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Procedia Engineering 199 (2017) 290–295

Procedia Engineering

www.elsevier.com/locate/procedia

X International Conference on Structural Dynamics, EURODYN 2017

Fluid dynamic interaction between train and noise barriers on High-Speed-Lines

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Abstract

Noise barriers are lineside structures next to the railway track, subject to vibrations due to fluid-dynamic excitation induced by the train passage in High-Speed-Lines (HSLs). The train, travelling along the railway track, is immersed in a fluid, which increases its resistance as the speed of the train increases; the generated pressure field, with sinusoidal trend, is orthogonal to the barrier and excites its dynamic response, testing strength and fatigue resistance. This phenomenon becomes particularly significant for the HSLs trains, travelling at 300 kph speed, and should be evaluated to ensure the transport safety. The aim of the study is to focus on the dynamic response of existing noise barriers, with special regard to fatigue aspects, and proposes the introduction of special devices, Tuned-Mass-Dampers (TMDs), to place on the top of each column in order to reduce structural vibrations. The noise barrier is modeled as a generalized single-degree-of-freedom (SDOF) system. The pressure field induced by the train passage is modeled by a dynamic action function of the barrier height and geometry, of the railway geometry and the train speed. Two case studies are illustrated with columns 4 and 5 m high and concrete noise panels. The design of the auxiliary system, the TMD, is carried out as first tentative solution for reducing the structural vibrations and dynamic analysis on the barriers with and without the TMD shows the effectiveness of the control system to reduce the amplitude of motion and the number of cycles of vibration.

Peer-review under responsibility of the organizing committee of EURODYN 2017.

Keywords: noise barriers; fluid-dynamic interaction; dynamic analysis; tuned-mass-damper system.

1. Introduction

The vibration induced on the trackside structures generated by aerodynamic pressure produced by the train transit represents a topic worthy of attention [1-3]. The noise barriers, due to the repeated passage of the train along High-Speed-Lines (HSLs), undergo cyclic loading and can be affected by fatigue phenomena. This aspect is intended to

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Peer-review under responsibility of the organizing committee of EURODYN 2017.
10.1016/j.proeng.2017.09.035

raise importance in the future for increased train running speeds which imply higher pressure fields. Until the last decade, only few European countries focused on this aspect; all the evaluations were done in absence of national codes or specific requirements considering the dynamic effects induced by the train passage. Italferr, the engineering company of the Italian State Railways, has carried out several theoretical and experimental studies on the HSLs along with their increasing spread in Italy [4].

Tuned-Mass-Dampers (TMDs) are acknowledged systems that produce passive vibration mitigation. They can be linear [5] or nonlinear [6], conventional or non-conventional [7] and can include innovative control elements to improve their performances [8].

The aim of this study is to focus on the fluid-dynamics interaction between existing noise barriers and train on HSLs and to develop possible solutions to mitigate the effects produced on such structures. The paper illustrates dynamic analysis of noise barriers, modeled as a generalized single-degree of freedom (SDOF) system, subject to the aerodynamic pressure field induced by the train passage. A first tentative solution for reducing the noise barriers vibrations is proposed with the design of a TMD to place on the top of each barrier column.

2. Noise barriers on high-speed lines: description and critical aspects

In Italy, the current HSLs barriers are often the result of older structures designed and, for years, built along the ordinary lines generally characterized by speeds not exceeding 200 kph. These types of barriers are composed by HE steel profiles columns, spaced each 3 m, adequately constrained to the base through anchor bolts, Fig. 1 (a). Between the flanges of the profiles, noise panels are inserted. These panels are usually made of metal (aluminum, stainless steel or galvanized steel) or concrete, or transparent material (glass or PMMA), also in mixed combination, according to environmental aspects and costs.

