Active Attenuation of Squeal Noise at Train Wheels

H. Siebald[1], H. Fehren[1], M. Schiedewitz[1], A. Cigada[2], S. Manzoni[2], M. Redaelli[2]

[1] *ERAS GmbH, 37085 Göttingen, Germany, Email: siebald@eras.de*

[2] *Politecnico di Milano, 20156 Milan, Italy, Email: matteo.redaelli@polimi.it*

Introduction

Squeal noise at train wheel emission is a widely discussed topic for many years. Squeal noise is an acoustic emission characterized by one or few harmonics; which makes it very annoying for the human hearing. The mechanical phenomenon responsible for such a radiation usually is a stick/slip one. The slip behaviour is responsible for the instability of some wheel (flexural) modes and causes a exponential increase, causing noise emission. The stick behaviour is responsible for the limitation of this growth and imposes a limit cycle[1][2].

City-trams, which usually travel along sharp bends on city tracks, are often affected by this phenomenon. Thus people living in those cities are pretty much exposed to such an acoustic pollution, which is the main reason pulling towards an effective solution for squeal attenuation. Very different strategies have been adopted in the past years. In [3] piezoelectric-patches are adopted to actively control the wheel squeal noise on a test rig, while [4] presents a solution based on the modification of the friction coefficient between the rail and the wheel by applying a friction modifying liquid. These are only two examples showing that very different approaches can be chosen to face the problem under analysis.

This paper presents an investigation about the possibility to effectively control squeal noise under operating conditions by means of piezoelectric-actuators [5]. One main feature of the present work is that the system has been tested successfully on a full scale test rig for complete wheel sets.

Tested Resilient Wheel

Nowadays, two main kinds of wheels are mostly adopted: solid wheels and resilient wheels [6]. The attention will focus on the latter, as it is mounted to most modern city-trams. Resilient wheels are composed by two iron parts with rubber blocks between them. This particular structure and the damping added by the rubber blocks allow to effectively reduce rolling noise ([7]), if compared to the traditional solid wheels. On the other hand improvements in terms of squeal noise are not so clear[6].

The resilient wheel which has been considered for the control tests exhibits a diameter of about 600 mm, a thickness of about 115mm and it is provided with 22 rubber blocks.

This wheel is affected by squeal noise mainly at two eigenmodes: mode (2,0) at 730 Hz (Figure 1), which is a flexural one with two nodal diameters and mode(3,0) at 1940 Hz, which is a flexural one with three nodal diameters. In the present paper only the first mode will be considered. More details on the tested resilient wheel can be found in[6].

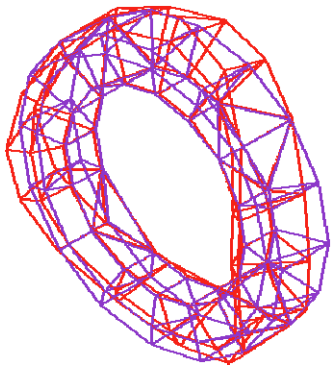

Figure 1: FEM estimated mode shape of wheel first flexural eigenmode.

The acceleration amplitude in the mode(2,0) at the wheel during squealing has been estimated with results of sound pressure measurements at a rolling train and laboratory tests with a non rotating wheel excited by a shaker. The tests show an amplitude of 140 m/s^2, measured in the four anti-nodal positions, where the vibration level is at its maximum.

Vibration Control Concept

Two different vibration control strategies have been discussed: semi passive damping and adaptive vibration control.

Piezoelectric Actuators

One demand of the project was to investigate piezoelectric actuators for the vibration attenuation. The best solution aims at introducing or damping energy in the flexural eigenmodes, and thus to control it, is the use of piezoelectric transducers able to provide a torque to the structure. Two kinds of actuator concepts have been investigated (s. Figure 2):

1. bending actuator fixed directly on the structure;
2. stack actuator places between lever arms.

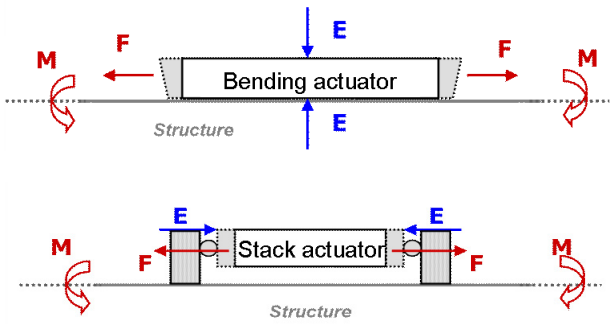

Figure 2: Functional scheme of piezoelectric bending actuator (above) and stack actuator (below).

