ from the cylinder center at each side, where $D[m]$ is the cylinder diameter.

The transverse response of the viscoelastically damped cylinder is modeled using a mass-visco-spring-damper model as illustrated in Fig. 2, where the motion of the cylinder is in the normal direction of the free-stream velocity $U[m/s]$ only. Also, the cylinder is assumed to be a rigid body. Therefore, the equation of motion generally used to represent the vortex-induced vibration problem is given as (Meirovitch, 1989) and (Thomson, 1998):

$$m\ddot{y}(t) + c\dot{y}(t) + k_v^* y(t) = F_\ell(t) \tag{4}$$

where $m\ [kg]$ is the mass, $c\ [Ns/m]$ is the damping coefficient, $k_v^*\ [N/m]$ is the frequency- and temperature-dependent viscoelastic stiffness value, $y\ [m]$ is the transverse displacement, and $F_\ell(t)\ [N]$ is the lift force generated by the flow.

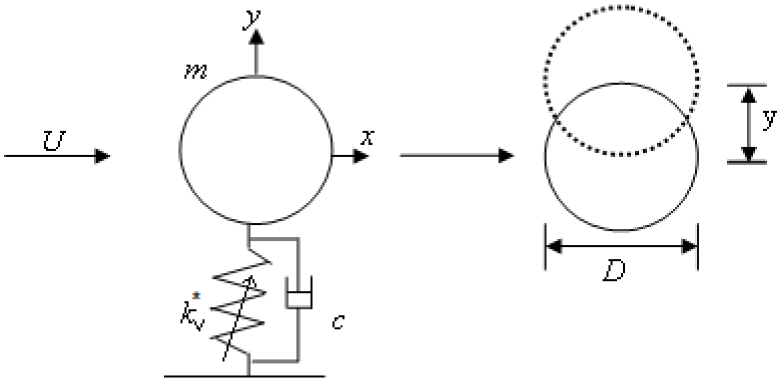


Figure 2- Illustration of the one viscoelastically damped degree of freedom system under consideration.

2.5. Fractional derivative model incorporated into the equation of motion

In the context of the present study Eq. (4) must be solved in order to generate the dynamic responses of viscoelastic cylinder in a flow. Also, it must be taking into account the dependency of the material *moduli* with respect to the frequency and temperature. Various mathematical models have been developed to represent the viscoelastic behavior and have been shown to be suitable to be used in combination with the discretization procedures. In this paper, as the interest is confined to time-domain analyses of VIV problem, the *Fractional Derivative Model* (FDM) initially proposed by (Bagley and Torvik, 1983) is used in combination with the Grünwald discretization technique(Galucio et al, 2004) to approximate the fractional operator appearing in the following one-dimensional stress state constitutive equation for linear viscoelastic materials:

$$\bar{\varepsilon}(t)+\tau^{\alpha}\,\frac{d^{\alpha}\bar{\varepsilon}(t)}{d\,t^{\alpha}}=\frac{E_{\infty}-E_{0}}{E_{\infty}}\,\varepsilon(t) \tag{5}$$

where E_0 and E_∞ are, respectively, the relaxed or static modulus, and non-relaxed or high-frequency limit value of the modulus, τ is the relaxation time, and α represents the fractional order of the time derivative $(0<\alpha<1)$.

One important aspect regarding the use of the complex modulus function, $E_v^*=\left(E_0+E_\infty(i\omega\tau)^{\alpha}\right)/\left(1+(i\omega\tau)^{\alpha}\right)$, obtained by applying the Fourier transform to Eq. (5) is the identification of the four parameters ($E_0, E_\infty, \tau, \alpha$) from experimental data-sheets provided by the manufactures containing the material storage modulus and loss factor as functions of frequency and temperature. Thus, from the complex modulus definition, the determination of the values of the material parameters can be carried out by formulating an optimization problem in which the objective function represents the difference between the experimental data points and the corresponding model predictions in the frequency band of interest for a fixed temperature value (Galucio et al, 2004). Also, it must be emphasize that Eq. (5) was obtained by introducing the following internal variable as a strain function, $\bar{\varepsilon}(t)=\varepsilon(t)-\sigma(t)/E_\infty$, in the standard one-dimensional stress state constitutive equation for viscoelastic materials (Nashif et al , 1985). As a result, Eq. (5) contains only one fractional derivative term, $d^{\alpha}\bar{\varepsilon}(t)/d\,t^{\alpha}$, instead of two presented in the classical constitutive equation, and can be approximated by the Grünwald definition, $d^{\alpha}\bar{\varepsilon}(t)/d\,t^{\alpha}\approx\Delta t^{-\alpha}\sum_{j=0}^{np}A_{j+1}\bar{\varepsilon}(t-j\Delta t)$, noting that $A_1=1$, to generate the following discretized form of the anelastic strain:

$$\bar{\varepsilon}(t)=(1-c)\frac{E_{\infty}-E_{0}}{E_{\infty}}\,\varepsilon(t)-c\sum_{j=1}^{np}A_{j+1}\bar{\varepsilon}(t-j\Delta t) \tag{6}$$

In Eq. (6), $\Delta t=t/n$ is the time step, $n_p\le n$ is the number of discretization points, $c=\tau^{\alpha}/\left(\tau^{\alpha}+\Delta t^{\alpha}\right)$ and A_{j+1} represents the Grünwald coefficients given by the following recurrence formulae, $A_{j+1}=(j-\alpha-1)A_j/j$.

