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Vibration mitigation of high-speed line switcher by tuned mass damper

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Abstract

This paper presents the practical implementation of passive tuned mass damper in the context of railway engineering and focuses on the vibrations experienced by the track equipment in the vicinity of switchers, specifically, the power switch machine that provides the force for opening and closing the switch. Such sensitive components are clamped to the sleepers and experience severe transverse displacement that damages the engine and results in the failure of the whole switcher. Finite element simulations were performed to identify kinematics likely to be responsible for damage and curative solutions were proposed and implemented. Particularly, a tuned mass damper was designed to provide an increased mitigation performance with respect to the targeted kinematic. This vibration absorber was designed according to the design rules available in the literature and its mechanical and geometrical properties were chosen to comply with in-situ constraints. The resulting device is a non-intrusive absorber, easily tunable depending on the installation site, and provides significant reduction the vibration level in the rather low frequency range.

Keywords: railway vibrations, vibration control, tuned mass damper.

1 Introduction

The issue concerned by this work falls within the general framework of vibrations induced by traffic. In this case, high-speed train traffic is a major source of vibration. This source has a very large frequency content that is transmitted to the structure by moving load and shocks. The emissions from this source are propagated through the connecting elements and the track equipment. This propagation medium plays a fundamental role in the transmission of the vibrations undergone. Finally, these induced vibrations repeatedly stress the track equipment over a wide and dense frequency spectrum. The control equipment thus undergoes significant accelerations for which they were not initially designed for, leading to the damage of fragile components. The damage caused by these vibrations is of different types, ranging from the development of micro-deformations, the increase in the density of dislocations, micro-cracks, up to the total rupture of the console at the fold.

Several solutions have been developed and implemented in the framework of traffic Induced vibration and railways engineering, to bring together vibroacoustic comfort, manufacturing constraint, and maintenance costs. According to the design rules of a damped tuned mass damper [1], reduction solutions acting on the excitation already exist and can be effectively combined with damping solutions for wheel [2] and rail [3]. Approved in 2007 in France, dynamic absorbers, also called tuned mass dampers, on rails demonstrated their ability to increase the rate of decrease of energy in the rail [4]. The combination of these two solutions, wheel and track absorbers, leads to an overall reduction in vibration levels and radiated noise, and are already deployed on light rails (metro, tram), the concept of vibration mitigation in a specific frequency range has been studied and integrated to wheels and rails [5, 6, 7,]. The design might also be improved using advanced design rules [8, 9], combination of multiple dampers [9, 10], or composite materials to achieve weight reduction while enhancing damping features.

The structure under study is depicted in Figure 1. It consists in a steel support that holds the switch engine. The switch engine slides the rod and the switch points and appears to be a fragile component sensitive to vibrations induced by high speed trains. The purpose of this work is to suggest a practical application of a tuned mass damper to prevent damage of the whole structure supporting engines actuating the switchers.

Figure 1 Picture of support and switch engine box actuating the point blade, clamped to sleepers and partially supported by ballast

2 Methods

This section describes the sequence of processes from diagnosis to in-situ implementation of the absorber.

Firstly, analysis and diagnosis of the vibratory state of the facilities were carried out. Particularly, an accurate experimental modal analysis of the structure resulted in a numerical model. The material properties were adjusted from a modal analysis of each component of the assembly. Introducing clamped boundary conditions to mimic the attachment to the sleepers, this stage results in a full FEM model to evaluate the effect of different treatments. This part was supplemented by in-situ measurements. According to the clamped conditions of the structure, its first natural mode exhibits a maximum transverse displacement on the engine box side, as shown in Figure 2. From this diagnosis, the critical mode to be targeted is the first cantilever mode of the platform.

Secondly, the proposal and recommendations of curative solutions consisted of a hybrid solution. As a first solution, reinforcement brackets were designed in such a way these are welded at the fold forming a right angle. The purpose of this is to mitigate the vibration level over the whole frequency range and stiffen the assembly. From the finite element model, it is verified that introducing such brackets stiffens the structure and leads to a shift of the first eigenfrequency from 22 Hz to 46 Hz. However, these brackets do not specifically target the first eigenmode. As a complementary solution, a tuned mass damper is designed according to [1], For the TMD design, the modal mass of the primary structure is that of the first cantilever mode, and the mass ratio between the modal mass of primary structure versus the total mass of the absorber is set to 10%. The resulting absorber consists in a cantilever beam with tip mass attached to free extremity, whose weight is about 6 kg. The position and weight of the mass can be adjusted in order to reach the desired frequency. This treatment is presented as hybrid because it brings an improvement in terms of overall vibration level with the reinforcements, and a local treatment of the desired frequency with the absorber.