Semi passive system

The conversion of mechanical energy into electrical energy and the dissipation afterwards is the basic idea of the semi-passive damping strategy. It does not require any electrical input: this is a great advantage in comparison to any active system. Moreover, the mechanical system stability is not affected by the semi-passive damper. On the other hand, each resonance frequency whose amplitude should be reduced requires a hardware implementation, and any possible modification of the wheel (e.g. stiffness changes due to operating conditions) reduces the solution efficiency. The effectiveness of this transformation is described by the electromechanical coupling factor, which is typically high for piezoelectric materials, and indicates the fraction of the energy that can be converted [8]. The coupling coefficient for the given configuration has been achieved by laboratory tests with four bending actuators mounted on the wheel: it is lower than 0.10%. Therefore the semi-passive damping has not been investigated for the wheel application beyond this point because it is not sufficient for this application.

Active Vibration Control

It is well known in literature [1][9][10][11] that one of the squeal noise peculiarities is that it is due to a self-excited phenomenon and that it can be reduced, or even suppressed, controlling the unstable wheel oscillation during the phenomenon birth, preventing its growth to the limit cycle[3][9]. Considering that previous research about disk brake squeal [12] shows the need to control vibrations ten times lower than the limit cycle amplitude, a vibration amplitude of about 14 m/s^2 has been taken into account for the actuator choice.

In the present work piezoelectric bending actuators Ceram-Tec Sonox P53 and piezoelectric stack actuators Tokin NEPEC N-10 have been used to investigate the different actuator concepts, s. Figure 3.

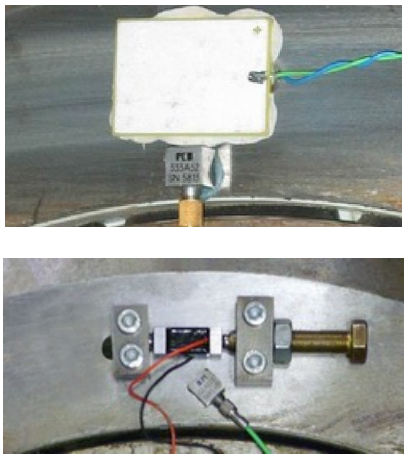

Figure 3: Piezoelectric bending actuator (above) and piezoelectric stack actuator (below) placed on the wheel.

The two different piezoelectric actuators are comparable concerning main parameters like power consumption and ceramic volume. Hence the mechanical action provided by the different actuators was compared in order to verify the compliance with the power requirements to prevent the growth of squeal noise-related vibration, and to identify the more efficient. Therefore the wheel has been excited on a laboratory test stand with the different actuator types. In each case two actuators had been mounted across from each other and the achieved acceleration amplitudes in the anti node lines in the resonance frequency have been measured.

The results show that both kinds of actuators would fulfill the power requirements (> 14m/s²) but the stack actuators are twice effective. Furthermore the available stacks are low voltage and the patches are high voltage actuators. Because the signal transfer into the rotating system is less difficult for low voltage signals due to a slip ring, the stack actuators have been chosen for the further investigations.

Actuator and Sensor Positions

The actuator and sensor positions have been defined with regard to a good effectiveness of the actuators in the 2nd flexural mode. From the theory the mode is fixed in the non rotating system whereas one of the two node lines is orthogonal on the wheel/track contact point. With this the actuator and sensor positions at the mode shape are varying during rotation. A high effectiveness of the actuators will be given if as many actuators as possible are out of a node at every point of time.

Therefore four actuators and sensors have been arranged in pairs at the wheel front surface. The angle between the position of the sensor/actuator pair 1 and 2 or 3 and 4 is 90 degree respectively. The angle between the position of sensor/actuator pair 1 and 4 is 45 degree (see Figure 4). If two actuators in this arrangement are in a node line, the other two will be in an anti-node line for the first flexural mode.

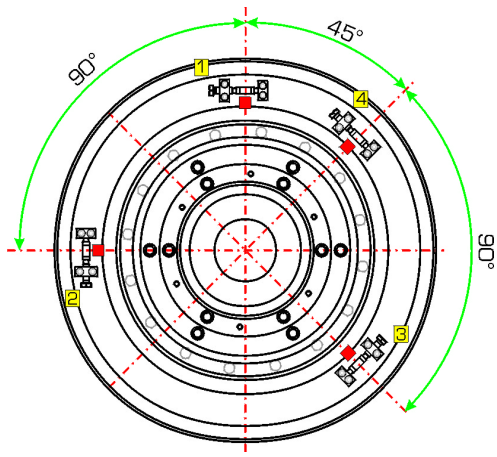

Figure 4: Position of actuator and sensor pairs (1 – 4) at the wheel.

