International Journal of Non-Linear Mechanics 81 (2016) 83–94

Contents lists available at ScienceDirect

International Journal of Non-Linear Mechanics

journal homepage: www.elsevier.com/locate/nlm

Usefulness of passive non-linear energy sinks in controlling galloping vibrations

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ARTICLE INFO

Article history:
Received 29 August 2015
Received in revised form
12 January 2016
Accepted 12 January 2016
Available online 18 January 2016

Keywords:
Non-linear energy sink
Galloping
Normal form
Hopf bifurcation

ABSTRACT

The suppression of vibration amplitudes of an elastically-mounted square prism subjected to galloping oscillations by using a non-linear energy sink is investigated. The non-linear energy sink consists of a secondary system with linear damping and non-linear stiffness. A representative model that couples the transverse displacement of the square prism and the non-linear energy sink is constructed. A linear analysis is performed to determine the impacts of the non-linear energy sink parameters (mass, damping, and stiffness) on the coupled frequency and onset speed of galloping. It is demonstrated that increasing the damping of the non-linear energy sink can result in a significant increase in the onset speed of galloping. Then, the normal form of the Hopf bifurcation is derived to identify the type of instability and to determine the effects of the non-linear energy sink stiffness on the performance of the aeroelastic system near the bifurcation. The results show that the non-linear energy sink can be efficiently implemented to significantly reduce the galloping amplitude of the square prism. It is also shown that the multiple stable responses of the coupled aeroelastic system are obtained as well as the periodic responses, which are dependent on the considered non-linear energy sink parameters.

1. Introduction

The issue on flow-induced vibrations (FIV) of structures has received a great research interest during the past decades due to its wide occurrence in many engineering fields including power transmissions, civil engineering, aerospace industry, and undersea technology [1–5]. In terms of engineering applications, FIV can be potentially utilized for energy harvesting to power the operation of micro-electronic devices [6–10]. On the other hand, FIV has potential hazards to the safety of engineering structures due to the resulted complex and diverse dynamic responses [11–15]. Therefore, the prediction and suppression of flow-induced vibrations of structures have been always attractive and of interest for many researchers in the last few years [16–19].

In general, structures in contact with cross flows mainly tend to subject flow-induced instabilities including vortex-induced vibrations (VIV), galloping, and flutter. Reducing or controlling such vibrations of structures has been widely explored by using active and passive approaches to enhance the stability, performance, and lifetime of structures. The feedback control methods, as a significant kind of active controls, were frequently applied in suppressing structural oscillations. Baz and Ro [20] proposed a velocity feedback controller to suppress VIV of a flexible circular cylinder. This scheme was effectively in the degradation of the single-mode vibration in the resonance region. Afterwards, Mehmood et al. [21] utilized linear and non-linear velocity feedback controllers to reduce the vortex-induced oscillations of an elastically-mounted rigid circular cylinder in the lock-in regime. Their results showed that the non-linear feedback force controller is more efficient than its linear velocity feedback counterpart. Recently, the time delay feedback controller was introduced by Wang et al. [22] to control the aeroelastic galloping responses of an elastically mounted square prism. Only the linear time delay was applied in their control strategy. Then, Dai et al. [23] improved the time delay feedback controller to investigate the effectiveness of combined linear and non-linear time delay feedback controls to suppress high amplitude oscillations of an elastically-mounted square cylinder undergoing galloping oscillations. Furthermore, some other significant researches on active controls such as acoustic feedback [24,25], flow blowing and suction [26], and adaptive control strategies [27–29] were proposed to control or suppress the flow-induced vibrations of structures.

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http://dx.doi.org/10.1016/j.ijnonlinmec.2016.01.007

On the other hand, several significant research studies on passive control techniques for controlling flow-induced vibrations were proposed and developed. Walshe and Wootton [30] explored passive control mechanisms of VIVs which utilizes the wire ropes to increase the natural frequency and damping of the structure. An experimental investigation has been carried out by Owen at al. [31] for suppressing vortex-induced vibrations of a circular cylinder. In their experiment, the hemispherical bumps were added on the cylinder to reduce the vortex shedding and hence the oscillations of structures. Recently, Quadrante and Nishi [32] studied the effects of tripping wires on the vibration of a circular cylinder. They found that the flow-induced vibrations can be increased or suppressed, which are associated with the placed position of tripping wires. As another important passive FIVs control, the concept of vibration suppression with non-linear energy sinks (NES) were proposed in recent years and significantly contributed by many researchers [33–43]. Tumkur et al. [38] and Mehmood et al. [39] added a non-linear energy sink (NES) that has a substantial non-linear stiffness to the VIV system. A clear reduction in the oscillation amplitude of the cylinder for different NES parameters was obtained. Moreover, by considering a two-degree-of-freedom long span bridge model, Vaurigaud et al. [40,41] investigated analytically and numerically the influence of including an NES on the response of an aeroelastic system.

In this research study, our main objective is to investigate the possibility of suppressing/controlling the vibration amplitude of a galloping system through using a non-linear energy sink (NES). To this end, the governing equations for modeling the flow-induced vibrations system coupled with the NES are constructed in Section 2. Linear analysis is performed in Section 3 to investigate the effects of NES parameters on the onset speed of galloping and natural frequency. In Sections 4 and 5, the non-linear normal form is derived in order to characterize the type of instability (supercritical or subcritical) and to determine the effects of linear and non-linear NES parameters on the performance of coupled system near bifurcation. Summary and conclusions are presented in Section 6.

