PASSIVE CONTROL APPLICATION USING TMD IN BRIDGE-VEHICLE INTERACTION

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Abstract— In recent years, several control devices have been developed to control the vibrations of the structures. Between these innovative control devices, we can distinguish three different approaches: passive, active and semi-active control; to these we can add a fourth which is the hybrid control. Among these devices, Tuned Mass Dampers (TMD) are increasingly used in different fields of application in civil engineering, such as buildings and bridge. In this paper, an integration of a passive control system based on Tuned Mass Dampers in the coupled system of vehicle-bridge interaction is presented, using a formulation of the motion equations by the Finite Element Method(FEM) of a simply supported beam model of Timoshenko attached with (n) TMD fixed on a half-car model. In the resolution methods we applied a numerical simulation based on the MATLAB/Simulink.The numerical simulation results of the controlled system will be compared with those of the uncontrolled structure.

Index Terms— Bridge-vehicle interaction, Dynamic load, Passive control, Tuned mass damper, Vibration suppression.

I. INTRODUCTION

Recently, the control of structural vibrations subjected to a dynamic loads have interested a several researchers. The control devices have been investigated, tested and developed. Four different approaches of control can be cited: passive, active and semi-active and hybrid control. In civil engineering especially in bridge one of the most used device is the tuned mass dampers. The research efforts established by [1-3] are among the earliest studies of application of this device to different mechanical systems [4]. The aspect of the TMD depends mainly on the parameters as the material mass (example: concrete or steel), the mass shape's, the available space for installation, the fixation possibilityof the main structure and the structure of the adjacent members, desired frequency and damping as well as architectural restrictions [5].

Passive control consists in superimposing a device on the structure which can modified the rigidity or the damping of the structural system without requiring an external energy source and without introducing energy for its operation. Contrariwise, the active control visedto impose a force or a displacement at a selected points of the system to be controlled depending inparticularlyon the measured state or the history state of the system. Therefore, this type of control requires a large source of external energy to operate the actuators that provide the control forces determined using the measurements from the sensors of the excitations and / or the structural responses.

Semi-active control combines the main features of active and passive systems. These systems require a low source of energy to modify the mechanical property of the control device. Thus, the control system is considered as hybrid one if it uses a combination of passive and active control system.

In this paper, an integration of a passive control system based on tuned mass dampers in the coupled system of the vehicle bridge interaction. The motion equations of the system is formulatedbasing on the Timoshenko model of a simply supported beam. The control of the system is assured with n fixed TMD on a half-vehicle model. The numerical simulation results shown the effectiveness and the robustness of the passive control using TMD in which the results of the controlled structuresimulations and the uncontrolled structure simulations are compared.

II. THE MOTION EQUATION OF THE ATTACHED TMD TO THE TIMOSHENKO BEAM MODEL

A simply supported beam with n-TMD as shown in Figure.1 is used to set up the equation of motion for the general case. Where x_n, M_{TMD_n}, C_{TMD_n} and K_{TMD_n} are defined as the position, mass, damping and stiffness of the n^{th} TMD respectively.

According to the principle of Hamilton used to obtain the motion equations of the model. The total kinetic energy of the system can be defined as follows

$$T = \frac{1}{2}\left\{\int_0^L \rho \dot{y}^2(x,t)dx + \int_0^L \rho I(x)\dot{\Psi}^2(x,t)dx + M_{TMD_1}\dot{Z}_1^2 + \cdots + M_{TMD_n}\dot{Z}_n^2\right\} \quad (1)$$

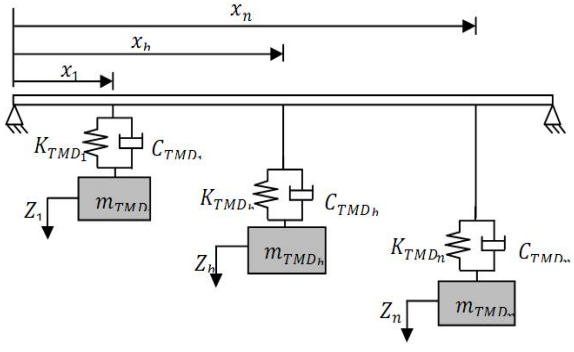


Fig. 1Simply supported beam attached with n-TMD

Therefore, the kinetic energy of the virtual system is described as follows

$$\delta T = \int_0^L \rho \ddot{y}(x,t)\frac{\partial \delta y}{\partial t}dx + \int_0^L \rho I(x)\ddot{\Psi}(x,t)\frac{\partial \delta \Psi}{\partial t}dx + M_{TMD_1}\ddot{Z}_1(t)\frac{d\delta Z_1}{dt} + \cdots + M_{TMD_n}\ddot{Z}_n(t)\frac{d\delta Z_n}{dt} \quad (2)$$

