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ACTIVE VIBRATION REDUCTION IN A LIGHT WEIGHT HIGH SPEED TRAIN BOGIE

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Abstract: Active control of vibration-induced noise in light weight train cars is of great interest since low weight trains are more economic but also have higher internal noise levels. This paper describes an approach to isolate bogie vibrations in such a way that the noise inside the car is reduced. An active control system with four inertial mass actuators fitted to the bogie were used to control bogie vibrations and noise inside the car. The tests were performed on an experimental light weight bogie fitted to a test car, in both lab environment and in full scale tests on track at full speed. *Copyright ©1999 IFAC*

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1. INTRODUCTION

In the design of future trains the manufacturer has two important goals to achieve, low cost, both in terms of the actual material costs and the total life cycle cost, and passenger comfort in terms of noise and vibrations. In order to minimize manufacturing and total life cycle costs low weight is essential but since a light structure will be more resonant, a mass reduction will have a negative impact on the noise inside the car and thus on passenger comfort.

Within train manufacturer Adtranz' advanced technology program, Adtranz Sweden has developed an experimental bogie, the X15-5, which is 30% lighter than the ordinary bogie fitted on the current line of trains, to be able to evaluate the negative side effects due to weight reduction and the possible countermeasures that can be applied.

Due to the decrease in weight the X15-5 bogie is more resonant than its predecessors and this results in increased interior noise compared to an ordinary train. Measurements performed by Adtranz show that structureborne noise dominates in the region below 200 Hz which makes it hard to compensate for the increased noise by classical methods. Active control of noise and vibrations works best at low frequencies and as a part of the advanced technology program the question if active control could be used was raised.

One distinct source of vibrations and noise is the vertical oscillating movement in the bogie due to the passing of sleepers and at a speed of 200 km/h the sleeper passage frequency (SPF) is approximately 85 Hz. From an active control point of view this is a well suited problem, since the number of acoustic modes at these frequencies are fairly low and the excitation source is well defined. At the sleeper passage frequency the main issue with respect to comfort is audible noise and not vibrations and thus the aim of the study was to investigate whether it was possible to reduce the noise level inside a train car, fitted with the experimental light weight bogie, using an active control approach.

The material presented in this paper emanates from two projects; a joint project between FFA (the Swedish Aeronautical Institute), Adtranz Sweden, ABB Corporate Research, The Royal Institute of Technology and the University of Karlskrona/Ronneby involving modelling and simulations as well as experimental work on different aspects of applying active control techniques to reduce noise in trains, see (Einarsson-Papadio, et al., 1997), followed by a joint project between Adtranz Sweden and the University of Karlskrona/Ronneby involving experimental studies of the bogie/car system as well as full scale tests with a train comprising of five cars running at full speed on ordinary track.

2. THE ACTIVE CONTROL SYSTEM

All experiments were performed on a prototype X2000 coach car fitted with the experimental light weight bogie. The bogie connects to the car at four connection points (AP1-AP4) at the end of the coupling beams which in turn are floating on air cushions mounted on the lower part of the bogie. Any vibrations that enters the car through the bogie will have to pass through these connection points in order to excite the car body structure.

Measurements of the coherence between the excitation and the noise inside the car indicated that it should be possible to decrease the periodic noise caused by the sleeper passage frequency significantly by preventing the vibrations in the bogie to excite the carbody structure. In figure 1 an example of the coherence measurements is shown.

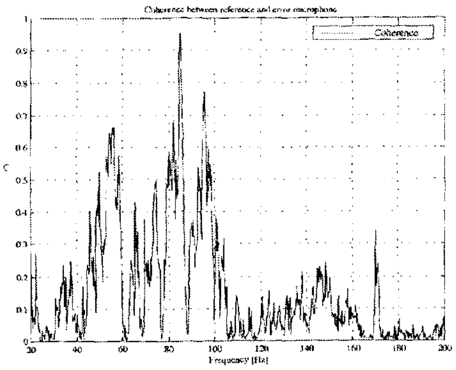

Fig. 1. Measured coherence between external excitation and interior noise in the car/bogie setup. The coherence at the sleeper passage frequency (85 Hz) is 0.96.

Since the four connection points were the only possible paths for vibrations entering the carbody structure it should be possible to achieve global reduction of sleeper induced noise by placing actuators close to the connection points. Due to safety restrictions no alteration to the bogie itself could be made and thus any actuators had to be mounted externally. In order to be able to insert force into the connection points a set of four inertial mass actuators, each capable of delivering a continuous force of 226 N and a peak force of 453 N, were rigidly mounted externally at the coupling beam, close to the connection points.

