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FEEDBACK-LMS CONTROL OF LATERAL VIBRATION IN A TRAIN CAR

P. Persson, S. Johansson, L. Håkansson, I. Claesson, T. Samuels

Department of Telecommunications and Signal Processing
Blekinge Institute of Technology
SE-372 25 Ronneby, Sweden
per.persson@bth.se

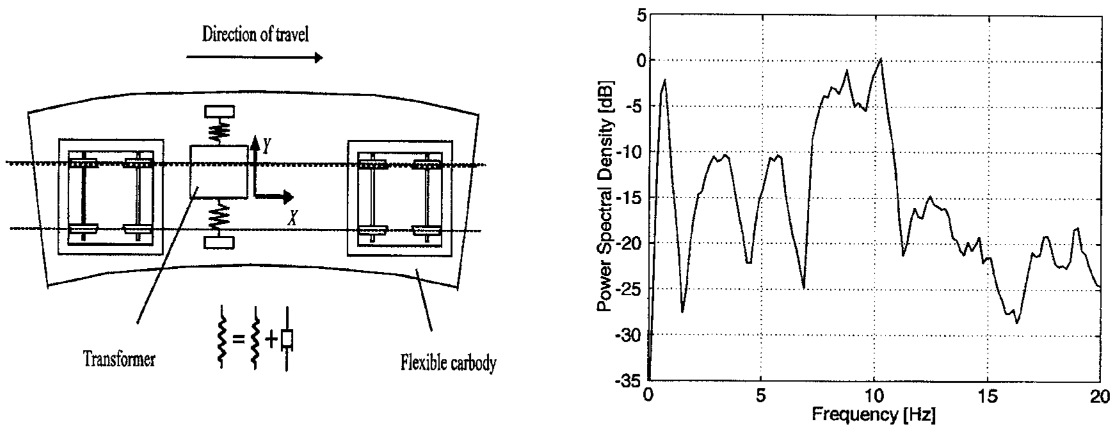
Abstract

The problem of low-frequency lateral train-car vibration is an important issue surrounding the design of high-speed trains. Passive solutions such as stiffening the car chassis are impractical because of the weight increase. Semi-passive solutions, such as modifying the structural dynamics of the carbody through non-rigid coupling of heavy underfloor equipment does not incur a weight penalty, but does not sufficiently reduce the vibrations. However, computer simulations based on signals derived from a dynamic computer model of a train car indicates that the incorporation of a multiple-reference feedforward active control system, in addition to the semi-passive approach, is likely to introduce a substantial reduction in the lateral vibration level. The complexity of the control system may be reduced by using a robust feedback-LMS controller. Results from computer simulations indicate that a feedback system would function as well as, and potentially better than, a feedforward system. The control results illustrate an attenuation of the lateral train-car vibration by up to 15 dB.

INTRODUCTION

Train designs are constantly being revised in order to improve transport economy and maximize safe travel speeds. The trend of continually designing trains for higher speeds has resulted in increasing noise and vibration problem inside railcars. In the past, however, proper attention has not been given to the increasing problem of railcar vibration. High-speed trains are significantly affected by both lateral and vertical vibrations which often have a negative impact on passenger comfort, a principal design criterion for new train architectures. In general, passenger comfort is much more affected by low-frequency lateral railcar vibrations than by similar vertical vibrations [1].

One of the primary mechanisms causing lateral vibrations is the excitation of the fundamental lateral bending mode of the carbody by yaw oscillations in the bogies. These bogie oscillations are induced by the irregularity of the tracks between railway sleepers. The oscillatory motion is enhanced due to the kinematic instability of the wheelset, which is built-in by the wheel conicity at the contact point between wheels and rail; the wheel conicity allows the lateral displacement necessary for the wheels to handle curved track. The frequency of the resulting bogie oscillations increases with the speed of the train and can reach as high as 8 Hz, which approaches the first resonant frequency of the carbody. The power-spectral density estimate for the uncontrolled lateral carbody vibration is shown in Figure 1(b). The peak just above 10 Hz corresponds to the first bending mode of the carbody.