2. Modeling of the galloping system with NES

The schematic of aeroelastic system, as shown in Fig. 1(a), is an elastically-mounted rigid square prism which is subjected to galloping vibrations in cross-flow direction. In order to enhance the stability of such an aeroelastic system, a NES system is introduced for suppressing the vibration amplitude of the prism, as seen in Fig. 1(b). It is noted that the introduced NES is placed inside the prism, according to Refs. [38,39], the governing equations of an elastically-mounted rigid square prism coupled with a NES

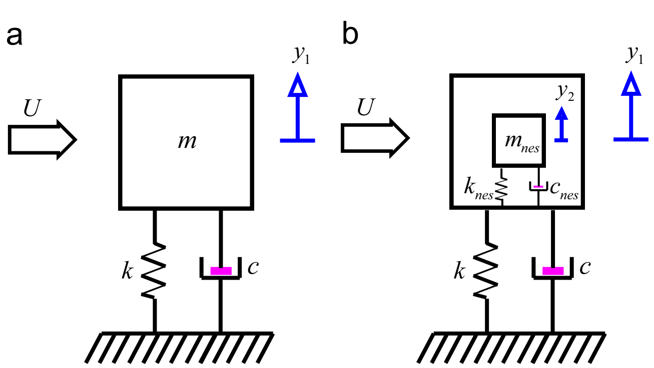

Fig. 1. Schematic of the square prism subjected to galloping vibrations (a): without NES; (b): with NES.

subjected to external fluid forces are given by:

$$(m-m_{nes})\ddot{y}_1+c\dot{y}_1+ky_1+c_{nes}(\dot{y}_1-\dot{y}_2)+k_{nes}(y_1-y_2)^3=f_L(y_1) \quad (1)$$

$$m_{nes}\ddot{y}_2+c_{nes}(\dot{y}_2-\dot{y}_1)+k_{nes}(y_2-y_1)^3=0 \quad (2)$$

where m and m_{nes} are, respectively, the mass of the cylinder and NES per unit length; c and k denote the structural damping and stiffness, respectively; c_{nes} and k_{nes} represent the damping and stiffness of the NES. It is stressed that the mass of prism is reduced in Eq. (1) so as to ensure no mass addition because of the added NES. f_L is used to indicate the time-dependent external force exerted by the fluid flow on the square prism. It follows from Belvins [2] and Barrero-Gil et al. [6] that the quasi-steady hypothesis is utilized to describe the aerodynamic force. Based on this assumption, therefore, the galloping force per unit length of a symmetric cross-section f_L can be expressed as:

$$f_L(y_1)=\frac{1}{2}\rho_a DU^2\sum_{n=1,2..}a_n\left(\frac{\dot{y}_1}{U}\right)^n \quad (3)$$

in which ρ_a is the density of the fluid, D is the effective diameter of square prism, and U denotes the fluid speed. In addition, the natural frequency of this aeroelastic system is much smaller than the vortex-shedding frequency. According to Blevins [2] and Barrero-Gil et al. [6], for high Reynolds number, it is accurate enough to characteristic the galloping force f_L by using the first three terms. As to the empirical parameter $a_n(n=1, 2, 3)$, which is the related with shape sections and determined by the experiments [44].

Using the diameter of the prism D as length scale, we introduce the following non-dimensional quantities:

$$w_1=\frac{y_1}{D},\ w_2=\frac{y_2}{D},\ \beta=\frac{m_{nes}}{m},\ \xi=\frac{c}{2\sqrt{km}},\ \xi_{nes}=\frac{c_{nes}}{2\sqrt{km}},\ \tilde{k}_{nes}=\frac{k_{nes}D^2}{m\omega_0^2} \quad (4)$$

Then, Eqs. (1) and (2) can be rewritten in the following form:

$$(1-\beta)\ddot{w}_1+2\xi\omega_0\dot{w}_1+\omega_0^2w_1+2\xi_{nes}\omega_0(\dot{w}_1-\dot{w}_2)+\tilde{k}_{nes}\omega_0^2(w_1-w_2)^3$$
$$=A_1\dot{w}_1+A_3\dot{w}_1^3 \quad (5)$$

$$\beta\ddot{w}_2+2\xi_{nes}\omega_0(\dot{w}_2-\dot{w}_1)+\tilde{k}_{nes}\omega_0^2(w_2-w_1)^3=0 \quad (6)$$

where $\omega_0^2=\frac{k}{m}$, $A_1=\frac{\rho_a DU}{2m}a_1$, $A_3=\frac{\rho_a D^3}{2mU}a_3$

3. Impacts of NES on the onset speed of galloping

In this section, a linear analysis is performed in order to study the effects of mass and damping of the NES on the coupled damping, coupled frequency, and onset speed of galloping of the aeroelastic system. The presence of the NES structure will affect the structural damping and frequency of the primary system. Due to this coupling, we name these modified damping and frequency by coupled damping and coupled frequency. The onset speed of galloping means the critical wind speed when the primary system loses stability and self-excited motions take place. To this end, the non-linear terms in Eqs. (5) and (6) are dropped. Hence, the equations of motion can be expressed as:

$$(1-\beta)\ddot{w}_1+(2\xi\omega_0+2\xi_{nes}\omega_0-A_1)\dot{w}_1-2\xi_{nes}\omega_0\dot{w}_2+\omega_0^2w_1=0 \quad (7)$$

$$\beta\ddot{w}_2+2\xi_{nes}\omega_0(\dot{w}_2-\dot{w}_1)=0 \quad (8)$$

Introduce the following state variables

$$X=\begin{bmatrix} w_1 & \dot{w}_1 & w_2 & \dot{w}_2\end{bmatrix}^T \quad (9)$$