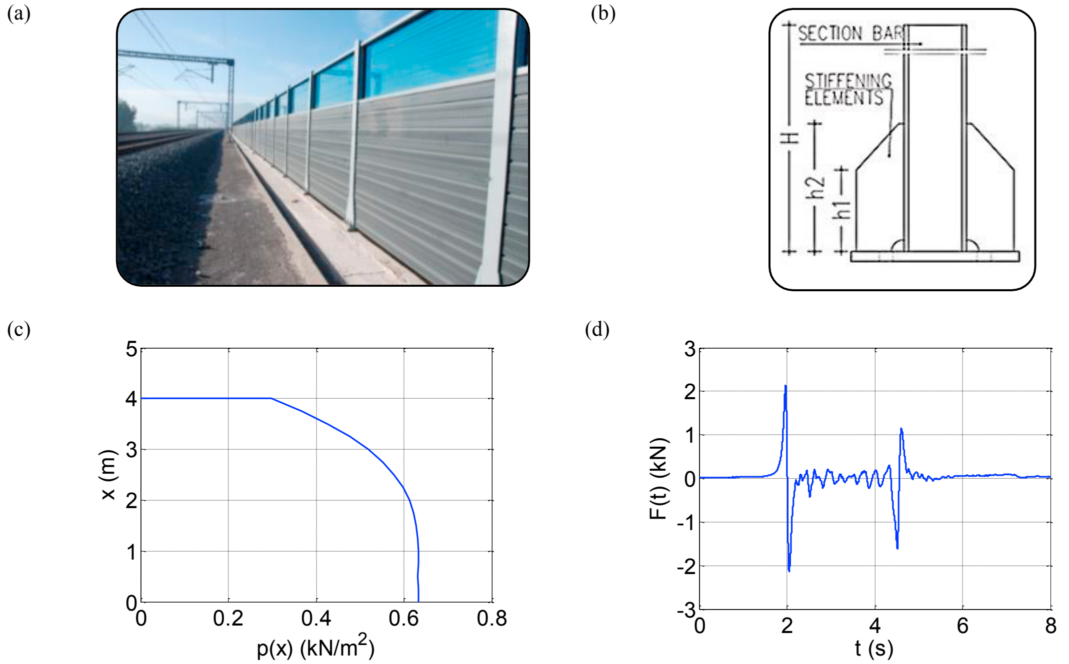

Fig. 1 (a) Noise barriers along Italian HSLs, (b) Geometrical features of the column, (c) Aerodynamic pressure along the barrier height (H= 4 m, s= 350 kph, d=4,3 m, i= 3 m) (d) Measured time history of the input to the SDOF system for a train speed s=350 kph.

The experimental trends obtained during testing campaigns on the noise barriers confirmed their dynamic interaction with the train passage. The input pressure dynamic signal is characterized by two main not symmetrical impulses corresponding to the entrance and exit of the train connected by oscillations of minor amplitude, Fig.1(d). Moreover, the signal: *i*) is roughly sinusoidal; *ii*) the number of sinusoids between the first wave (head locomotive) and the last wave (tail locomotive) is equal to the number of the train coaches (the oscillations are due to the presence of the junctions between wagons); *iii*) the distance between two subsequent peaks is a function of the speed of the travelling train.

The distribution of the maximum pressures along the height is not constant, with maximum in the low part, Fig.1(c). Whereas, by observing the behavior along the longitudinal direction, it emerges that the pressure wave propagates with equal speed to those of the train.

Concerning the structural response of the noise barriers, it emerged that the time-history develops on through a higher number of cycles compared to those of the input pressure; it highlights therefore the importance of the fatigue phenomenon induced on them.