(a) (b)

Figure 1: *(a) Top-view schematic of bending train car with non-rigidly coupled transformer. (b) Power-spectral density of lateral carbody vibration.*

Studies have been carried out in order to select the most suitable solution to alleviate the vibration problem. It has been concluded that purely passive means such as stiffening the chassis are not advantageous [2], partly because of the weight increase. Further, it has been shown that semi-passive solutions such as modifying the structural dynamics of the carbody by elastically suspending the heavy transformer underneath, while they do generally reduce vibration [1], do not do so to a sufficient degree. Therefore, an active control implementation is very likely a more appropriate solution to the vibration problem, with a minimal increase to the weight of the railcar. The active controller would regulate the vibrational movement of the train car by forcing it in opposition to the existent lateral motion. Figure 1(a) demonstrates the non-rigidly coupled transformer, which is used for semi-passive control and potentially for active control.

Active vibration control systems are based on either feedforward control or feedback control. Feedforward control relies on the existence of some prior knowledge of the vibration to be controlled, which is contained in a reference signal that drives the secondary (control) source through the controller. Generally speaking, the ideal active controller is of feedforward type and is based on the fact that the primary source or sources of the undesired vibration can be sensed in order to produce a reference signal, or combination of signals, that is to a high degree correlated with the undesired vibration [3, 4, 5]. Unfortunately, when the primary vibration is caused by a large number of uncorrelated sources, it often becomes impractical and costly to use feedforward control methods because of the large number of reference signals that must be used [6]. There do arise situations where the primary source of the vibration cannot be observed and used as a feedforward reference signal [7]. In these cases a feedback control system, which utilizes the residual error signal as a reference, must be considered.

There are some advantages of the feedback method over feedforward control. First of all, the often challenging task of locating suitable reference signals is eliminated. Also, the control algorithm is easier to code for a digital signal processor (DSP) used in an actual implementation. Moreover, the performance of the feedforward controller is highly dependent on the *quality* of the reference signals, so that in many cases a feedback system may perform equally well or better compared with feedforward control. Whereas feedforward control systems are theoretically more robust than feedback control systems, particularly when the feedforward system has a reference input isolated from the secondary source [4], the use of a leaky-LMS implementation of the feedback controller has been shown to increase the margin with respect to the Nyquist stability criterion [7].

In practice both the excitation signal and the system under control will change with time and an *adaptive* controller that is able to handle variation in the excitation signal and the plant is neccessery. In order to solve the lateral train-car vibration problem, two adaptive controller-algorithms have been investigated, namely the feedforward multiple-input/single-output (MISO) filtered-x Least-Mean-Square (LMS) algorithm and the feedback filtered-x LMS algorithm.

The investigation of the adaptive algorithms for the control of lateral train-car vibration is based on static computer simulations in the MATLAB environment, using vibration and reference signals calculated in advance. However, the objective is to implement the different vibration suppression algorithms directly within a dynamic-model environment, that is, allow the train car to react to the control forces during the simulation, and thereby obtain control results that would indicate a more accurate performance potential. The results presented in this paper and in [8] show real promise for an active control solution to the lateral vibration problem in high-speed trains.

ADAPTIVE CONTROL SYSTEMS

The adaptive control system used is based either on feedforward or feedback control strategy. A feedforward system is based on reference signal(s) derived from the vibration source(s). These signals contain *a priori* information about the vibrations and are used to generate driving signals for the active system. Two primary criteria must be fulfilled for a signal to be chosen as a candidate reference to a feedforward control system. First, the signal must be readily obtainable. To be readily obtainable implies that the signal should be easily measured by a sensor and fed to the controller. Second, the signal shall have high coherence with the signal to be controlled. In the case of multiple-reference feedforward control, any single reference signal may or may not have high coherence with the output, but the *combination* of reference signals must have it. The potential effect of a feedforward active control system can be estimated in advance of implementation from the coherence between the input, or combination of inputs, and the *disturbance* signal. An estimate of the maximum attenuation, $A(f)$, in decibel of the lateral vibration that can be expected is given by