We can write

$$\int_{t1}^{t2}\delta T dt = -\int_{t1}^{t2}\left\{\int_0^L \rho \ddot{y}(x,t)dx\delta y + \int_0^L \rho I \ddot{\Psi}(x,t)dx\delta\Psi + M_{TMD_1}\ddot{Z}_1\delta Z_1 + \cdots + M_{TMD_n}\ddot{Z}_n\delta Z_n\right\}dt \quad (3)$$

The total potential energy of the system can also be written as

$$U = \frac{1}{2}\left\{\int_0^L EI\left(\frac{\partial \Psi}{\delta x}\right)^2 dx + \int_0^L k_s AG\left(\frac{\partial y}{\partial x} + \Psi\right)^2 dx + K_{TMD_1}(y(TMD_1,t) - Z_1)^2 K_{TMD_n}(y(TMD_n,t) - Z_n)^2\right\} \quad (4)$$

The virtual potential energy can be written as

$$\delta U = \int_0^L EI\frac{\partial \Psi}{\partial x}\frac{\partial \delta\Psi}{\partial x}dx + \int_0^L k_s AG\Psi\delta\Psi dx + \int_0^L k_s AG\frac{\partial y}{\partial x}\delta\Psi dx + \int_0^L k_s AG\frac{\partial \delta y}{\partial x}\Psi dx + \int_0^L k_s AG\frac{\partial y}{\partial x}\frac{\partial \delta y}{\partial x}dx + K_{TMD_1}(y_{TMD_1} - z_1)(\delta y_{TMD_1} - \delta z_1) + \cdots + K_{TMD_n}(y_{TMD_n} - z_n)(\delta y_{TMD_n} - \delta z_n) \quad (5)$$

The integration of (5) give

$$\int_{t1}^{t2}\delta U dt = \int_{t1}^{t2}\left\{\left(\int_0^L -EI\frac{\partial^2\Psi}{\partial x^2}dx + \int_0^L k_s AG\left(\Psi + \frac{\partial y}{\partial x}\right)dx\right)\delta\Psi - \left(\int_0^L k_s AG\left(\frac{\partial^2 y}{\partial x^2} + \frac{\partial \Psi}{\partial x}\right)dx\right)dy + K_{TMD_1}(y_{TMD_1} - z_1)(\delta y_{TMD_1} - \delta z_1) + \cdots + K_{TMD_n}(y_{TMD_n} - z_n)(\delta y_{TMD_n} - \delta z_n)\right\}dt \quad (6)$$

However, the virtual work in the system can be expressed as

$$\int_{t1}^{t2}\delta W_{nc}dt = \int_{t1}^{t2}\left\{-\int_0^L c\dot{y}dx\delta y - C_{TMD_1}(\dot{y}_{TMD_1} - \dot{z}_1)\delta(y_{TMD_1} - z_1) - \cdots - C_{TMD_n}(y_{TMD_n} - z_n)(\delta y_{TMD_n} - \delta z_n)\right\}dt \quad (7)$$