Signal and Power transfer

A slip ring has been prepared for the transmission of the actuator driving signals into the rotating system and to transmit the sensor signals from the rotating system outward.

The used slip ring (G200, Schleifring, Germany), s. Figure 5, has got a very small size (outer diameter: 90mm, length: 116 mm). The maximum number of tracks is 60 and the operating voltage is 250 VAC in maximum.

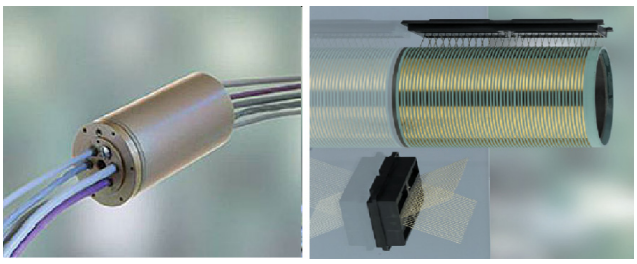

Figure 5: Slip ring for transmission of driving and sensor signals.

Controller

The active system works with an adaptive feedback controller. Figure 6 shows the block diagram of the controller. As known from the state of the art, the adaptive feedback control strategies are based on FIR filters for system model implementation and for control. The great advantage of the adaptive control is the self-tuning of its parameters to the optimal values. This allows to get a broadband control action, and a maximal effectiveness in the eigenfrequencies which are really excited. The feedback controller investigated uses a Least Mean Square (LMS) algorithm.

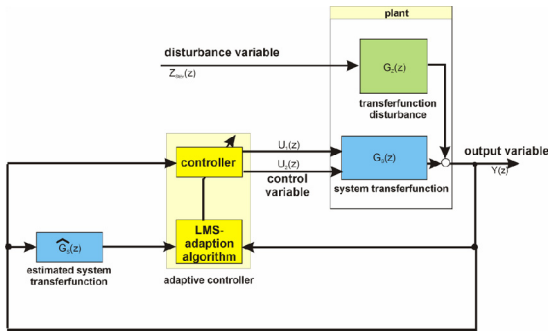

Figure 6: Block diagram of the adaptive feedback control with LMS algorithm.

Tests

Implementation

The prepared wheel with actuators, sensors and slip ring has been implemented on a wheel set and mounted in a full scale test stand at Lucchini, Lovere, Italy, Figure 7 and Figure 8.

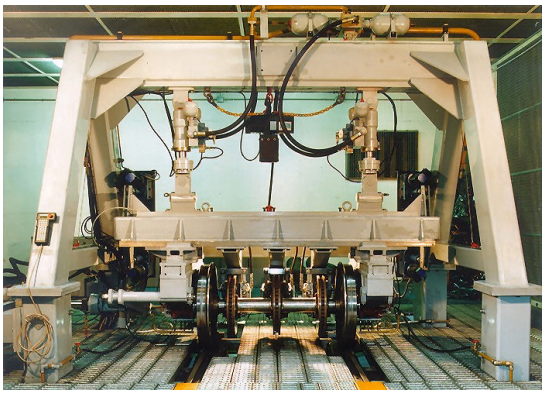

Figure 7: Full scale wheel test stand.

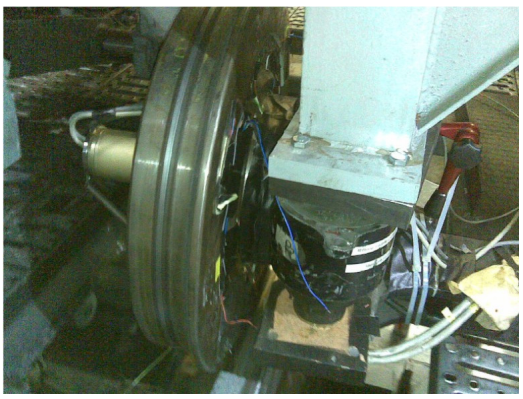

Figure 8: Resilient wheel implemented with actuators, sensors and slip ring at the wheel test stand.

Results

The airborne sound has been measured with two microphones in the near field of the wheel, one coplanar to the contact surface of the wheel and one in orthogonal direction to the wheel front surface.

Figure 9 shows the Campbell diagram of the sound pressure measured orthogonal to the wheel front surface. High squeal noise levels especially in the 251th order of the rotational frequency (between 740 and 750 Hz) can be seen. The level depends on the wheel rotational frequency which was between 2.95 and 3.0 Hz during this measurement period.