$$A(f) = -10log_{10}\left(1 - \gamma^2_{y:x}(f)\right) \quad \text{dB} \tag{1}$$

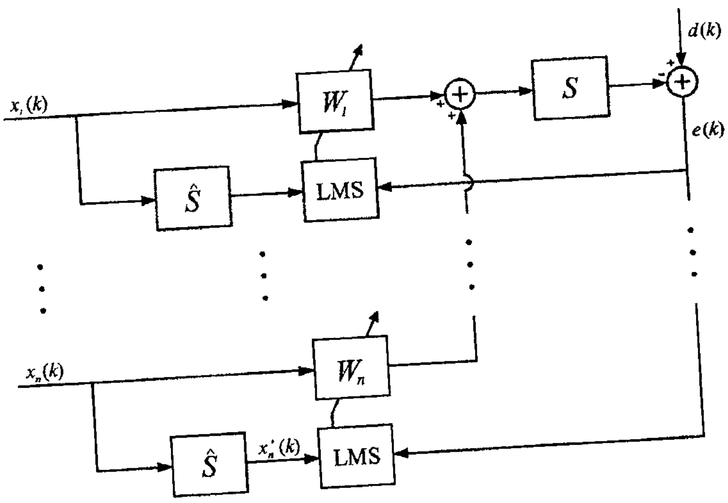


Figure 2: *Schematic diagram of multiple-input/single-output control system using the filtered-x LMS algorithm.*

where $\gamma^2_{y:x}(f)$ is the multiple coherence function between the selected reference combination and the response [4, 5]. The closer to unity the coherence is at any given frequency, the greater the attenuation that can be expected at that frequency. Various combinations of candidate reference signals were used in performance prediction calculations in order to choose the combinations most likely to yield good control results. A schematic diagram of the multiple-input/single-output filtered-x LMS controller is shown in Figure 2. In the figure, the signals $x_n(k)$; $n \in \{1, \cdots N\}$ are the feedforward reference signals, $x_{n,\hat{S}}(k)$ are the filtered reference signals, $d(k)$ is the disturbance signal, which physically represents the vibrating carbody before control, and $e(k)$ is the error signal, the residual vibration at the carbody center. S is the true secondary path, the transfer function between the controller output and the error sensor, and $\hat{S}$ is a secondary path estimate. $\mathbf{w}_n(k)$ are FIR filters of length L, whose coefficients are updated by the Least-Mean-Square (LMS) algorithm using the filtered reference (filtered-x) signals and the error signal. Regarding the schematic in Figure 2 the leaky filtered-x LMS update algorithm, with leakage factor $\gamma; 0 < \gamma \leq 1$, is as follows:

$$\mathbf{w}_n(k+1) = \gamma \mathbf{w}_n(k) + \mu_n \mathbf{x}_{n,\hat{S}}(k) e(k) \tag{2}$$

where μ_n is the adaptation step length for $\mathbf{w}_n(k)$,

$$\mathbf{w}_n(k) = \left[w_{n_0}(k)\; w_{n_1}(k)\; \ldots\; w_{n_{L-1}}(k)\right]^T, \tag{3}$$

the adaptive filters corresponding to the controller channels and

$$\mathbf{x}_{n,\hat{S}}(k) = \left[x_{n,\hat{S}}(k)\; x_{n,\hat{S}}(k-1)\; \ldots\; x_{n,\hat{S}}(k-L+1)\right]^T \tag{4}$$

the reference signals filtered with the secondary path estimate, $\hat{S}$.

Because there is no need to choose reference signals in a feedback system such a controller is relatively much simpler compared with the feedforward system described above. Using the error signal in Figure 2 as an input to the adaptive FIR filter, $\mathbf{w}(k)$, which controls the plant, causes the adaptive filter to act as a feedback controller. A schematic diagram of the adaptive feedback