At this point, if the viscoelastic stiffness value is considered to be frequency- and temperature-independent, the term $k_v^* y(t)$ appearing in Eq. (4) is defined as $k_v^* y(t) = \int_y B^T(y)\sigma(y,t)dy$ where the vector $\sigma(y,t)$ can be written in terms of the anelastic strain Eq. (6) and the strain relation, $\bar{\varepsilon}(t) = \varepsilon(t) - \sigma(t)/E_\infty$, as follow:

$$\sigma(y,t) = \left(1 + c\frac{E_\infty - E_0}{E_0}\right)\varepsilon(y,t) - c\frac{E_\infty}{E_0}\sum_{j=1}^{np} A_{j+1}\bar{\varepsilon}(y,t - j\Delta t) \tag{7}$$

By combining the Eq. (7) with the elastic and anelastic strain-displacement relations, $\varepsilon(y,t) = B(y)y(t)$ and $\bar{\varepsilon}(y,t) = B(y)\bar{y}(t)$, the resulting modified elementary viscoelastic stiffness matrix assumes the following form:

$$k_v^* y(t) = \tilde{k}_v y(t) + \tilde{f}_v(t - j\Delta t) \tag{8}$$

Where $\tilde{k}_v = \left(1 + c\frac{E_\infty - E_0}{E_0}\right)k_v$ and $\tilde{f}_v(t - j\Delta t) = c\frac{E_\infty}{E_0}k_v\sum_{\boldsymbol{j=1}}^{np} A_{j+1}\tilde{y}(t - j\Delta t)$.

Upon introduction of the Eq. (8) into Eq. (4), the resulting equation of motion of the viscoelastically mounted one degree of freedom cylinder in a flow can be written as:

$$m\ddot{y}(t) + c\dot{y}(t) + \tilde{k}_v y(t) = F_\ell(t) - \tilde{f}_v(t - j\Delta t) \tag{9}$$

Where $\tilde{f}_v(t - j\Delta t)$ is the viscoelastic loading dependent on the anelastic displacements $\tilde{y}(t)$, and the Grünwald coefficients which represent the fading memory phenomenon in the viscoelastic material.

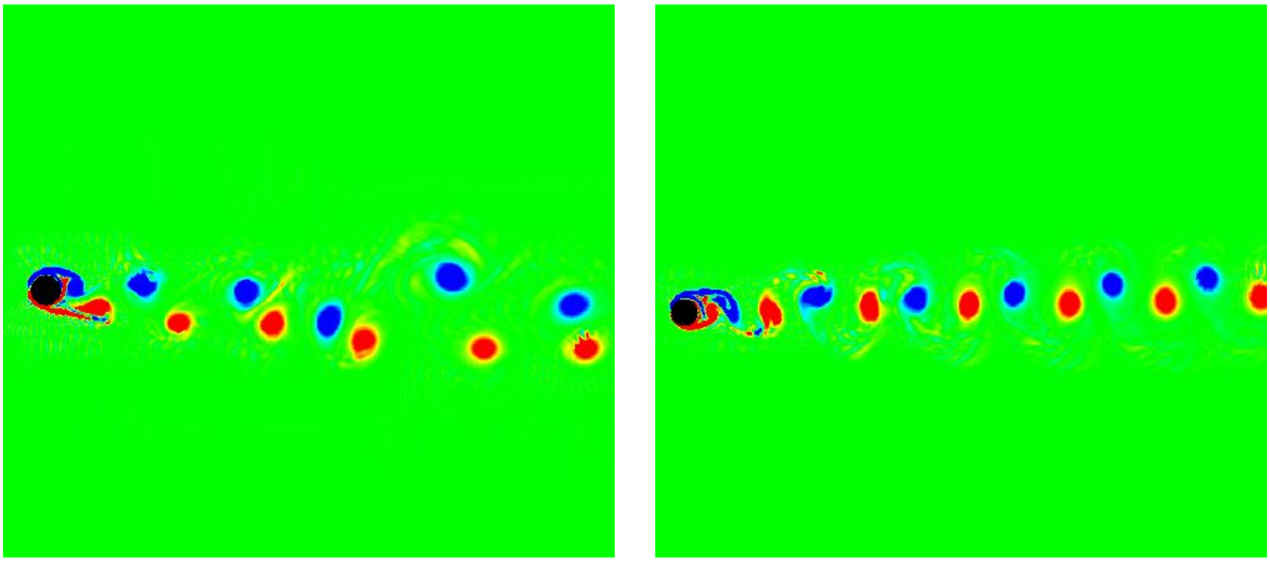
3. NUMERICAL RESULTS

In this section, numerical simulations are presented to illustrate the interaction of the fluid-structure with and without the viscoelastic damping. Now, we considered the follows values of dimensionless parameters: $m^* = 10.30$, $\zeta = 0.07736$, $V_r = 3.5$ and $R_e = 10.000$.

