A numerical study on passive suppression of vortex-induced vibration (VIV) using an elastic rotative non-linear vibration absorber

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Summary. The focus of this contribution is to present a numerical study on passive suppression of the vortex-induced vibration phenomenon (VIV) using an elastic rotative non-linear vibration absorber (ERNVA). The cylinder (structure to be controlled) is constrained to oscillate only in the cross-wise direction (i.e., the direction orthogonal to the direction of incoming flow). The hydrodynamic loads are modeled using a wake-oscillator model. Curves showing the efficiency of the device are presented and discussed.

Introduction

Vortex-induced vibration (VIV) is a particular flow-induced vibration phenomenon particularly important on risers' dynamics. Phenomenlogical aspects of the phenomenon can be found in the surveys [1] and [2]. Its self-excited and self-limited character plays a role in the structural lifespan due to fatigue. Hence, VIV suppression is of interest for both industrial and academic communities.

A number of studies focusing on VIV suppression can be found in the literature. For the sake of conciseness of this extended abstract, focus is placed on investigations using rotative non-linear vibration absorbers (RNVAs) or non-linear energy sinks (RNES). A rotative NVA is composed of a rigid arm fitted with a tip-mass, linked to the main structure by means of a linear dashpot in which energy is dissipated.

References [3, 4] bring numerical studies focusing on the behavior of a cylinder fitted with a RNVA subjected to VIV. In both papers, the cylinder was constrained to oscillate in the cross-wise direction and the forces applied by the fluid to the solid are computed using computational fluid dynamics (CFD) techniques. In these references, the authors point out the existence of strongly modulated responses for the cylinder and a decrease in the cylinder response. [5] also deals with the passive suppression of VIV using a RNVA, but using reduced-order models based on wake-oscillators for calculating the hydrodynamic load. The latter reference numerically investigates the cases in which the cylinder is constrained to oscillate in one or two directions of the horizontal plane. Due to the lower computational cost compared to the CFD, [5] presents maps showing the sensitivity of the response with respect to the RNVA parameters.

This paper extends the work presented in [5]. Instead of considering a rigid arm, an elastic linear spring allows motion of the suppressor mass in the radial direction. This type of suppressor is herein defined as elastic non-linear vibration absorber (ERNVA). At least to the author's knowledge, this is the first application of such device for VIV suppression.

Mathematical model and results

Consider the problem sketched in Fig. 1. The rigid cylinder has mass M, length L, diameter D and is immersed in a fluid of specific mass ρ and characterized by an uniform free-stream velocity U_∞. The cylinder is assembled onto a viscoelastic support of stiffness k_y and damping constant c_y and its cross-wise displacement is Y. The ERNVA is composed of a point mass m_N placed at the tip of an elastic arm of axial stiffness and damping constant of constant k_r and c_r respectively. The elastic arm is hinged to the cylinder by means of a dashpot of constant c_θ. The instantaneous radial position of the point mass is $r_0 + r(t)$, r_0 being the unstretched length of the arm. Two reference frequencies can be defined, namely $\omega_{n,y} = 2\pi f_{n,y} = \sqrt{k_y/(M + m_N + m_a)}$ (m_a is the potential added mass) and $\omega_r = \sqrt{k_r/m_N}$. Now, consider the dimensionless quantities defined in Eq. 1.

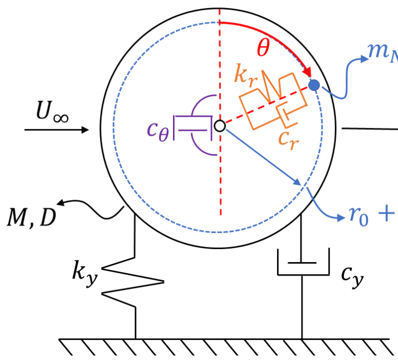


Figure 1: Schematic representation of the problem. The cylinder is constrained to oscillate in the cross-wise direction.

$$y=\frac{Y}{D},\eta=\frac{r}{D},\tau=\omega_{n,y}t,\hat{r}=\frac{r_0}{D},\hat{m}=\frac{m_N}{M+m_N},\zeta_r=\frac{c_r}{2m_N\omega_r},\zeta_y=\frac{c_y}{2(M+m_N+m_a)\omega_{n,y}},\hat{\omega}=\frac{\omega_r}{\omega_{n,y}}$$
$$U_r=\frac{U_\infty}{f_{n,y}D},m^*=4\frac{(M+m_N)}{\rho\pi D^2L},C_a=4\frac{m_a}{\rho\pi D^2L} \tag{1}$$

