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EXPERIMENTS ON ACTIVE VIBRATION CONTROL OF A SCALE MODEL RAIL WHEEL

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ABSTRACT

The reduction of the vibration of a scale model rail wheel disk by means of piezo-electric patches was investigated. Two different approaches were studied. Firstly, semi-active methods with resonance-tuned damping networks consisting of resistors, inductances and the capacitance of the piezo-patch itself were used in order to absorb vibrational energy. Secondly, manually tuned analogue active feedback control was performed in connection with an accelerometer as error sensor, phase shifter, filter and high-voltage amplifier. Both approaches were investigated on a freely vibrating hanging scale model rail wheel excited by a shaker and on a scale model rail wheel built into a rolling test rig with a counter rotating disk used to simulate the excitation due to the rail. Details of the latter setup and results of the semi-active and active approaches will be given and discussed.

1 INTRODUCTION

Rail wheel disks are regarded as an important origin of noise radiated by trains. In order to reduce sound radiation as a first step, reduction of the vibration of the wheel disk was investigated by means of active and semi-active control. So-called piezo-patches were used for both, active vibration control by feedback of a vibration sensor signal via a controller to the piezo-element, and semi-active vibration control making use of energy conversion from mechanical energy into electrical energy inside the piezo-electric-element and then into heat by means of a suitable passive electrical network.

2 EXPERIMENTAL INVESTIGATION

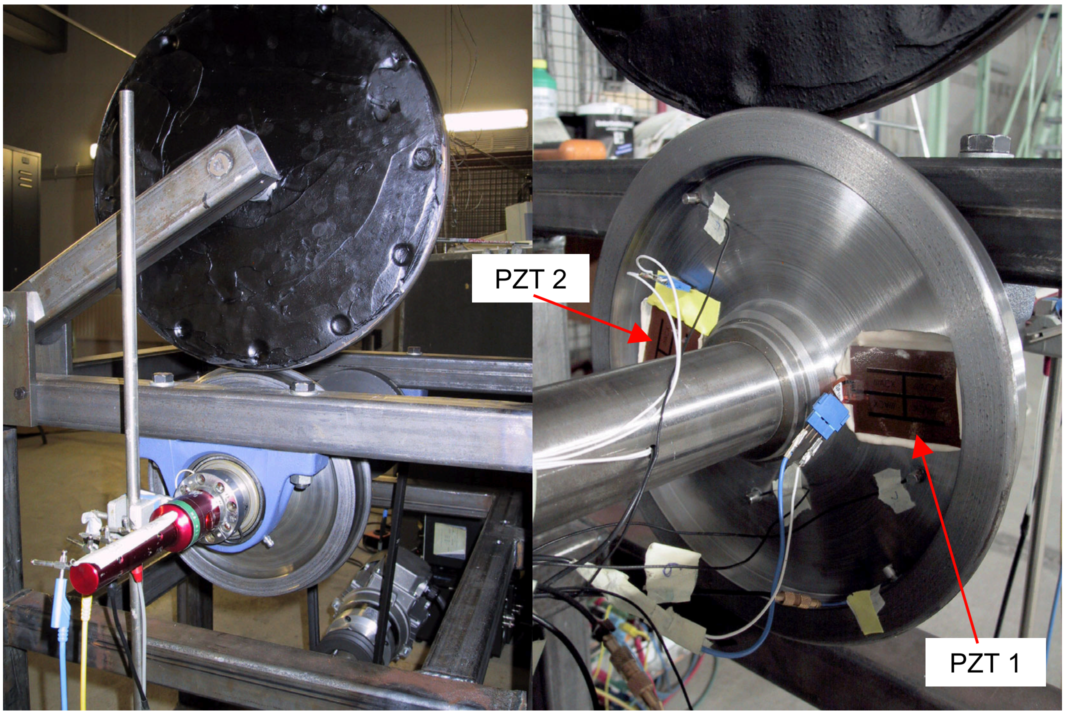

Fig. 1. Experimental setup: scale model rail wheel built into rolling test rig with damped counter rotating disk used to simulate the excitation due to the rail (left), position of piezo-patches and accelerometers on back side of wheel disk (right)

2.1 Experimental setup

A scale model rail wheel was mounted on a rolling test rig. It was driven by an electric motor and a drive belt (cf. Fig. 1). A counter disk made of steel and damped lightly with some bitumen coating was used to simulate the excitation of the wheel due to the rail. The counter disk could be lifted to investigate the wheel only. Two piezo-patches were attached to the wheel disk in-plane on its back side. One was oriented radially (PZT 1) and one was orientated circumferentially (PZT 2). Four accelerometers were attached on the wheels back side and rim. All signals were fed via a rotational transducer with mercury contact elements (red device in Fig. 1) either to the controller or the shunting network and to the measuring system.