The heart of the system was a multiple input/multiple output (MIMO) filtered-X LMS-algorithm, schematically drawn in figure 2, with K outputs and M inputs and a single reference.

The vector $\mathbf{w}(n)$ represents the adaptive filter weights for all K adaptive filters of order L stacked into one long vector, i.e.

$$\mathbf{w}(n) \equiv [\mathbf{w}_1^T(n)\ \mathbf{w}_2^T(n)\ \ldots\ \mathbf{w}_K^T(n)]^T \quad (1)$$

where

$$\mathbf{w}_k(n) \equiv [w_{k,0}(n)\ w_{k,1}(n)\ \ldots\ w_{k,L-1}(n)]^T. \quad (2)$$

The output signal vector $\mathbf{y}(n)$ is used to drive the K secondary sources, and each element $y_k(n)$ is obtained by filtering the reference $x(n)$ by the corresponding filter, $\mathbf{w}_k(n)$:

$$y_k(n) = \mathbf{w}_k^T(n)\mathbf{x}(n), \qquad k = 1, 2, \ldots, K \quad (3)$$

where

$$\mathbf{x}(n) \equiv [x(n)\ x(n-1)\ \ldots\ x(n-L+1)]^T. \quad (4)$$

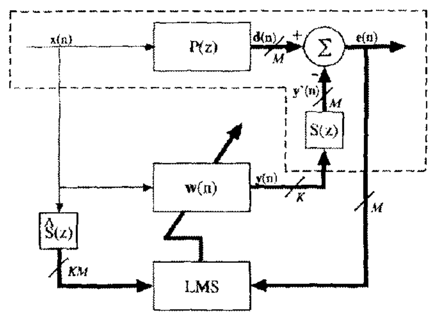

Fig. 2. A schematic view of the car/bogie system (within the dashed area) and the controller. The reference signal, $x(n)$, was taken from a bearing box on the front wheelshaft and the output signal vector, $\mathbf{y}(n)$, was driving the actuators. The error signal vector, $\mathbf{e}(n)$, was taken either from accelerometers in the connection points between the car and the bogie or from microphones inside the car.

The reference signal, $x(n)$, was taken from an accelerometer on one of the bearing boxes of the front wheelshaft of the bogie and the output signals, $y_k(n)$, were fed to the inertial mass actuators. The error signals were provided by either a set of accelerometers in the connection points (AP1-AP4) between the car and the bogie, see section 3.1, or a set of microphones inside the car, see section 3.2.

2.1 *Stability and output limiting*

When implementing an active control application two aspect are of great importance, besides the actual performance of the system, the long term stability of the adaptive filter weights and the limiting of the output signals. For simplicity, the following calculations will be done for the case of $K = 1$ and $M = 1$ but it is straightforward to extend the calculations to any choice of K and M.

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The basic weight updating scheme of the LMS-algorithm:

$$\mathbf{w}(n+1) = \mathbf{w}(n) - \frac{\mu}{2}\nabla\xi(n) \qquad (5)$$

where $\xi(n) = E[e^2(n)]$ is the expectation value of the error and μ is the step-length of the gradient decent, is simplified by using the instantaneous squared error, $\hat{\xi}(n) = e^2(n)$, instead of $\xi(n)$, where

$$e(n) = d(n) - \mathbf{w}^T(n)\mathbf{x}'(n) \qquad (6)$$

and

$$\mathbf{x}'(n) = \hat{s}(n) * \mathbf{x}(n) \qquad (7)$$

is the linear convolution of the reference signal, $\mathbf{x}(n)$, and the impulse response of the estimated secondary path, $\hat{\mathbf{S}}(z)$.

This gives

$$\nabla\hat{\xi}(n) = -2\mathbf{x}(n)e(n) \qquad (8)$$

which together with equation 5 gives

$$\mathbf{w}(n+1) = \mathbf{w}(n) + \mu\mathbf{x}'(n)e(n). \qquad (9)$$

Divergence of the adaptive weights, due to insufficient spectral excitation of the LMS-algorithm, can be avoided by adding a leaking mechanism to the weight updating scheme given in equation 9.

The leaky-LMS algorithm is formulated as

$$\mathbf{w}(n+1) = \nu\mathbf{w}(n) + \mu\mathbf{x}'(n)e(n) \qquad (10)$$

where $0 < \nu \leq 1$ is the leakage factor, see (Gitlin, et al. 1982).

