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Optimization of a passive vibration absorber for a barrel using the genetic algorithm

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ABSTRACT

The non-linear vibrations of a barrel, induced by the interaction with a high-speed moving projectile, negatively affect the shooting accuracy of a weapon. This study presents a new method that determines the non-linear behavior of the barrel with a passive vibration absorber and optimizes the absorber using the genetic algorithm (GA). Since both the barrel geometry and its coupling with the absorber are non-linear, a new finite element method (FEM) approximation has been developed for the interaction of barrel and projectile and combined with the classical finite element method. The final coupled equation of motion of entire system has been solved by a step by step integration, and for minimum tip deflection of the barrel, a GA has been then used in order to optimize the some parameters of the absorber. The results of analyses of the proposed FEM model were compared, and a good agreement was seen with the existing literature. In another example, the FEM–GA integrated optimization procedure was also used for the optimization of a passive vibration absorber, and a more accurate result (0.5% better) was obtained when compared to the experimental study given in literature.

1. Introduction

The dynamic behavior of a structural system subjected to a moving mass is a key problem for defense systems, machine design and civil constructions, which have been studied by several researchers. For example, FEM solutions of some different kinds of moving mass problems are given by Esen (2011, 2013), Kahya (2012), Sharbati and Szyszkowski (2011), Wu, Whittaker, and Cartmell (2001). An analytical solution of moving-load-motion for Timoshenko beams can be found in Lee (1996). The importance of dynamic behavior of structural systems, subjected to a variable velocity load, has been increasing, and some researchers Dyniewicz and Bajer (2012), Lee (1996), Michaltsos (2002), Wang (2009) have studied the dynamic behavior of different kinds of beams under accelerating loads. Inertia effects of a moving mass still continue to be a point of interest for bridge dynamics, railroad design and other high-velocity delicate motion processes, and studies like Dehestani, Mofid, and Vafai (2009), Michaltsos and Kounadis (2001), Michaltsos, Sophianopoulos, and Kounadis (1996) are valuable in this respect. The accurate solution of moving mass problems has been facilitated via the usage of computers, and the study Bulut and Kelesoglu (2010) has compared numerical methods for response of beams, while the others (Ataei & Mohammadzade, 2010; Nikkhoo, Rofooei, & Shadnam, 2007) have studied the modal control of beams under the effect of a traveling mass.

The moving load problem is also vital for defense industries in order to provide an accurate shooting for Cannons, and the study Tawfik (2008) is valuable for the effects of an unbalanced mass of a projectile on the vibration of a barrel. Balla (2011) has studied the eight degree of freedom model of a weapon system with its body, and analysed the vibration of its barrel. In order to compare the results between modeling and actual test data, Alexander (2007) has modeled and analysed the projectile and barrel interaction dynamics, and then compared analysis results to the test data of 155 mm cannon. Researchers (Littlefield, Kathe, Messier, & Olsen, 1997, 2002) have studied the dynamics of barrels and proposed a muzzle-brake for reducing the tip-deflection of a 120 mm-cannon-barrel. It was reported that, the muzzle-brake, working as a passive vibration absorber, could reduce the deflection of the barrel by about half. As any moving load-structure interaction problem may be found in a lot of application fields, the studies Bathe (1982), Clough and Penzien (2003), Fryba (1999), Reddy (1984), Wilson (2002) can be considered valuable references for analytical and FEM solutions of systems affected by a moving mass.

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In order to get more efficient and economical solutions, artificially intelligent techniques (GA, fuzzy logic, neural network, etc.) have been applied to many complex engineering problems such as the damage identification of structures (Guo & Li, 2012; Miguel, Miguel, Kaminski, & Riera, 2012), vibration analysis and control (Ebersbach & Peng, 2008; Wang, Wang, & Chai, 2013), vibration absorber optimization (Torbati, Keane, Elliott, Brennan, & Rogers, 2003).

Optimization in engineering problems has always been of an important topic and interest in solving complex and nonlinear real-world problems like Zadeh, Salehpour, Jamali, and Haghgoo (2010). GA has turned out to be a powerful tool in the field of global optimization and has been applied successfully to real-world problems and exhibited, for a better search efficiency, compared to the traditional optimization algorithms by Chou, Wu, and Chen (2010). On the other hand, GA is the most popular method to optimize a structural system by determining fitness function and coding its parameters.

The studies given in existing literature are generally analytical methods and used for simplified cases of applications. In addition, there are also some experimental ones, yet they are expensive and time-consuming since they need many cases of trial and error works. However, the passive vibration absorbers are commonly used in industry in a lot of practical applications from machine tools to transportation vehicles. Moreover, especially for complex systems, the determination of the accurate dimensions and masses are impossible due to the non-linear behavior of the systems, and an experimental study may be inevitable. Vibrations of a barrel in gun systems is very vital and should be damped or reduced in order to provide better shooting accuracy. Besides, the determination of the non-linear dynamic behavior of such system is very hard, and studies in this area are limited, and those in this field are only for simple cases. For this reason, a more accurate and easy method is necessary in order to define the nonlinear behavior of a vibrating system such as gun barrels and the design and optimization of a passive vibration absorber for reducing the vibrations of main systems. This study presents a new optimization algorithm that combines the classical FEM and GA, and can also be used for the determination and analyses of the nonlinear vibrations of systems by using an algorithm that contains a step-by-step time integration method (Wilson, 2002). The FEM model was validated by being adapted to a simply-supported beam under a moving load, which had been studied widely in literature. Another numerical example given in this study is that the developed method has been used for investigating the dynamics of an anti-craft cannon barrel and optimization of a passive absorber for the barrel. Using the current method, a more accurate result (see Table 3) was obtained when compared to that of the experimental study.