In the same way, we have fixed the value of temperature to investigate the behavior of viscoelastic material in the structure. So, the value of temperature was $T = 10°C$. Therefore, the characteristic of the viscoelastic material in this temperature of work is: $E_0 = 1.2526$ MPa, $E_\infty = 572.44$ MPa, $\tau = 0.0034$ ms and $a = 0.6682$.

3.1 Vorticity Fields

The flow visualization by the vorticity contours is presented in the Fig. 3(a) (without viscoelastic damping) and Fig. 3(b) (with viscoelastic damping). It can be seen a different mode of vortex shedding comparing the two figures. Where there is a ‘2S’ in Fig. 3(b) and for Fig. 3(a) obtained a little different of ‘2S’. This classification is taking into account (Williamson and Roshko, 1988).



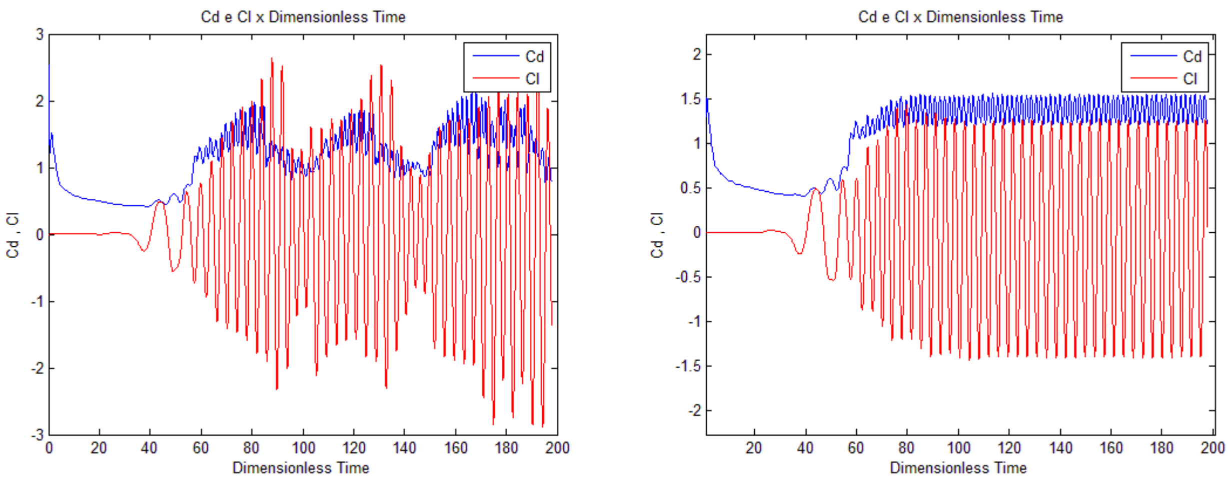
(a) (b)

Figure 3 - Vorticity contours for number Reynolds equal to 10.000, (a) without viscoelastic; (b) with viscoelastic

3.2 Time Histories of the Hydrodynamic Coefficients

Now, as another illustration, Fig. 4 shows the time evolution of the drag (blue) and the lift (red) coefficients for the flow in the cylinder. It can be verified that fluctuations amplitudes of the drag coefficient is different in the Fig. 4(a) (without viscoelastic material) compared by Fig. 4(b) (with viscoelastic material). Then, it can be provided a significant reduction of the oscillation in the inclusion of viscoelastic damping. Moreover, the stability over time of the fluid-structure in the Fig. 4(b) it can be noted.

Also, it can be seen a better behavior of the lift coefficient, that the Fig. 3(a) there is an inconvenient fluctuation of lift coefficient. So, taking into account the Fig. 3(b), it has a stable behavior compared the other one.



(a) (b)

Figure 4. Time histories of the hydrodynamic coefficients number Reynolds equal to 10.000, (a) Without viscoelastic damping and (b) With viscoelastic damping.

4. CONCLUDING REMARKS

In the present work, the numerical simulations over a single circular cylinder immersed in two-dimensional, incompressible, viscous and isothermal flows were carried out at Reynolds number equal to 10.000. In the same way, has been suggested a modeling of viscoelastic structural system containing translational mount.

The first step of simulations, it was illustrated the vorticity fields and compared the structure with and without viscoelastic damping. Then, it can be observed a different mode of vortex shedding, that to the structure with viscoelastic damping obtained a '2S' and the other one a little different of '2S'. So, for the second results, has been showed the different amplitudes of the drag and lift coefficients, that the simulations of the viscoelastic damping improve the both coefficients and turn its more stable.

This paper has purpose an alternative and efficient model of passive control of vibrations induced by vortex (VIV), that the next steps will be analysis the influence of temperature in the system as well as mass ratio and reduced speed. Therefore, it is believed that this strategy can be advantageously used for the next studies of VIV or same applications.

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