3. Generalized single-degree-of freedom system

Columns of noise barriers are cantilever beams with distributed mass $m(x)$ and flexural rigidity $EI(x)$. Such systems, subject to dynamic action, can deflect in an infinity variety of shapes and, for exact analysis, they must be treated as an infinite degree of freedom system. Experimental tests and finite element analyses conducted on the barriers, shown that each column deflects independently from the others; moreover, the first vibrational mode is distant enough from the followings [1]. For this reason, each column, with the associated part of paneling, can be studied independently and modeled as a generalized SDOF system. It is possible to obtain approximate results by restricting the deflection of the beam $v(x,t)$ to a single shape function $\psi(x)$ that approximates the fundamental vibration mode:

$$v(x,t) = \psi(x)y(t) \tag{1}$$

where the generalized coordinate $y(t)$ is the deflection of the cantilever beam at a selected location, e.g. at the free end. Such model, even if synthetic, is adequate to study the dynamic interaction between the barrier and the train.
The shape function $\psi(x)$ must satisfy the displacement boundary conditions, here, it has been chosen:

$$\psi(x) = 1 - cos\left(\frac{\pi x}{2H}\right) \tag{2}$$

with *H* is the total length of the column.

3.1. Equation of motion and time history of the input

The equation of motion of the generalized SDOF system can be obtained by applying the principle of virtual work leading to:

$$\tilde{m}\ddot{y} + \tilde{k}y = \tilde{L}f(t) \tag{3}$$

with generalized mass $\tilde{m}$, stiffness $\tilde{k}$ and force $\tilde{L}$ evaluated as:

$$\begin{aligned} \tilde{m} &= \int_0^H m(x)[\psi(x)]^2\, dx \\ \tilde{k} &= \int_0^H EI(x)[\psi''(x)]^2\, dx \\ \tilde{L} &= \int_0^H q\,(x)\psi(x)dx \end{aligned} \tag{4}$$

The natural frequency of the SDOF system is $\omega_n = (\tilde{k}/\tilde{m})^{1/2}$ whereas $q(x)$ is the distributed load for unity of length applied to the column estimated as:

$$q(x) = p(x) \cdot i \tag{5}$$

i is the interaxle spacing between two columns and *p*(*x*) is the pressure along the height of the column induced by the passage of the train. It is expressed, [4], as a polynomial cubic function of the height *x* and depends on the train speed *s*, the rail-barrier distance *d* and the interaxle spacing, Fig. 1 (c). *f*(*t*) is the normalized time history related to the train passage: it varies in the abscissa with the train speed *s*. Figure 1 (d) shows the force $F(t) = \tilde{L}f(t)$ applied to the generalized SDOF system for a train speed s=350 kph.

Responses obtained from the generalized SDOF model are related to those of the barrier by applying Eq. (1). The maximum displacement of the column is expected at the top, i.e. for $x=H$, it is $v_{max} = \max\ [y\ (t)]$. The maximum bending moment $\mathcal{M}_{max}$ is obtained at x=0 and is evaluated as:

$$\mathcal{M}(0,t) = EI(x)\psi''(0)y(t)\ ,\ \mathcal{M}_{max} = max\ [\mathcal{M}\ (0,t)] \tag{6}$$

4. Case studies

Typical noise barriers high H=4 and 5 m of the HSLs are considered as case studies. The columns are made with industrial metal section beams and the paneling is realized in concrete. In order to have a one-dimensional problem, the paneling contribution is considered with a length equal to the interaxle spacing between two columns (i), moreover its damping effect is considered by assuming an equivalent damping factor of 2%. The total mass $m(x)$ is evaluated as the sum of the column $m_c(x)$ and the panelling $m_p(x)$ mass respectively. The elasticity modulus E is constant along the height. The moment of inertia is variable along the height from I_1 to I_2 according to the following:

$$\begin{cases} \quad I_1 & for\ x = 0 - h_1 \\ I(x) = \left(\frac{I_2 - I_1}{h_2 - h_1}\right) \cdot (x - h_1) + I_1 & for\ x = h_1 - h_2 \\ \quad I_2 & for\ x = h_2 - H \end{cases} \tag{7}$$

where I_1 and I_2 are the moment of inertia of the section beam with stiffening and without respectively. Table 1 shows the geometrical and inertial characteristics of the cases considered, Fig. 1 (b).

Table 1. Geometrical and inertial characteristics of the case studies.