2. Mathematical modeling

2.1. Finite element equation of a barrel element under an accelerating projectile

For the interaction of an accelerating projectile with the mass m_p and the barrel, a clamped-free cantilevered Euler–Bernoulli beam shown in Fig. 1 is considered. The projectile moves from the left end of barrel to the right end with a variable velocity $v_m(t)$, and a constant acceleration a_m. Fig. 2 shows mesh discretion of the barrel-beam under accelerating projectile and absorber, while Fig. 3 shows the sth beam element over which the projectile m_p passes at time t. The sth barrel element that interacts with projectile has three equivalent nodal forces as well as displacements at each nodal point.

When the barrel is in vibration, the transverse (z) and longitudinal (x) force components between the projectile and the beam, induced by the vibration and curvature of the deflected beam, are given by (Fryba, 1999):

$$f_z(x,t) = \left[m_p g - m_p \frac{\mathrm{d}^2 w_z(x_p,t)}{\mathrm{d}t^2}\right]\delta(x - x_p),$$
$$f_x(x,t) = m_p \frac{\mathrm{d}^2 w_x(x_p,t)}{\mathrm{d}t^2}\delta(x - x_p), \tag{1}$$

where $f_z(x, t)$ and $f_x(x, t)$ are the applied forces by the projectile at point x at time t. $\delta(x - x_p)$ and g are respectively the Dirac-delta function and gravitational acceleration. The time dependent position, velocity and acceleration of the projectile are:

$$x_p = x_0 + v_0 t + a_m t^2/2, \quad \mathrm{d}x_p/\mathrm{d}t = v_0 + a_m t, \quad \mathrm{d}^2 x_p/\mathrm{d}t^2 = a_m, \tag{2}$$

where, x_0 and v_0 are respectively the initial position and initial speed of the projectile when the time is zero.

The longitudinal and transverse equivalent nodal forces of the sth beam element under a lumped projectile mass are derived from Eq. (1) using the total differentiation with respect to the variable contact point ($x = x_m$), and they are as follows:

$$f_{si} = \phi_i m_p \frac{\partial^2 w_x(x,t)}{\partial t^2} \quad (i = 1,4),$$
$$f_{si} = \phi_i m_p \frac{\partial^2 w_z(x,t)}{\partial t^2} + 2\phi_i m_p v(t)\frac{\partial^2 w_z(x,t)}{\partial x \partial t} + \phi_i m_p v(t)^2 \frac{\partial^2 w_z(x,t)}{\partial x^2}$$
$$+ \phi_i m_p a_m \frac{\partial w_z(x,t)}{\partial x} + \phi_i m_p g \quad (i = 2,3,5,6), \tag{3}$$

where ϕ_i ($i = 1$–6) are interpolation functions of a beam element given by (Clough & Penzien, 2003):

$$\phi_1 = 1 - \frac{x}{l}, \quad \phi_2 = 1 - 3\left(\frac{x}{l}\right)^2 + 2\left(\frac{x}{l}\right)^3, \quad \phi_3 = x\left(1 - \frac{x}{l}\right)^2,$$
$$\phi_4 = \frac{x}{l}, \quad \phi_5 = 3\left(\frac{x}{l}\right)^2 - 2\left(\frac{x}{l}\right)^3, \quad \phi_6 = \frac{x^2}{l}\left(\frac{x}{l} - 1\right). \tag{4}$$

The other property matrices of a barrel element in Fig. 3, having both transverse and longitudinal nodal forces and deflections, are derived from the usage of the principle of virtual works. Introducing the principle of virtual displacements, and applying unit displacements at the nodal points and then equating the work done by the external forces to the work done on the internal forces: $W_E = W_I$ (Clough & Penzien, 2003; Smith & Griffiths, 2004) for a uniform beam segment using the interpolation functions of Eq. (4), the stiffness equation may be expressed by

$$\{f_{sis}\} = [K_e]\{u_{si}\}. \tag{5}$$

Any stiffness coefficient associated with beam flexure and axial displacements in Eq. (5) is as follows:

$$k_{ij} = \int_0^l EI(x)\phi_i''(x)\phi_j''(x)dx, \quad (i,j = 2,3,5,6),$$
$$k_{ij} = -\int_0^l EA\phi_i'(x)\phi_j'(x)dx, \quad (i,j = 1,4). \tag{6}$$

In the same manner, for the relation between nodal accelerations and resisting inertial forces, the elemental balance equation can be obtained. The mass coefficients are computed in this way, using the same interpolation functions which are used for calculating the stiffness coefficients. In the special case of a beam with uniformly distributed mass, the results are (Clough & Penzien, 2003):

$$\{f_{sib}\} = [M_e]\{\ddot{u}_{si}\}. \tag{7}$$