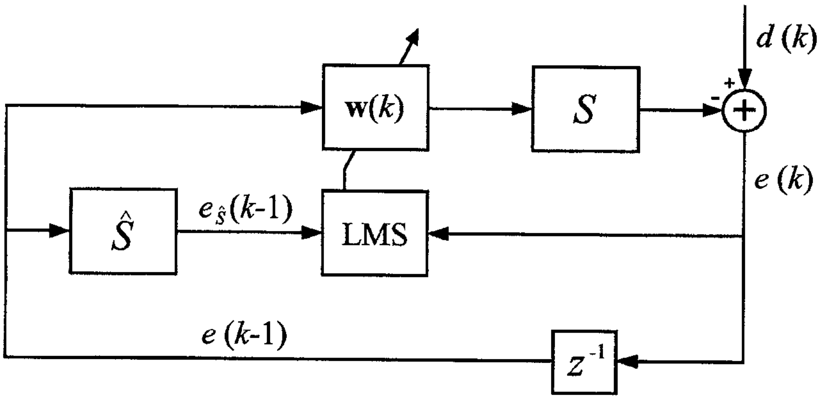


Figure 3: *Schematic diagram of adaptive feedback active control system.*

controller is shown in Figure 3. The box with the unit-delay operator, z^{-1}, describes the fact that we are dealing with an adaptive digital filter in a feedback control system. Notice that the reference signal from the discussion of feedforward control, that is $x(k)$, is here equal to $e(k-1)$, showing that the residual signal at the error sensor is the only information available to the controller. The control filter vector $\mathbf{w}(k)$ is updated by the LMS algorithm using the filtered signal $e_{\hat{S}}(k-1)$ and the error signal $e(k)$. The feedback controller update equation, based on the LMS scheme, is given by

$$\mathbf{w}(k+1) = \mathbf{w}(k) + \mu e(k)\mathbf{e}_{\hat{S}}(k-1) \tag{5}$$

where

$$\mathbf{e}_{\hat{S}}(k-1) = [e_{\hat{S}}(k-1) \;\; e_{\hat{S}}(k-2) \;\; \ldots \;\; e_{\hat{S}}(k-L)]^T . \tag{6}$$

The stability of the feedback control system depends on the ability of the filtered-x LMS algorithm to control the adaptive FIR filter – the time varying controller response – without violating the closed-loop stability requirements, namely the Nyquist stability criterion [9]. The criterion states that if the closed-loop system is to be stable, then the polar plot of the open-loop frequency response for the feedback control system, $S(f)w(f)$, must not enclose the point $(-1, 0)$. The larger the distance between the polar plot and the $(-1, 0)$ point, the more robust the feedback control system becomes, with respect to variation in the controller response and secondary path response. The feedback filtered-x LMS algorithm has been reported to be robust in control applications involving narrowband vibration control [7].

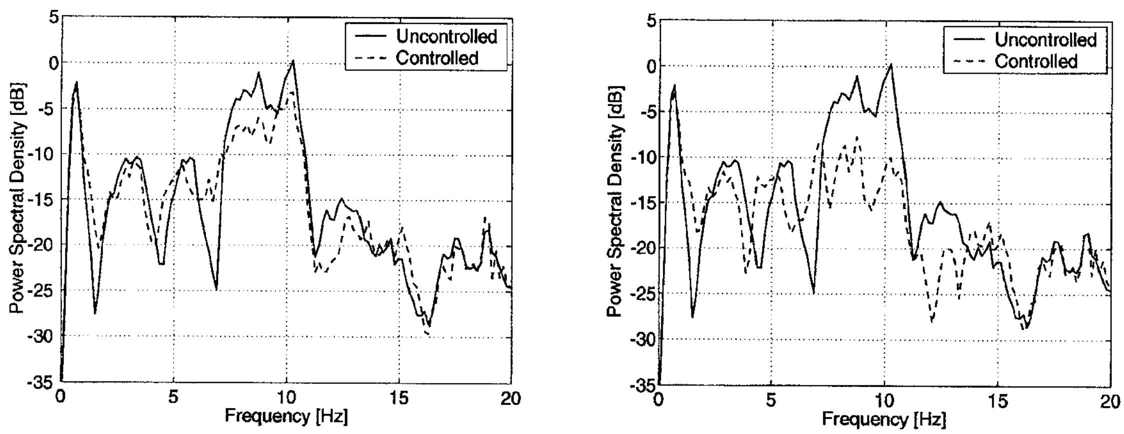
SIMULATIONS AND RESULTS

All the signals used in the MATLAB simulations were obtained from a dynamic computer-model simulation of a vibrating railway car with a non-rigidly coupled transformer. Actual track-irregularity data served as input to the dynamic model and simulations provided the vibrational movement of the carbody.