Substituting (3), (6) and (7) in the Hamilton's principal gives

$$0 = \int_{t_1}^{t_2}\left\{-\int_0^L \rho\ddot{y}(x,t)dx\delta y - \int_0^L \rho I\ddot{\Psi}(x,t)dx\delta\Psi - M_{TMD_1}\ddot{Z}_1\delta Z_1 - \cdots - M_{TMD_n}\ddot{Z}_n\delta Z_n + \left(\int_0^L EI\frac{\partial^2\Psi}{\partial x^2}dx - \int_0^L k_s AG\left(\Psi + \frac{\partial y}{\partial x}\right)dx\right)\delta\Psi + \left(\int_0^L k_s AG\left(\frac{\partial^2 y}{\partial x^2} + \frac{\partial\Psi}{\partial x}\right)dx\right)\delta y - \int_0^L K_{TMD_1}(y - z_1)\delta^*(y - y_{TMD_1})dx\delta y + \int_0^L K_{TMD_1}(y - z_1)\delta^*(y - y_{TMD_1})dx\delta z_1 - \cdots + \cdots - \int_0^L K_{TMD_n}(y - z_n)\delta^*(y - y_{TMD_n})dx\delta y - \int_0^L K_{TMD_n}(y - z_n)\delta^*(y - y_{TMD_n})dx\delta z_n - \int_0^L C_{TMD_1}(\dot{y} - \dot{z}_1)\delta^*(y - y_{TMD_1})dx\delta y + \int_0^L C_{TMD_1}(\dot{y} - \dot{z}_1)\delta^*(y - y_{TMD_1})dx\delta z_1 - \cdots + \cdots - \int_0^L C_{TMD_n}(\dot{y} - \dot{z}_n)\delta^*(y - y_{TMD_n})dx\delta y + \int_0^L C_{TMD_n}(\dot{y} - \dot{z}_n)\delta^*(y - y_{TMD_n})dx\delta z_n\right\}dt \quad (8)$$

Through, the dynamic of the system can described by (n+2) second order differential equations obtained by minimizing(δy, $\delta\Psi$ /, y_{TMD_1} , et ... δy_{TMD_n}) to 0.

The vertical beam deformation's can be written as

$$\int_0^L \rho\ddot{y}(x,t)dx - \left(\int_0^L k_s AG\left(\frac{\partial^2 y}{\partial x^2} + \frac{\partial\Psi}{\partial x}\right)dx\right) + \int_0^L c\dot{y}(x,t)dx + \int_0^L C_{TMD_1}(\dot{y} - \dot{z}_1)\delta^*(x - x_1)dx + \cdots + \int_0^L C_{TMD_n}(\dot{y} - \dot{z}_n)\delta^*(x - x_n)dx + \int_0^L K_{TMD_1}(y - z_1)\delta^*(x - x_1)dx + \cdots + \int_0^L K_{TMD_n}(y - z_n)\delta^*(x - x_n)dx = 0 \quad (9)$$

The orientation of the transversal section of the beam Ψ is defined by

$$\int_0^L \rho I\ddot{\Psi}(x,t)dx - \int_0^L EI\frac{\partial^2\Psi}{\partial x^2}dx + \int_0^L k_s AG\left(\Psi + \frac{\partial y}{\partial x}\right)dx = 0 \quad (10)$$

Finally, the equation of motion of the n^{th} attached TMD is

$$M_{TMD_h}\ddot{z}_h + \int_0^L C_{TMD_h}(\dot{z}_h - \dot{y})\delta^*(x - x_h)dx + \int_0^L K_{TMD_h}(z_h - y)\delta^*(x - x_h)dx = 0 \quad (11)$$

III. APPLICATION OF THE FEM TO THE TIMOSHENKO BEAM WITH ATTACHED TMD

The formulation of the considered system using Finite Element Method is obtained using the same procedure of [6]. Thus, the (10) and (11) can be used for an element with L length in FEM application. Where, the same linear interpolation of the vertical deformation y and the rotation around Z axis Ψ areconsidered. The beam properties and the interpolation functions are the same. The total motion equations of a beam element with attached n-TMD can be written using FEM by

$$[M_T]\{\ddot{A}(t)\} + [C_T]\{\dot{A}(t)\} + [K_T]\{A(t)\} = [F_T] \quad (12)$$

Where $A(t)$ is the nodal displacement vector given by

$$\{A(t)\} = \begin{Bmatrix} Y_1 \\ \Psi_1 \\ Y_2 \\ \Psi_2 \\ Z_1 \\ \vdots \\ Z_n \end{Bmatrix} \quad (13)$$

$[M_T]$,$[K_T]$and$[C_T]$ are the mass, stiffness and damping matrices respectively of the beam element with n-TMD attached. The assemblage of the matrices of the all classical beam elements and the beam elements with attached TMD appeared clearly in the general motion equations of the assembled system.

