On FE Modeling of a Multibody Flexible System with Moving Parts acting as Controllers for Attenuation of Vibrations

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Abstract

Attenuation effects in a vibrating flexible system (main body) can be generated by imposing a precisely synchronized relative motion of one or more of its components (parts). The dynamic interaction between the main body and the moving parts triggers the Coriolis type forces that are capable of producing both attenuation and amplification effects in every cycle of the system's vibrations. In particular, by carefully controlling the relative motion's phase and frequency the attenuation can be made prevailing, which provides a unique means for actively reducing vibrations in systems with any internal/external damping almost absent[1]. A new finite element (FE) procedure is presented that handles interaction of a main body of any shape with several masses moving along arbitrary paths. The procedure is suitable for control purposes, it treats the moving part as a controller (with its imposed motion as input and the system's response as output) to generate its motion patterns resulting in attenuation.

The 'standard' FE analyses (such as used in vehicle-bridge or vehicle-rail tract simulations involving typically only constant, i.e. not controlled, relative velocities [2]) are essentially inapplicable to such a problem, mostly because it requires imposing too complex geometrical constraints on the relative motion of the components considered.

In our procedure the main body (Figure 1a) can be meshed by any elements appropriate to simulating its vibrations with sufficient accuracy. A controlled part (of mass m) is then forced to move along a prescribed path relative to the system. The path itself is modeled by a guiding beam attached to the vibrating body at nodes (Figure 1b). The properties of the beam represent either a real guiding rail or they may be fictitious. The moving mass interacts with the system via the beam, and at a particular instant only one beam element is affected by it. This element, referred to as the composite element, and the forces related to the relative motion (referred to as the Coriolis forces), are indicated in Figure 1c.

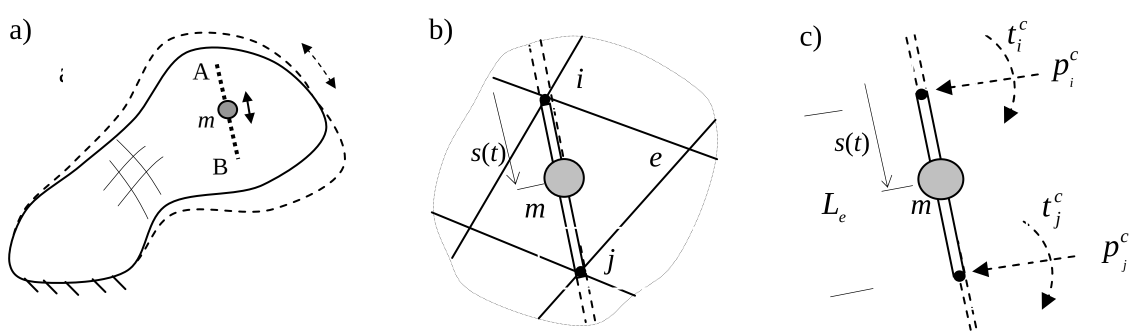

Figure 1: The vibrating system (a) the guiding beam (b), and the composite element (c)

Due to the relative motion the mass matrix of the composite beam is time-dependent. This makes the mass matrix for the whole system also time-dependent that substantially complicates the analysis.

As shown in [3], in order to identify directly the Coriolis forces in the composite element, the inertia forces for the composite element are considered in the form:

$$\frac{d}{dt}[\mathbf{M}_e(t)\dot{\mathbf{u}}_e] = \mathbf{M}_e(t)\ddot{\mathbf{u}}_e + \mathbf{C}_e\dot{\mathbf{u}}_e = \mathbf{M}_e(t)\ddot{\mathbf{u}}_e + \mathbf{f}_e^c \tag{1}$$

where $\mathbf{C}_s = \frac{d\mathbf{M}_e}{dt} = m\dot{s}\frac{\partial}{\partial s}(\mathbf{N}^T\mathbf{N})$ can be treated as an 'instantaneous' damping matrix (where **N=N**($s(t)$) are the values of the beam's shape functions at the current mass location). The term $\mathbf{f}_e^c = \mathbf{C}_e\dot{\mathbf{u}}_e$ defines the nodal Coriolis forces indicated in Fig 1c. The matrix $\mathbf{C}_s$ and vector $\mathbf{f}_e^e$ depend explicitly on the current relative velocity $\dot{s}$ of the mass and implicitly on its current position between the nodes of the element (the position is hidden in functions $\mathbf{N}$).

Formula (1) permits writing the element equation in the following forms (in order to concentrate on the active attenuation any passive damping is omitted):

$$\mathbf{M}_e(t)\ddot{\mathbf{u}}_e + \mathbf{C}_e\dot{\mathbf{u}}_e + \mathbf{K}_e\mathbf{u}_e = \mathbf{F}_e \quad \text{or} \quad \mathbf{M}_e(t)\ddot{\mathbf{u}}_e + \mathbf{K}_e\mathbf{u}_e = \mathbf{F}_e - \mathbf{f}_e^c \tag{2}$$

The first form indicates how the relative motion relates to periods of attenuation ($C_e^{ii} > 0$ if $\dot{s} > 0)$ and to periods of amplification ($C_e^{ii} < 0$ if $\dot{s} < 0)$, while the second form is more suitable for the practical simulation due to the fact that all the LHS terms can be routinely handled by typical FE software (in our case by ANSYS), in which $\mathbf{f}_e^c$ on the RHS can be easily calculated and added at each time step of the integration procedure. The system's attenuation is achieved by controlling forces $\mathbf{f}_e^c$ that are to be maximized in the periods of attenuation and minimized in the periods of amplification. The control scheme and the implementation and accuracy of the procedure will be discussed in details.

As illustration the test case of using relative motion of two small masses to attenuate vibrations of a frame is presented in Figure 2. For this particular case an effective active damping ratio of about 2.7% was generated.

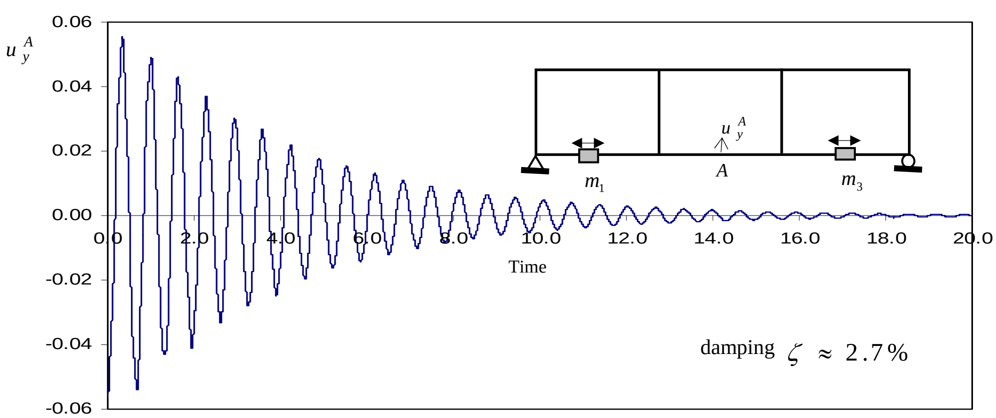

Figure 2: The response of vibrating frame controlled by motion of two masses.

References

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