The next section provides numerical and experimental results on each aspect of the work flow presented above.

3 Results

A preliminary modal analysis performed on the FEM model leads to the eigen frequencies and modes shapes of the structure, as depicted in Figure 2. It is outlined that due to manufacturing conditions and installatation configurations, the first eigenfrequency measured experimentally vary by a few hertz. However, this model is accurate enough to predict the eigen frequencies. From this computations, two solutions were implemented.

A first treatment consists in introducing stiffners located at the fold angle of the structure. This results in a global decrease of the vibration level over the whole frequency range, as well as in a shifting of all eigen modes in high frequencies. This effect is observed in the ideal numerical case, as shown in Figure 2.

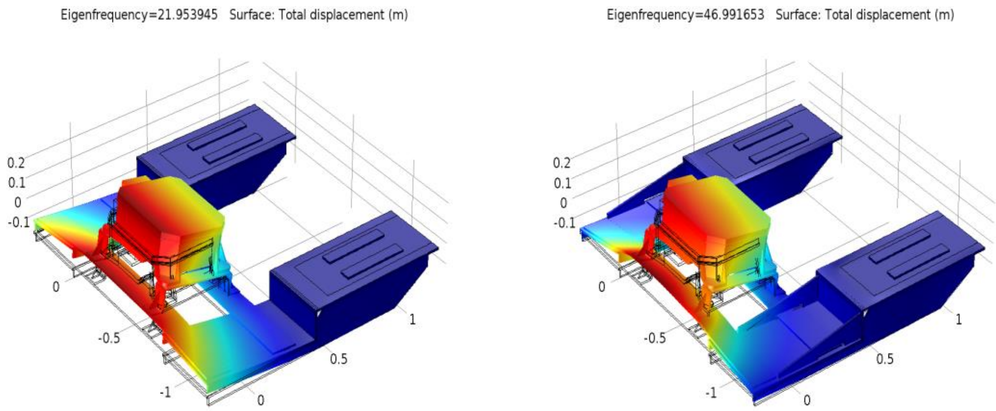

Figure 2 : Mode shape associated with the first eigen mode in original configuration : without stiffners (left), after welding of stiffners (right)

It is also verified from in-situ measurements of frequency response function. These were obtained by exciting the structure using an shock hammer, and mesuring the resulting acceleration. This measurement is performed on the original structure without stiffners, and after welding of stiffners. The comparison is represented in depicted in Figure 2 and shows the frequency shift due to the stiffening effect. This frequency shift is not as large as that predicted by the numerical model because of the uncertainties on the embedding, contacts between the components, and the effect of the ballast.

A second narrow band vibration control strategy is suggested by introducing a tuned mass damper. Following the general design rules derived in [1], a tuned mass damper was designed. Its frequency range is centered around the first eigenfrequency of the newly stiffened structure. Its mass is 10% of the modal mass associated with the first eigenmode. The damper is attached at the free extremity of the structure, where the transverse displacement is maximum. It can be tuned so that it is effective on an interval on approximately fifteen hertz.

This tuning ability make the absorber adapted to real environnement since the natural frequency of the structure can vary.