2.1.1 Measurements without control

Velocity was measured on the wheel disk without active control. In the spectra below 1.6 kHz one main resonance appears and a lot of minor resonances. Thus, the resonance at 1428 Hz is mainly considered here and it corresponds to a (2,0)-mode, i.e. 2 radial nodal lines and no circumferential nodal line (cf. Fig. 2). The corresponding cut-out of the spectra is shown in Fig. 2. Without rotation and a lifted counter disk, i.e. without contact between wheel disk and counter disk, the velocity spectra cut-out shows a single resonance (green line). The excitation was done with an impulse hammer. Still without rotation but with the counter disk lowered, i.e. with contact between wheel and counter disk, a second minor resonance appears which is a resonance of the lightly damped counter disk (blue line). When rotation is switched on, the resonances split into two peaks each. This is because the mode shape seems to stand still, when observed from a point outside the test rig and the excitation remains at the top of the wheel disk. The wheel including the accelerometer is turning and thus moves through the mode shape. The frequency distance between the peaks can be explained by amplitude modulation with modulation frequency depending on the number of nodal lines, i.e. here 2, and the rotational speed (cf. [2]).

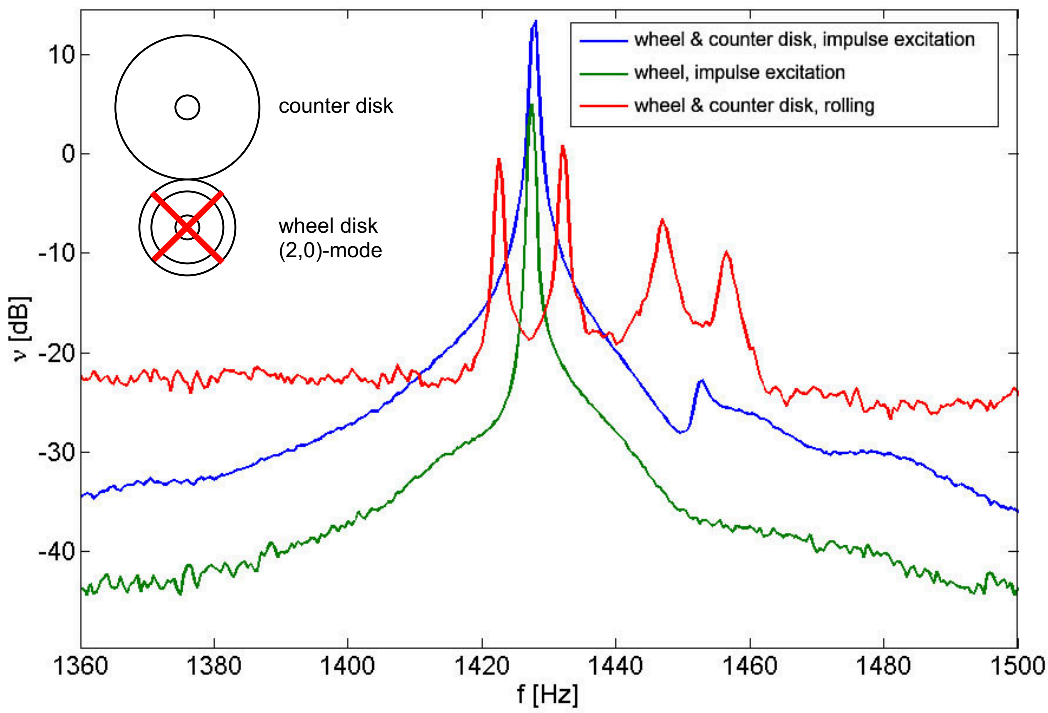

Fig. 2. Power spectra of velocity measured on resting wheel disk, both with and without counter disk, and measured on rolling wheel disk with counter disk, sketch of corresponding (2,0)-mode shape of wheel disk.