In the simulations the secondary path, S, is simply an FIR filter obtained from the dynamic computer-model of a carbody, and $\hat{S}$ is identical to S. The undesired vibration signal, $d(k)$, is the lateral movement of the center of the carbody (the undesired vibration) from a running train. The length of the control-filter vector chosen was L=40. In the case of multiple references feedforward control the control filters, $\mathrm{w}_n(k)$, were always of common length. For the feedforward control situation the adaptation step length, μ_n, for each reference was manually adjusted to optimize the attenuation achieved at the target frequency of 10 Hz. The values of μ_n chosen for best performance during these simulations were $\mu_n = 0.04 \; n \in \{1, 2\}$. The leakage factor, γ, used was always $\gamma = 0.99999$.

For the feedback controller the value of μ chosen for best performance was $\mu = 0.03$. The performance of the two controller structures was evaluated by analyzing the steady-state error signal.

In previous studies [3, 10] a multiple-reference controller performed dramatically better than a single-reference. Moreover, a combination of four reference-signals did not yield significantly improved results when compared with two references. Figure 4 shows the power spectral densities of the original vibration signal and the steady-state error signal corresponding to single-reference 4(a) and twin-reference 4(b) feedforward control.



(a) (b)

Figure 4: *Simulation results of feedforward control. Solid line: lateral vibration before control. Dashed line: lateral vibration during control. Plot (a) is the single-reference case and plot (b) is the twin-reference case.*

As can be seen in Figure 4, the multiple-reference controller had greatest effect in the region of frequencies between 8 and 11 Hz, while the single-reference controller most prominently affected the vicinity of 8 Hz. The improvement of using a multiple-input system over a single-input system is obvious from these plots.

Because there is no need to choose reference signals in a feedback system, obtaining simulation results of feedback control is relatively much simpler compared with the feedforward system described above. Plotted in Figure 5 are two power spectral densities, namely the original disturbance signal and the steady-state error signal resulting from adaptive feedback control. At specifically the 10 Hz target frequency there is excellent control, approximately 15 dB, which is better than that achieved by any of the multiple-input feedforward systems. The attenuation achieved at other frequencies within the entire applicable frequency range of course depends upon the frequencies of interest, and therefore at some frequencies control is better, and at some it is worse when compared with feedforward control.

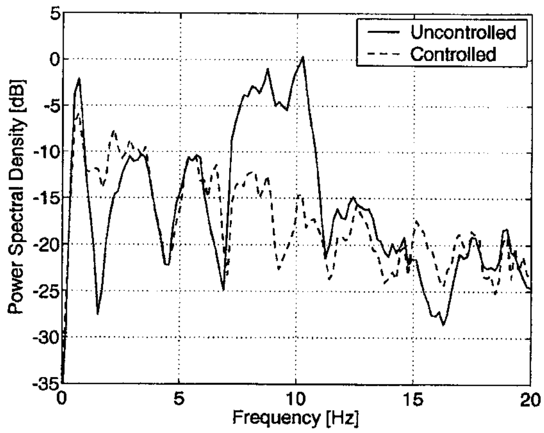


Figure 5: *Simulation result for adaptive feedback active control.*

Examining the overall shape of these power spectral densities at all frequencies, for both the uncontrolled and controlled signals, one can see that the highest peaks in the original vibration have been lowered, at the expense of some frequencies where the vibration has increased. This effect can be explained by the fact that the basic motive of the adaptive feedback controller is to decorrelate the reference signal with the estimation error.

To describe this result simply, the feedback controller has basically annihilated the main peaks between 7 Hz and 11 Hz, bringing them down to the broadband vibration level. Based on this result, it is fair to say that for this application, the feedback controller should be considered as seriously as the feedforward controller.

CONCLUSIONS

The results shown here suggest real promise for an active control solution to the lateral vibration problem in high-speed trains. Both feedforward and feedback solutions are conceivably applicable in this situation. A functional feedforward system would require multiple inputs in order to achieve satisfactory vibration control. The computer simulations based on signals derived from a dynamic computer model of a train car indicates that the incorporation of a twin-reference feedforward active control system, in addition to the semi-passive approach, is likely to effect a substantial reduction in the lateral vibration level. The complexity of the control system may be reduced by using a robust feedback-LMS controller. Results from computer simulations indicate that a feedback system would function as well as, and potentially better than, a feedforward system. The control results illustrate attenuation of the lateral train-car vibration of up to 15 dB.

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