The problem of limiting the output signal to the inertial mass actuators in order to avoid non-linear distortion can be solved by adding a second term containing the squared output term with a limiting factor, γ, to the cost function, see (Elliot, et al.,1987).

$$\hat{\xi}(n) = e^2(n) + \gamma y^2(n) \qquad (11)$$

or by constraining the adaptive filter weights

$$\hat{\xi}(n) = e^2(n) + \gamma\mathbf{w}^T(n)\mathbf{w}(n). \qquad (12)$$

The error gradient can then be expressed as

$$\nabla\hat{\xi}(n) = -2\mathbf{x}'(n)e(n) + 2\gamma\mathbf{w}(n) \qquad (13)$$

and the weight updating scheme becomes

$$\mathbf{w}(n+1) = (1-\gamma\mu)\mathbf{w}(n) + \mu\mathbf{x}'(n)e(n) \qquad (14)$$

where, by comparing equations 10 and 14

$$\nu \equiv (1-\gamma\mu). \qquad (15)$$

This degrades the performance somewhat but since the excess error power is proportional to $[(1-\nu)/\mu]^2$, see (Bellanger, 1987). This degradation can be kept at a minimum by making sure that $1-\nu$ is smaller than μ.

3. LAB TESTS OF THE ACTIVE CONTROL SYSTEM

The first series of tests were performed in lab settings where a controlled environment with respect to primary excitation an external noise could be obtained. The primary excitation of the bogie was provided by four shakers connected to the wheel shafts, driven by a sinusoidal input signal at a frequency corresponding to the sleeper passage frequency. A time delay, corresponding to the distance between the front and rear weel shaft of the bogie, was added to the signal controlling to the rear pair of wheels. The estimate, $\hat{S}$, of the secondary path was obtained off-line by a linear chirp excitation.

3.1 *Active Vibration Control*

The first tests used the acceleration in the connection points as error signals and the aim was to simply minimize the vibrations due to the sleeper passage frequency. The active vibration control (AVC) system performed very well indeed, giving an attenuation in vibration levels in the order of 40 dB, see Table 1.

Table 1 The acceleration levels in the control sensors in the AVC test

Pos	Acceleration AVC Off	Acceleration AVC On	Attenuation
1	$16mm/s^2$	$0.14mm/s^2$	$41dB$
2	$19mm/s^2$	$0.16mm/s^2$	$41dB$
3	$9.5mm/s^2$	$0.10mm/s^2$	$40dB$
4	$9.9mm/s^2$	$0.14mm/s^2$	$37dB$

The results, however, were ambigous with respect to the noise levels inside the car and a set of four evaluation microphones placed inside the car showed increasing as well as decreasing levels, see Table 2. This is due to the fact that different vibration modes in the bogie couples differently to the acoustic modes in the car and even though the inertial mass actuators minimizes the vibration levels at the connection point there is no way of making sure that it also minimizes the noise levels inside the car.

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Table 2 The noise levels in the evaluation microphones inside the car in the AVC test

Pos	SPL AVC Off	SPL AVC On	Attenuation
1	$42dB$	$54dB$	$-12dB$
2	$44dB$	$36dB$	$8dB$
3	$48dB$	$52dB$	$-4dB$
4	$53dB$	$42dB$	$11dB$

3.2 *Active Structure Acoustic Control*

In the light of the tests with active control of vibrations, clearly showing that the impact of the inertial mass actuators on the structure was enough to minimize the vibration levels in the connection points and alter the vibrational modes of the bogie a new approach was tested. Instead of trying to minimize the vibration levels the target for the active control algorithm was the narrow band noise, centered on the excitation frquency, inside the car. This technique is often referred to as Active Structure Acoustic Control (ASAC), see (Fuller, et al., 1992).

The setup was the same as in section 3.1 except that the error sensors were microphones inside the car instead of accelerometers at the connection points. The results are shown in Table 3.

Table 3 The noise levels in the control microphones inside the car in the ASAC test

Pos	SPL AVC Off	SPL AVC On	Attenuation
1	$54dB$	$49dB$	$5dB$
2	$50dB$	$46dB$	$4dB$
3	$55dB$	$53dB$	$2dB$
4	$54dB$	$50dB$	$4dB$

4. FULL SCALE TESTS WITH ASAC

Given the results from the lab testing of the system a full scale test was decided upon. The setup was similar to the ASAC lab tests with inertial mass actuators in the connection points between the car and the bogie (AP1-AP4) and a total of 24 microphones placed in a grid with a spacing of 0.5m at 1.2m above the floor inside the car, and out of these 24 microphones eight where used as error sensors. The microphone positions and the numbering scheme is depicted in figure 3. As before, the secondary path estimate, $\hat{S}$, was calculated off-line between testruns.

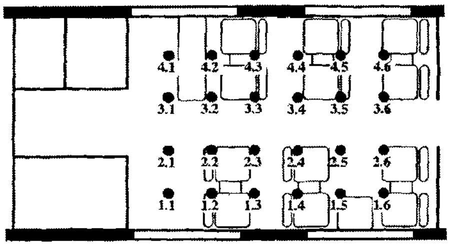

Fig. 3. The microphone positions used in the full scale test of the ASAC system. The centre of the bogie is approximately 0.5m to the left of the first line of microphones (positions 1.1-4.1).

