Active vibration reduction applied to the compressor of an air-conditioning unit for trams

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Within the framework of the European Integrated Project InMAR (Intelligent Materials for Active noise Reduction) active vibration control approaches are applied to an air conditioning unit used to air-condition the driver's cab of a tram. Measurements previously performed by the manufacturer of the unit indicated that annoying tonal noise in the 50 Hz and 100 Hz one-third octave bands inside the driver's cab is mainly caused by the vibration of the compressor mounted in the unit on the roof of the tram.
Therefore, two different concepts for the design of active compressor mounts were developed that are used to reduce the vibration excitation of the unit's housing. The first one is an active mount based on four piezoelectric stack actuators and an elastomer part that deflects the effective direction of the piezo stack displacement by 90 degrees while amplifying the displacement by a factor of approximately 30. The second one is an active tuned vibration absorber whose natural frequency can be adapted by means of piezoelectric patch actuators and which behaves as a vibration compensator at higher frequencies. These two concepts are compared with each other.

1 Introduction

Electric locomotives of trains and trams are usually equipped with HVAC (heating, ventilation, and air conditioning) units that are placed on top of the locomotive's roof. These units are used to air-condition the driver's cab of the train or tram and consist of two heat exchangers, a compressor, a condenser fan, and a ventilation fan (see Fig. 1). These components cause vibrations and noise, often at an annoying level, both on the inside and on the outside of the locomotive.

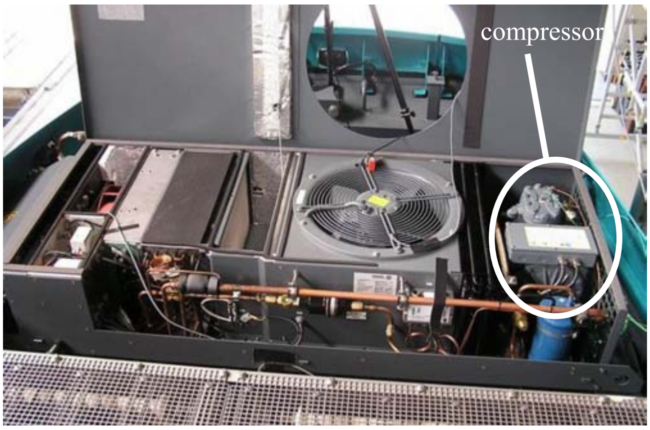

Fig. 1: General view of the HVAC unit mounted on top of a tram roof.

2 Annoying noise inside the driver's cab

The manufacturer of the HVAC unit performed noise and vibration measurements both inside the driver's cab of a tram and at the unit itself. It was possible to significantly reduce particularly annoying tonal noise in the 50 Hz and 100 Hz one-third octave bands inside the driver's cab simply by lifting up the compressor a couple of millimeters by means of a crane (see Fig. 2). This indicates that the compressor mounted in the HVAC unit on the roof of the tram is the main vibration source that causes the annoying noise levels inside the driver's cab.

Therefore, some experiments were performed that demonstrated the potential of active vibration control measures [1]. Based on these preliminary results two different con-

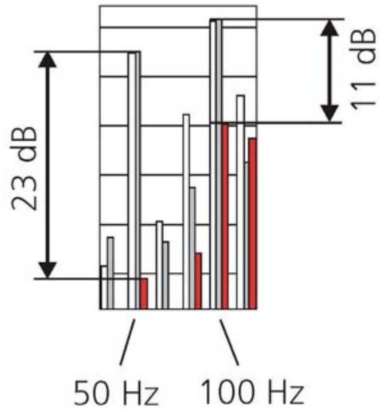

Fig. 2: Significant reduction of the annoying noise levels in the 50 and 100 Hz one-third octave bands inside the driver's cab by means of lifting the compressor.

cepts for the design of active compressor mounts were developed that are used to reduce the vibration excitation of the HVAC unit's housing. The first one is an active tuned vibration absorber whose natural frequency can be adapted by means of piezoelectric patch actuators and which behaves as a vibration compensator at higher frequencies. This active tuned vibration absorber is described in detail in [2] and [3]. The second one is an active mount based on four piezoelectric stack actuators and an elastomer part that deflects the effective direction of the piezo stack displacement by 90 degrees while amplifying the displacement by a factor of approximately 30 [4]. The design and the testing of this active interface are described in this paper.