Therefore, the application of the FEM for a Timoshenko beam with under-fixed TMD on half-vehicle model is considered. Hereafter, the superposition of the motion equations of the beam element with attached n-TMD is used on a half-vehicle model and the final motion equations of the system is obtained and can be written [6]

$$[M]\{\ddot{X}(t)\} + [C(t)]\{\dot{X}(t)\} + [K(t)]\{X(t)\} = \{F(t)\} \quad (14)$$

Where $X(t)$ is the nodal displacement vector given by

$$\{X(t)\} = \begin{Bmatrix} Y_1 \\ \Psi_1 \\ Y_2 \\ \Psi_2 \\ y_s \\ J \\ y_{t1} \\ y_{t2} \\ Z_1 \\ \vdots \\ Z_n \end{Bmatrix} \quad (15)$$

Where $[M],[K]$and$[C]$ are the mass, stiffness and damping matrices respectively of the beam element with attached n-TMD traversed by half-vehicle model. Finally, the force's vector can be presented as

$$F = \begin{Bmatrix} \bar{\omega}_1 + \bar{\omega}\bar{\omega}_1 \\ 0 \\ \bar{\omega}_2 + \bar{\omega}\bar{\omega}_2 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \vdots \\ 0 \end{Bmatrix} \quad (16)$$

Where $\bar{\omega}_1$, $\bar{\omega}_2$, $\bar{\omega}\bar{\omega}_1$ and $\bar{\omega}\bar{\omega}_2$ are the vehicle weights applied from the first and second tires on the first and second node of the beam respectively.

The general equations of motion for the entire system of vehicle, TMD and beam can be obtained by assembling the matrices of all beam elements: the conventional beam elements joined with n-TMD and the beam element attached with the displacement of the vehicle and the n-TMD. After that, (12) can be solved using direct time integration of the Newmark β method.

IV. NUMERICAL RESULTS FOR THE QUARTER VEHICLE MODEL

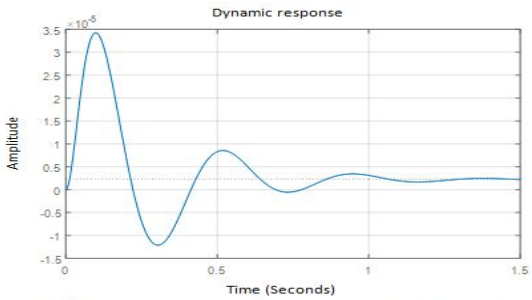


Fig. 2 Dynamic responses with m_1=87.15 Kg and m_2=897Kg

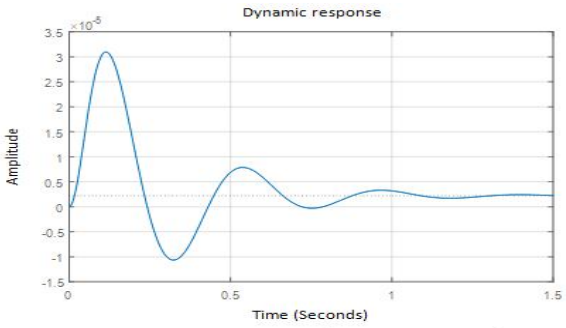


Fig. 3 Dynamic responseswith m_1=43.56Kg and m_2= 448.5 Kg

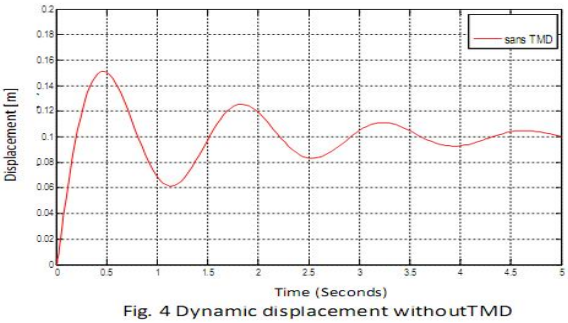


Fig. 4 Dynamic displacement withoutTMD

Figures (2) and (3) represented the amplitude variation function of time. The variation of amplitude is proportional with the weight variation. This variation of weight affected directly the dynamic response of the system.