An example of the behavior of the ASAC system can be found in figure 4 where a reference run, with the ASAC system switched off, is compared to a testrun over the same stretch of track. The ASAC system was switched on after approximately 90 seconds of travel. The system was stable over the full stretch of track and handled transients well, as can be seen in figure 4 where the peak after approximately 125 seconds stems from a railway crossing.

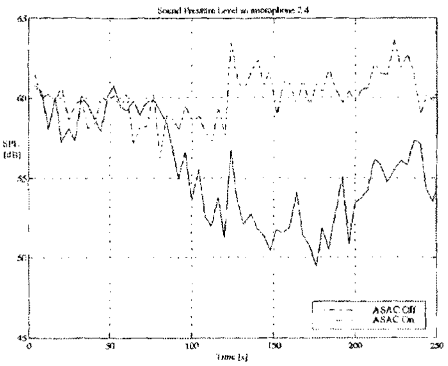

Fig. 4. The noise level inside the car at one of the control microphones for a reference run (dashed) and a test run (solid). The ASAC system was activated after approximately 90 seconds of the test run.

In all tests eight microphones were used as error sensors while the remaing microphones were used as evaluation microphones and the results from one of the testruns can be found in table 4. In this particular case microphones 2.2, 3.2, 2.3, 3.3, 2.4, 3.4, 2.5 and 3.5 (underlined in table 4) were used as error sensors.

Table 4 The narrow band noise levels and attenuation inside the car in the full scale test [1]

Pos	SPL AVC Off	SPL AVC On	Attenuation
1.1	$54.6dB$	$58.0dB$	$-3.4dB$
2.1	-	-	-
3.1	$55.0dB$	$54.8dB$	$0.2dB$
4.1	$61.2dB$	$58.3dB$	$2.9dB$
1.2	$59.7dB$	$54.8dB$	$4.9dB$
2.2	$61.0dB$	$53.1dB$	$7.9dB$
3.2	$57.7dB$	$51.3dB$	$6.4dB$
4.2	$59.6dB$	$53.6dB$	$6.0dB$
1.3	$61.4dB$	$57.1dB$	$4.3dB$
2.3	$62.6dB$	$55.7dB$	$6.9dB$
3.3	$61.8dB$	$54.5dB$	$7.3dB$
4.3	$60.8dB$	$55.9dB$	$4.9dB$
1.4	$60.4dB$	$54.3dB$	$6.1dB$
2.4	$61.7dB$	$54.8dB$	$6.9dB$
3.4	$60.1dB$	$55.0dB$	$5.1dB$
4.4	$58.1dB$	$55.1dB$	$3.0dB$
1.5	$58.0dB$	$56.0dB$	$2.0dB$
2.5	$59.2dB$	$54.2dB$	$5.0dB$
3.5	$56.7dB$	$52.8dB$	$3.9dB$
4.5	$51.0dB$	$51.1dB$	$-0.1dB$
1.6	$58.1dB$	$59.1dB$	$-1.0dB$
2.6	$56.0dB$	$55.6dB$	$0.4dB$
3.6	$53.4dB$	$53.9dB$	$-0.5dB$
4.6	$53.4dB$	$55.3dB$	$-1.9dB$

5. CONCLUSIONS

The results from the various tests verify that a light weight bogie is more susceptible to excitation from the sleeper passage frequency than the original, heavier, structure. It has also been shown that an active control approach using inertial mass actuators is capable of suppressing these vibrations substantially. By applying an ASAC- instead of ANC-approach it is possible to achieve a global attenuation of the noise, induced by the sleeper passage frequency, inside the car. For the active control algorithm it can be concluded that

- The AVC algorithm converged and reduced the vibration levels in the connection points between the bogie and the car when excited with a single frequency in lab environment.
- The ASAC algorithm converged and reduced the sound pressure levels in the car when excited with a single frequency in lab environment.
- The ASAC algorithm converged and globally reduced the narrowband sound pressure levels, centered on the sleeper passage frequency, in the car when run at full speed on track.
- The inertial mass actuators were capable of handling the forces needed to suppress vibrations in the connection points and change the vibration modes of the bogie in order to effect the sound inside the car.
- The algorithm was stable with respect to transients due to rail crossings etc.
- The output limitation inherent in the leaky implementation of the filtered-X LMS algorithm prevented the actuators from distorting due to high driving signals.

[1] The values for microphone 2.1 are omitted because of data corruption due to a faulty connector.

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