H (m)	Section	$m_c(x)$ (kg/m)	$m_p(x)$ (kg/m)	$m(x)$ (kg/m)	E (MPa)	I_1 (m^4)	I_2 (m^4)	h_1 (m)	h_2 (m)
4	HEB 180	51.2	1070.3	1121.5	2.06 10^5	28.3 10^{-5}	3.831 10^{-5}	1.35	1.5
5	HEM 180	88.9	1070.3	1159.2	2.06 10^5	36.23 10^{-5}	7.483 10^{-5}	1.35	1.5

The dynamic properties of the generalized SDOF system are reported in Table 2. In order to match the period of the generalized SDOF with the same natural period obtained from a more refined FE analysis the mass and elasticity of the model have been reduced of the 15-20% and increased of the 15-20% respectively for H=5 and 4 m.

Table 2. Dynamic properties of the generalized SDOF system.

H (m)	Section	$\tilde{m}$ (kg)	$\tilde{k}$ (N/m)	$\tilde{L}$ (N)	$\tilde{c}$ (N sec/m)	T (sec)	f (Hz)
4	HEB 180	1220.7	1.5326 10^6	2134.1	1730.1	0.177	5.65
5	HEM 180	1511.5	9.7283 10^5	2174.0	1533.8	0.247	4.05

5. Design of the visco-elastic TMD

As a first tentative solution for reducing the noise barriers vibrations, a visco-elastic tuned mass damper (TMD) has been considered. The idea is to arrange an auxiliary mass (m_{TMD}) on the top of each column and to realize the link between the two masses with a passive isolation system. At this purpose, high damping rubber bearings (HDRB) have been chosen. For shear strains greater than 75-100 percent, experimental studies have shown that the shear modulus of the high-damping rubber can be considered constant [9], hence the behaviour of HDRB can be approximated as linear. In this study, a target shear strain of 100% has been assumed and a linear Kelvin-Voigt visco-elastic model is adopted to represent the mechanical behavior. The resultant 2-DOF mechanical system is reported in Fig. 3.

Explicit formulas for the optimum parameters of this kind of TMD attached to an undamped or damped structure are available in literature [5]. The optimal frequency ratio $\alpha_D = \omega_D/\omega_n$, defined as the ratio between the frequency of the TMD by itself and that of the main structure, and the optimal damping ratio ξ_D for a damped TMD are expressed as functions of the mass ratio $\mu = m_{TMD}/\tilde{m}$ and are given as:

$$\alpha_D = \frac{1}{1+\mu}\sqrt{1-\frac{\mu}{2}}, \qquad \xi_D = \sqrt{\frac{3\mu}{8(1+\mu)(1-\frac{\mu}{2})}} \tag{8}$$

For the noise barriers, the assumed mass ratio is μ=0.05 and the optimal parameters of the TMD are reported in Table 3, where the stiffness coefficient of the elastic force is evaluated as $k_{TMD} = \omega_D^2 m_{TMD}$ and the damping coefficient of the viscous force is $c_{TMD} = 2\xi_D\omega_D m_{TMD}$.

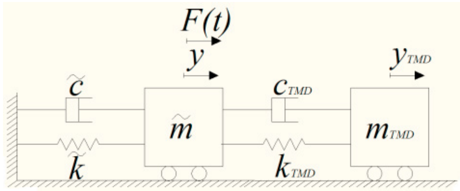

Fig. 3. Mechanical model.

Table 3. Optimal parameters of the TMD.

H (m)	μ	α_D	ξ_D	ω_D(rad/sec)	m_{TMD}(kg)	k_{TMD} (N/m)	c_{TMD} (N sec/m)
4	0.05	0.9404	0.1353	33.32	61.0	67768	550.34
5	0.05	0.9404	0.1353	23.86	75.5	43016	487.90

6. Dynamic analyses

Dynamic analysis on the generalized SDOF system equipped with the visco-elastic TMD designed in Section 5 subject to the input reported in Fig. 1 (d) are carried out. Resuts are reported in terms of the top displacement time history $y(t)$. Base bending moment $\mathcal{M}$ of the column can be derived from the top displacement time history by utilizing Eq. (6). The situation without applying the TMD on the top of the column, here called no-connected, NC case, is reported for comparison purposes to evaluate the effectiveness of the control system.