In addition to the above quantities, the mathematical model also depends on the Strouhal number St, the amplitude of the lift coefficient obtained for a fixed cylinder $\hat{C}_L$ and the mean drag coefficient (also for a fixed cylinder) C_D. Following the wake-oscillator model proposed in [6], the loads due to the fluid-structure interaction can be obtained by coupling a van der Pol equation to the structural oscillator by means of empirically calibrated parameters, namely A_y and ϵ_y. Using this approach, the mathematical model is governed by the following dimensionless mathematical model:

$$\ddot{y}+2\zeta_y\dot{y}+y=\frac{1}{2\pi^3}\frac{U_r^2}{(m^*+C_a)}\sqrt{1+\left(\frac{2\pi\dot{y}}{U_r}\right)^2}\left(\frac{q_y}{\hat{q}_y}\hat{C}_L-C_D\frac{2\pi\dot{y}}{U_r}\right)=\varepsilon\left[2\sin\theta\dot{\eta}\dot{\theta}-\ddot{\eta}\cos\theta+(\hat{r}+\eta)\frac{d}{d\tau}(\dot{\theta}\sin\theta)\right] \tag{2}$$

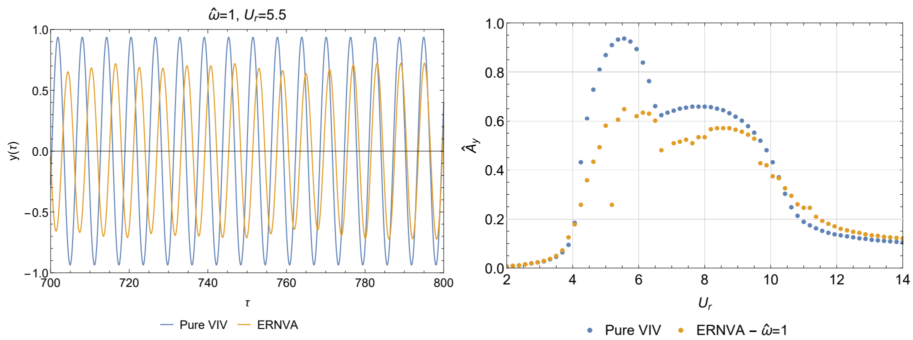
$$(\hat{r}+\eta)^2\ddot{\theta}+(2\zeta_\theta\hat{r}^2+2(\hat{r}+\eta)\dot{\eta})\dot{\theta}=(r+\eta)\ddot{y}\sin\theta \tag{3}$$

$$\ddot{\eta}+2\zeta_r\hat{\omega}\dot{\eta}+(\hat{\omega}^2-\dot{\theta}^2)\eta=\hat{r}\dot{\theta}^2-\ddot{y}\cos\theta \tag{4}$$

$$\ddot{q}_y+\epsilon_y StU_r(q_y^2-1)\dot{q}_y+(StU_r)^2q_y=A_y\ddot{y} \tag{5}$$

where $\varepsilon=(\hat{m}m^*)/(m^*+C_a)$. Equations 2 - 5 are numerically integrated using the Mathematica® NDSolve function during $\tau_{max}=1000$. For the sake of limitation on the number of pages of this extended abstract, just one set of parameters is tested. The chosen parameters for the cylinder and the ERNVA are $m^*=2.6$, $\zeta_y=0.0001$, $\zeta_r=\zeta_\theta=0.10$, $\hat{\omega}=1$, $\hat{r}=0.5$ and $\hat{m}=0.07$. The investigated reduced velocity is $U_r=5.5$, a favorable scenario for VIV. Only two non-trivial initial conditions are considered, namely, $\theta(0)=\pi/6$ and $q_y(0)=0.10$.

Fig. 2(a) shows two displacement time-histories $y(\tau)$, one of them being labeled as "Pure VIV" and corresponding to the condition in which no suppressor is attached to the cylinder. From this figure, one clearly notice that the presence of the ERNVA decreased the characteristic oscillation amplitude from $\hat{A}_y\approx 0.94$ ("Pure VIV" case) to $\hat{A}_y\approx 0.65$, a reduction close to 33%. The complete curve of characteristic oscillation amplitude as a function of reduced velocity is depicted in Fig. 2(b), which shows that the ERNVA is able to suppress VIV for a certain range of reduced velocities corresponding to the lock-in.



(a) $y(\tau)$. (b) $\hat{A}_y(U_r)$.

Conclusions

This extended abstract presented a numerical investigation on passive suppression of VIV using an elastic rotative non-linear vibration absorber (ERNVA). Considering the ERNVA mass equal to 7% of the total oscillating mass, it was found that a decrease in the characteristic oscillation amplitude has been achieved, specially close to $U_r=5.5$. In the full paper and at the conference, more comprehensive studies will be presented.

Acknowledgments

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