2.2 Resonance-tuned damping network

In conjunction with an electrical network the piezo-patch can be used to absorb vibrational energy. As a consequence of vibration the piezo generates free electrical charges at its connection terminals. If this electrical charge is discharged via a suitable electrical network, it can be converted into thermal energy and thus the vibrational energy is reduced. The amount of energy converted depends on the mechanical-electrical coupling between the pizeo and the structure of the electrical network. Most efficient are usually those networks which are characterized by a resonant behaviour. Here a simple series circuit consisting of a resistor and an inductance was connected to the piezo-patch. This is a 1-modal arrangement, which can be tuned to exactly one resonance frequency. The method to derive the neccessary resistance and inductance follows the investigations by Hagood and von Flotow [3]. Unfortunately the calculated inductance of several Henry was much too high for practically available parts, thus an electronic gyrator cuircit with an operational amplifier was used to simulate the inductance according to [4].

2.2.1 Measurement results

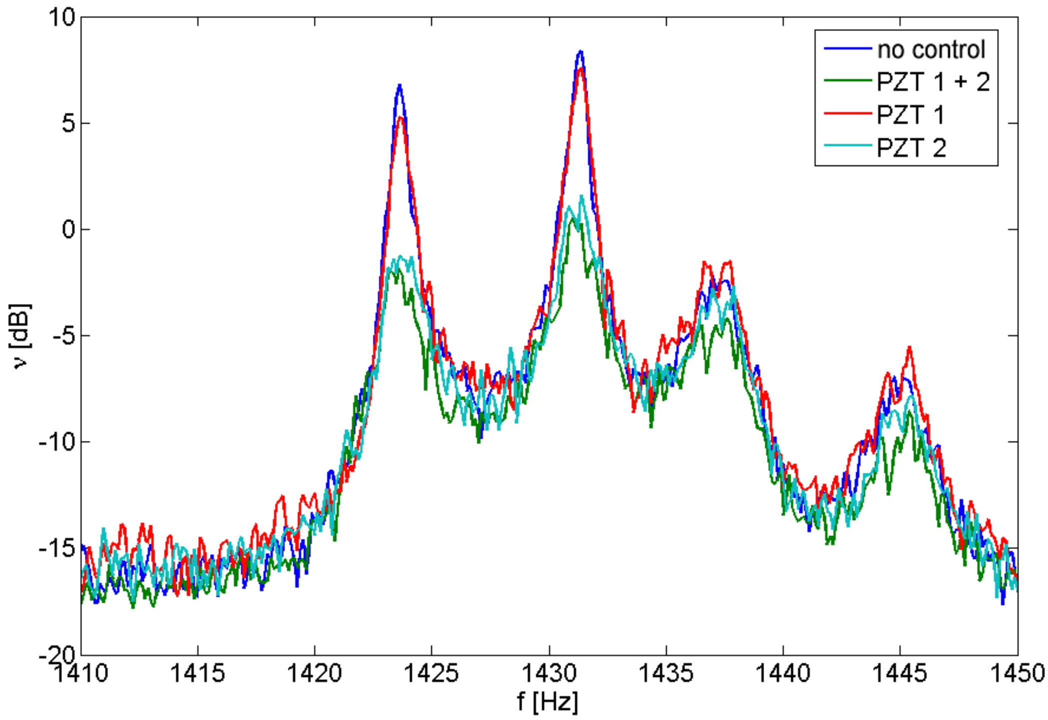

Fig. 2. Power spectra of velocity measured on rolling wheel disk with counter disk and resonance-tuned damping networks.

PZT 1 gave only 1-2 dB improvement in both modulation components, while PZT 2 gave 7 dB improvement in the higher modulation component and 9 dB in the lower component. The reason for the difference is that PZT 2 couples better with the (2,0)-mode of the wheel disk than PZT 1 because the (2,0)-mode is bending mainly in a circumferential direction and this is the orientation of PZT 2. Thus, the combination of both piezo-patches yields only very small improvements. Measured results were similar for all accelerometers on the wheel disk, thus global reduction was achieved.

2.3 Manually tuned feedback control

Manually tuned feedback control was also investigated. The signal of one accelerometer was fed back via a charge amplifier, a band-pass filter, a variable phase shifter, and a variable high-power amplifier to either piezo-patch PZT 1 or piezo-patch PZT 2. The center frequency of the band-pass filter was adjusted to include the considered resonance frequency in the pass-band. Phase and gain were adjusted manually during the rolling tests in order to achieve stable operation and to yield as much reduction in error signal as possible. However, care had to be taken to keep the system stable for all tested rotational speeds. In some cases it could be observed that the control loop was stable for a number of rotational speeds but became unstable when the wheel was stopped.