Figure 4 (a)-(b) depicts $y(t)$ for the two case studies of barriers examined. It is possible to observe after the first peak of the input, the strongly reduced amplification of the response along the whole time interval in the case of TMD equipment. As the relative motion between the column mass and the auxiliary mass is activated, the TMD starts to be effective to reduce the peaks. By evaluating the *rms* of the response the percentage of reduction comparing TMD with NC case is around the 40% and 46% for barriers 4 and 5 m high respectively. Smaller oscillation at the top of the coloumn will produce smaller bending moments with the consequence of reducing the fatigue phenomenon at the base of the column itself. Figure 4 (c)-(d) shows the relative displacement between the two masses $\Delta y = (y_{TMD} - y)$ for barriers 4 and 5 m high respectively. The maximum expected relative displacement represents the parameter to design the isolators, evaluated around 8 and 12 mm for barriers 4 and 5 m high respectively.

In order to give preliminary suggestions on the realization of this control system some approximate indications are furnished in the following. The TMD mass can be realized with a circular steel plate 153 mm thick, with diameter $\emptyset$ =255 and 283 mm for barriers 4 and 5 m high respectively. HDRB can be manufactured in circular shape with high-damping rubber having, when shear strain is 100 percent, shear modulus G from 0.4 to 1.4 MPa, for soft to hard blend and guaranteeing an equivalent viscous damping ratio of 10-15%. The rubber total thickness, according to the expected relative displacement, will be around t_r= 8 and 12 mm, for barriers 4 and 5 m high respectively.

7. Conclusions

The study investigated the response of the noise barriers subject to the dynamic pressure field induced by the train passage in HSLs. The input dynamic signal has distribution of the maximum pressures variable along the height with cubic function and depending on the train speed, rail-barrier distance and interaxle spacing. It was modelled with a time history characterized by two main not symmetrical impulses corresponding to the passage of the first and the last coach with oscillations of minor amplitude among them. The barrier column, modeled with a generalized SDOF system, showed an oscillatory response with high number of cycles denoting vulnerability to fatigue phenomena. The design of an auxiliary mass to be placed at the top of the column via HDRB, reproducing a TMD system, was carried out as a first tentative solution for reducing the noise barriers vibrations. Two case studies with columns 4 and 5 m high and concrete noise panels were illustrated. Dynamic analysis on these modeled barriers with the insertion of the TMD indicated the effectiveness of the control system to reduce the amplitude of motion and the number of cycles of vibration, which has benefits on stresses and fatigue phenomena.

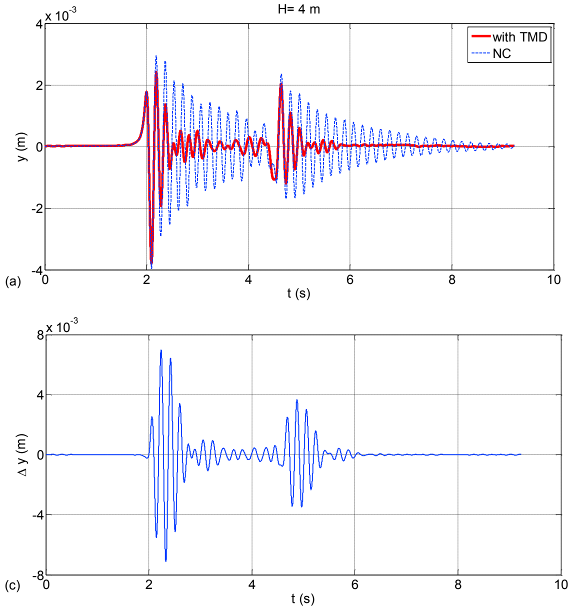

Fig. 4. Time history of the (a)-(b) top displacement without TMD (NC) and with TMD, (c)-(d) relative displacement between the two masses.

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