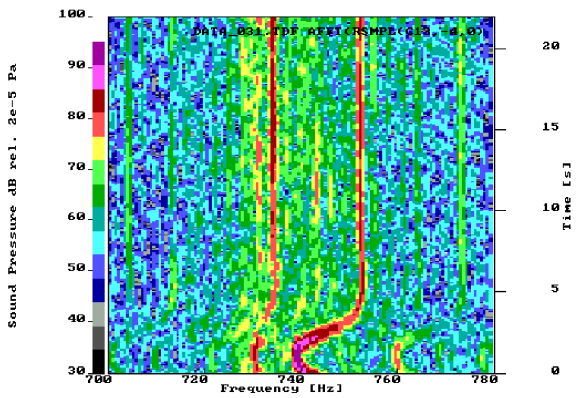

Figure 9: Campbell diagram of the sound pressure orthogonal to the wheel front surface, wheel rotational frequency between 2.95 and 3.0 Hz.

Due to the rotating acceleration sensors and the two node lines of the flexural mode the acceleration sensors measure two modulated squealing frequencies. This frequencies are the squealing frequency plus/minus of the twice rotating frequency (249th and 253rd revolution order). Figure 10 shows the spectrum of axial acceleration at the wheel at sensor 2 and 3 for active system switched on and off. Due to the modulation the squealing frequencies in the acceleration signals are at 740,97 Hz and 753,17 Hz. These tests have been conducted by driving only one of the four actuators. Tests with all actuators couldn't be performed because the wheel set derailed during the test period and sensors and actuators have been destroyed.

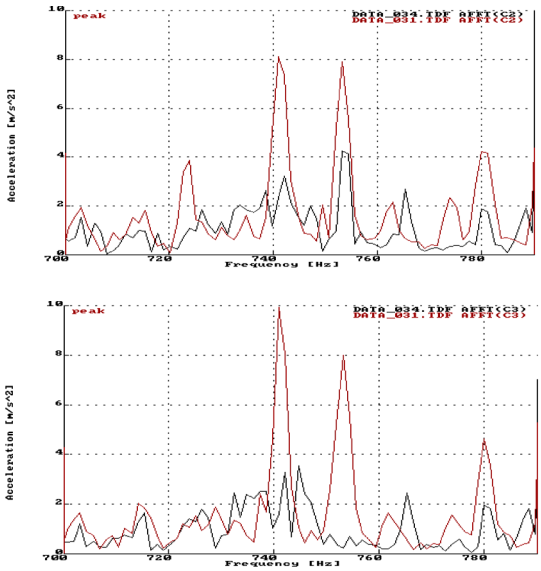

Figure 10: Acceleration spectrum (sensor 2 – above, sensor 3 - below): active system off – red curve, active system on – black curve

In Figure 11 the spectrum of the corresponding microphone signal in orthogonal direction to the wheel front surface can be seen. The considered squealing frequency (measured with the microphone) is at 747,07 Hz. In the active controlled mode the amplitudes of the acceleration are smaller by a factor of 2 - 8 than in the non controlled case. The sound pressure level in the squealing frequency of 747.07 Hz is reduced for 11 dB.

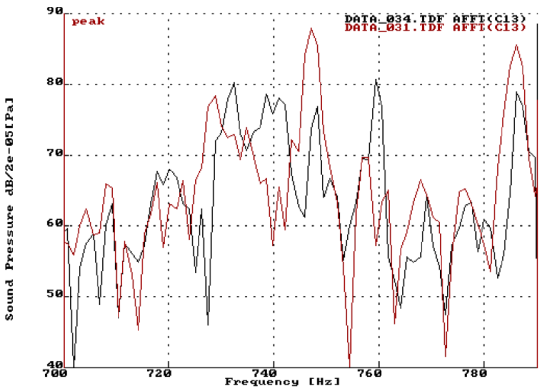

Figure 11: Spectrum of air borne sound orthogonal to the front surface: active system off – red and on – black curve.

Conclusion

The active system has been implemented in a train wheel-set and tested on a rolling wheel test stand at Lucchini Sidermeccanica S.p.A. (Lovere, Italy). The wheel set has been driven in normal operation mode and also a set up of squealing has been realized. In both cases the axial acceleration on the wheel and the sound pressure in the near field have been measured simultaneously. The functionality of the active system has been investigated. The measurement results show a good functionality of the active system. The axial wheel acceleration is reduced by a factor of 2 – 8 in the squeal frequency. The corresponding airborne sound is reduced by 11 dB.

Acknowledgments

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