3 Analysis of the compressor's vibration behavior

As a basis for the design of active noise and vibration reduction devices, various vibration measurements were performed at the HVAC unit. For this purpose the vibrations and the forces at the HVAC unit's compressor mounts were measured while the compressor was mounted to the bottom plate and while it was suspended from a rope (see Fig. 3).

Figure 4 shows a comparison of the measured acceleration level of the HVAC unit's bottom plate with the compressor mounted in the unit's housing and with the compressor suspended from a rope.

The displacement of the compressor's mounts can be calculated by integrating its acceleration twice. The displacement equals the maximum stroke an active interface should be able to provide (see Fig. 5). The maximum amplitude of the compressor's mounts (worst case scenario) can be calculated adding up the displacements at the individual frequencies, which results in approximately 65 µm. The maximum displacement of the compressor's mounts equals the double amplitude and is therefore calculated to be approximately 130 µm.

Fig. 3: Compressor suspended from a flexible rope.

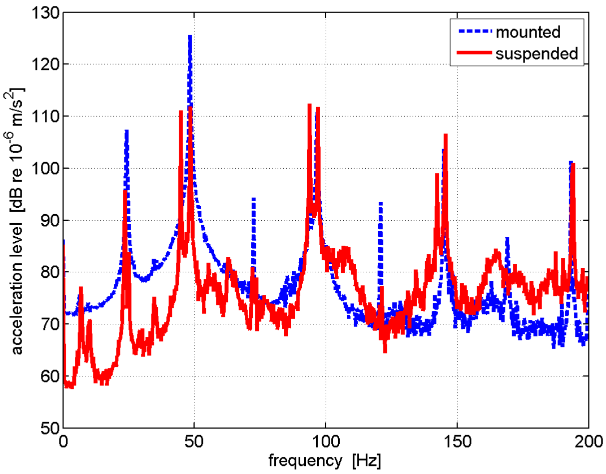

Fig. 4: Acceleration of the bottom plate: Comparison between the mounted and the suspended compressor.

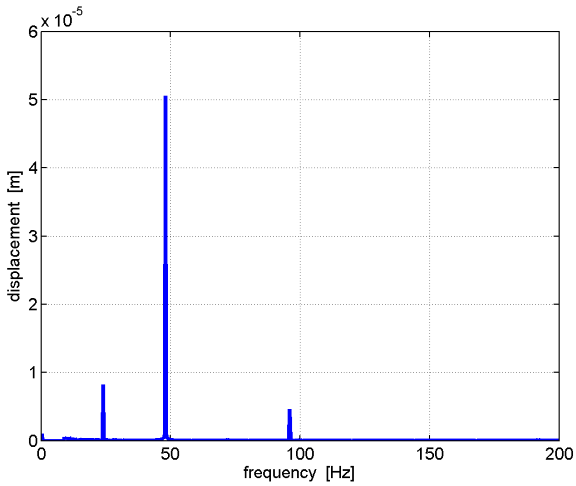

Fig. 5: Displacement of the mounts of the suspended compressor.

In order to measure the force the compressor exerts on the bottom plate, four force sensors are integrated in the system at its contact points. The measured force (see Fig. 6) equals the force an active device must provide. It can be seen that all active mounts together should be able to produce a force of at least 40 N.

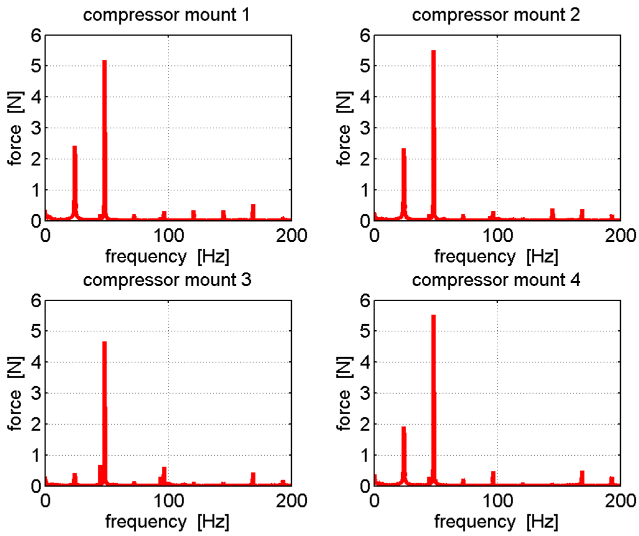

Fig. 6: Measured forces at the mount points of the compressor.