2.3.1 Measurement results

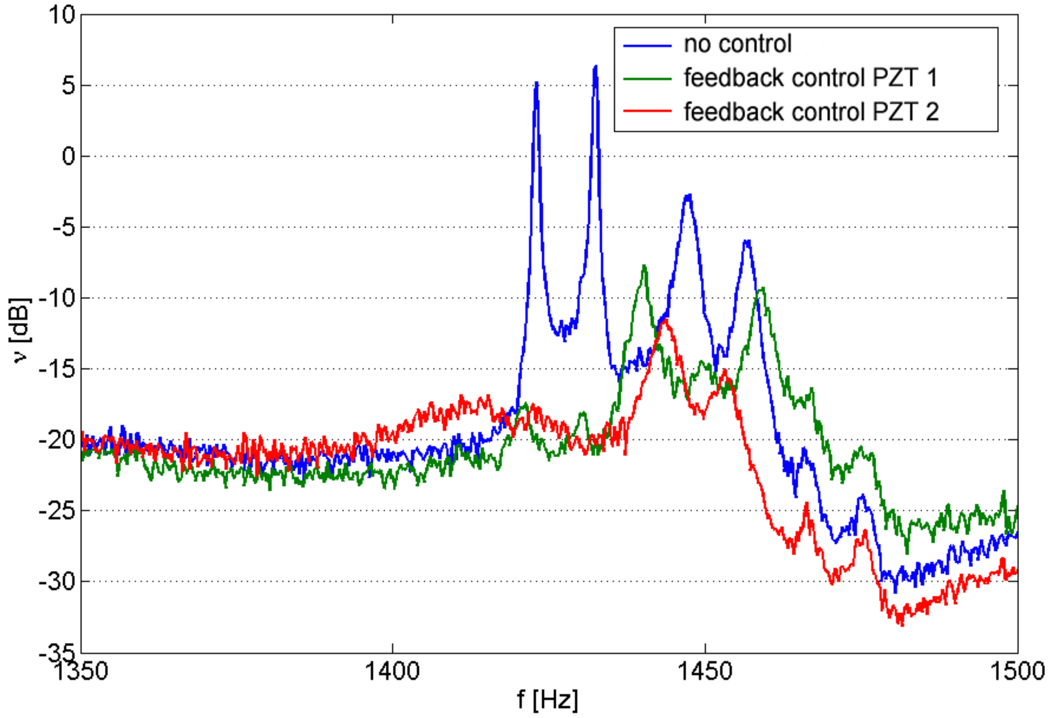

Fig. 3. Power spectra of velocity measured on rolling wheel disk with counter disk and manually tuned feedback controller.

When stable operation was maintained, reductions of more than 20 dB of the velocity in the region of the split resonance was achieved in the error signal. Results are shown in Fig. 3. Both, PZT 1 and PZT 2 performed equally well. Additionally, the two peaks of the resonance of the counter disk could be reduced by approximately 5 dB, PZT 2 performing considerably better than PZT 1. The reason is, again, the better coupling of PZT 2 to the (2,0)-mode.

The measured results held not only for the error signal but for all accelerometers on the wheel disk, thus global reduction was achieved. The tests were performed for three different rotational speeds, i.e. 87, 114 and 155 rpm and the results were very similar in all cases.

3 CONCLUDING REMARKS

Active control of vibration of a scale model rail wheel disk, built into a test rig with an electrical drive motor and a rotating counter disk to simulate the rail, was successfully applied to a representative dominant resonance at approximately 1428 Hz. Control was performed by means of piezo-electric patches. Both active and semi-active vibration control was investigated experimentally. With manually tuned active feedback control more than 20 dB in global reduction of the vibration of the wheel disk was achieved in the split resonance peak. With semi-active control the improvements depend very much on the orientation of the piezo-patch. The orientation must be "tuned" to the mode shape considered. In the best case improvements up to 9 dB of global velocity reduction was measured.

A comparison of both methods from a practicality point of view shows only small advantages of the semi-active method over the active method, i.e. somewhat simpler construction without the need for a vibrational sensor. But due to the high inductances needed, which are not realizable with passive parts, special gyrator circuits are needed, and thus both control approaches, the semi-active and the active, need a power supply. Keeping the latter point in mind, and considering the much higher global improvements of the feedback control approach, there is rather no advantage of the semi-active control approach here.

It might be interesting to investigate and compare both methods for the reduction of curve squeal as was done in [5] for the feedback controller.

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