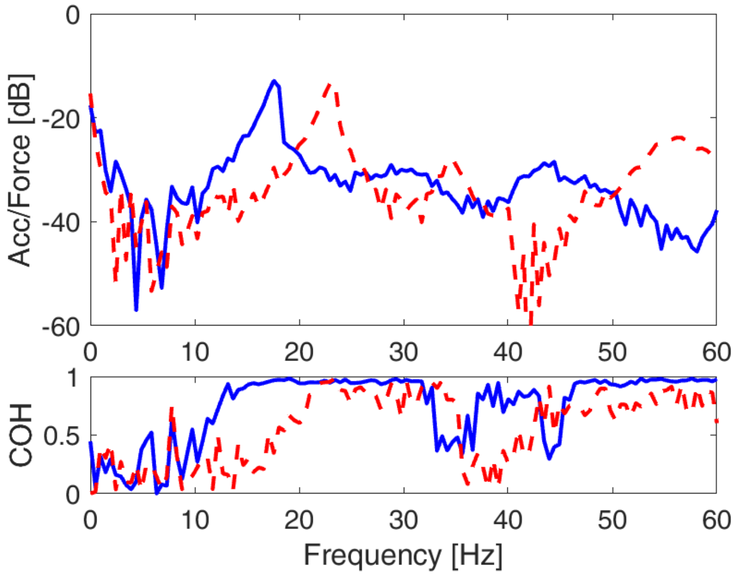

Figure 3 Frequency response measured at free extremity of the deck original configuration without stiffners (-) , and with stiffners at fold (- -)

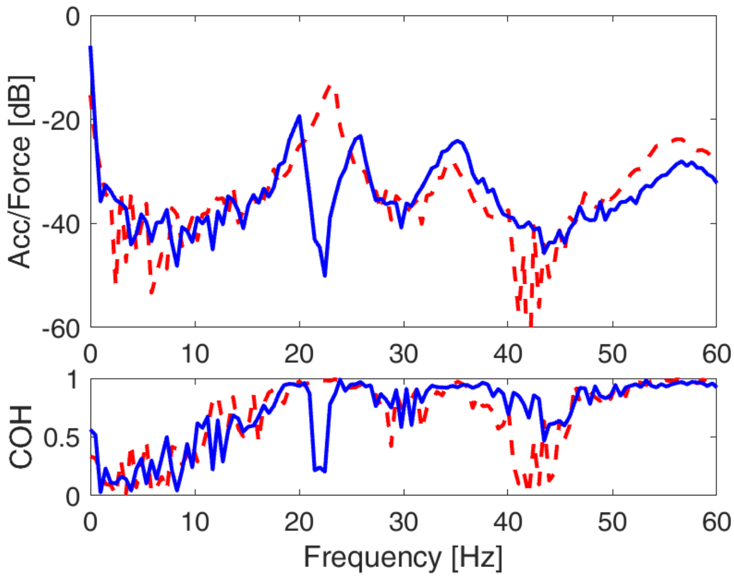

Figure 4 : Frequency response measured at free extremity of the deck without absorber (- -), and with tuned mass damper (-)

The typical response of the tuned mass damper is illustrated in Figure 4, that is the experimental frequency response measured between an accelerometer located at the free extremity of the structure as it is excited by an impact at the same location. The peak at 22Hz corresponding to the first mode of the double deck without absorber is significantly attenuated after introducing the absorber. This also results in the emergence of secondary resonances at 19 Hz and 24 Hz more than 5dB below the initial amplitude.

4 Conclusions and Contributions

This paper presented the practical implementation of passive vibration control in the context of railway tracks vibrations and focuses on the vibrations experienced by the track equipment in the vicinity of switchers. Such sensitive components are actuated by the switch motor, supported by a deck clamped to the sleepers. When a train goes by, this support deck experiences a transverse displacement of +/- 4 mm that damages the engine and results in the failure of the whole switcher.

After in-situ measurements and numerical simulations, kinematics likely to be responsible of damage were identified and two curative solutions were proposed and implemented. Further investigations suggested to focus on the first cantilever mode of the deck supporting the switch engine. The introduction of stiffeners located at the fold angle enabled to reduce by more than 50% the transverse displacement at the free extremity of the deck. In addition, a tuned mass damper was designed to provide an increased mitigation performance. This single mode tuned mass damper was designed so that it is less than 10% of the modal mass associated to the first cantilever mode, and its mechanical and geometrical properties were chosen to comply with tuned mass damper design rules available in the literature. This resulted in an additional reduction by -3dB of the vibration level at the frequency of the cantilever mode. However, this damper has an optimal operating frequency and should be tuned after a preliminary in-situ measurement of the cantilever frequency.

Several options could be implemented to widen the frequency range of absorption and improve the performance of the proposed absorber, such as, introduction of dissipation, either by viscoelastic components or composite materials, or changing the features of the support. Apart from the reduction of vibratory level, this study is also driven by a compromise between efficiency of the vibration control and lifetime of both the proposed absorber and the support deck.

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