V. STRUCTURAL RESPONSES WITHOUT TMD

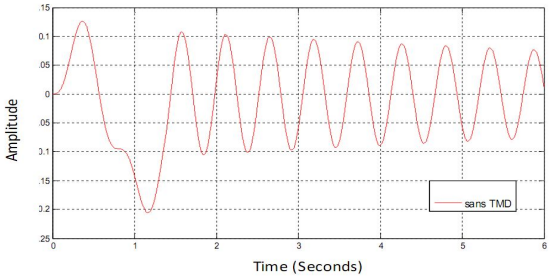


Fig. 5 The amplitude without TMD

The maximum vertical displacement of the middle point of the bridge is 1.13mm. The system is uncontrolled and supposing that the bridge have no damping. When the vehicle passed-on, the bridge

motion's is considered as a free vibration.

VI. TMD STRUCTURAL CONTROL RESPONSES

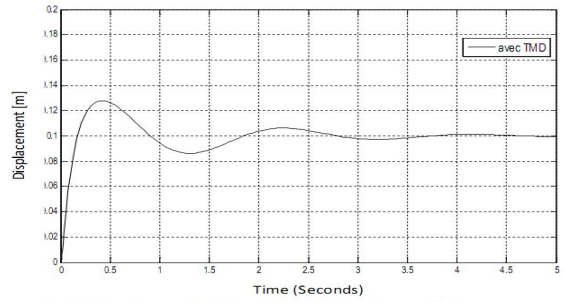


Fig. 6 Dynamic displacement of the controlled system

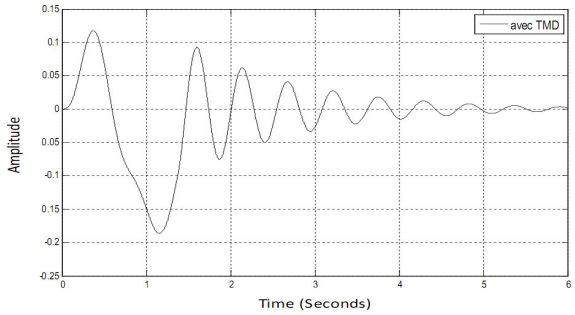


Fig. 7 The amplitude of the controlled system

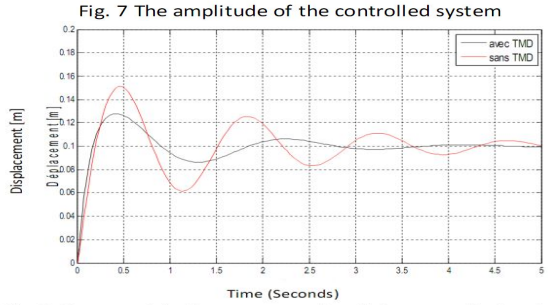


Fig. 8 Compareddisplacement results of the controlled and uncontrolled system

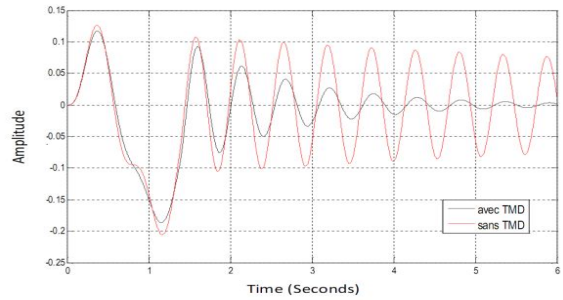


Fig. 9 Comparedof the amplitude results of the controlled and uncontrolled system

The integration of the passive control using a TMD controller revealed a decrease of displacement and the maximum amplitude. Once the vehicle crosses the bridge the response is depreciated due to the TMD depreciation.

From Figure 8 it can be seen that the TMD reduces the displacement at the beginning of the seconds with a percentage of 8% up to 60%.

CONCLUSION

The most widely used mechanical vibration damping devices in practice are the tuned mass dampers (TMDs) which are commonly used in various fields of civil engineering applications.For example, vibration damping of buildings, flexible structures such as footbridges and bridges.

The equations of motion of a half vehicle-beam coupled system with n-TMD are derived and transformed by the FEM using the Galerkin low-form approximation. Subsequently, using the superposition principle, the determination of the motion equations for the beam element are combined with the equations of motion for the beam element attached with (n) TMD under the half-vehicle model.

In this work we have created a simple model of a coupled bridge-vehicle system in MATLAB (by integrating FEM into MATLAB). The model is then used to study the influence of the parameters relevant to the bridge-vehicle vibrations validated by the numerical simulations results.

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