4 Design of the active interface

An active interface can act in different ways to reduce the vibrations of a vibrating structure. It can be used as an active damping device by means of reducing some of the resonance peaks without changing the vibration amplitudes at higher frequencies. Furthermore, it can be utilized to compensate the movement of the vibrating structure, thus causing vibration isolation to the structure whose vibration level is to be reduced. A literature survey revealed several different possible concepts that all have various advantages and drawbacks.

Due to the disadvantages of most concepts, such as installation height, durability, and costs, the hydraulic transmission seemed to be the most promising approach [4]. To avoid a leakage of hydraulic fluid this concept led to the use of a rubber-like substance as transmission medium (see Fig. 7).

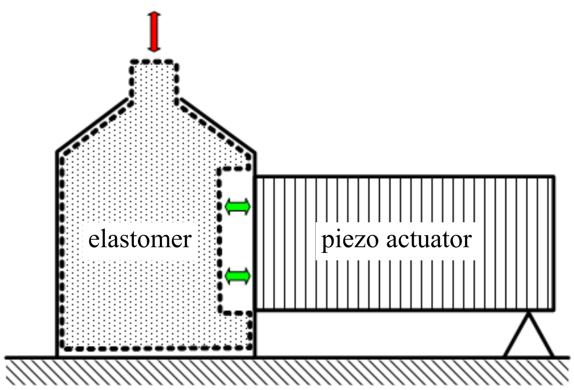

Fig. 7: Schematic view of a transmission by means of an elastic medium.

Such an interface exhibits various advantages: The amplification of the very low actuator stroke, the abolition of an additional stiffness, and the deflection of the actuators effective direction by 90 degrees, which results in a decreased installation height. By the use of an elastic medium the risk of a leakage is avoided.

5 Preliminary experiments

To determine the general behavior of a flexible transmission medium, a demonstrator was designed and filled with sanitarian silicon. The geometric transmission ratio of 1:45 is defined by the pistons diameters. Figure 8 shows the demonstrator embedded into a test rig.

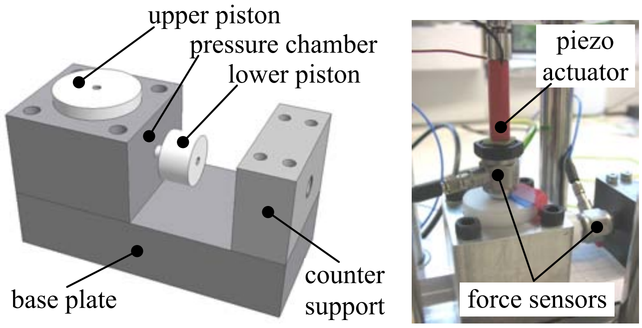

Fig. 8: Preliminary investigations on a transmission using an elastic transmission medium.

In spite of the improper medium, in which air bubbles are enclosed, promising results were achieved. As a result of the transmission, the free stroke of the actuator was amplified from approximately 40 µm to approximately 800 µm, which corresponds to a real transmission ratio of approximately 1:20. Based on this result, a more sophisticated transmission by means of a more effective flexible medium was designed.

6 Prototype of the active interface

Based on the results of the demonstrator a prototype of the elastomer transmission was designed (see Fig. 9). The pressure chamber is integrated in the structure of the interface and is filled with a two-component polyurethane cast elastomer. Four piezoelectric actuators[1] exert pressure on two opposed arranged input pistons made of polyoxymethylene (POM) with an effective surface of 800 mm^2 each. The output piston made of aluminum has an effective surface of 50 mm^2 and is firmly connected to the mount of the compressor. The effective surfaces equal a geometric transmission ratio of approximately 1:32.

7 Testing and validation

To obtain specifications of the elastomer transmission's behavior a test rig is built up. The active interface is firmly placed between a heavy bottom plate and a weight of 10 kg.

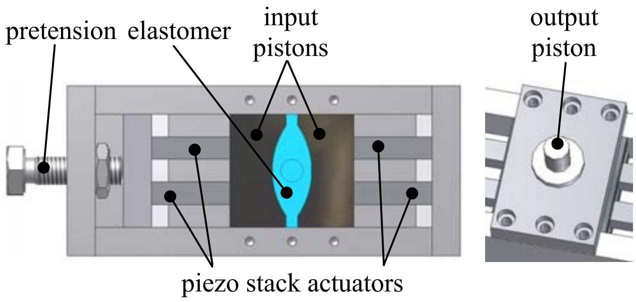

Fig. 9: Prototype of the elastomer transmission.

To measure the displacement of the mass and the force performed by the interface, a laser triangulator is placed above the test rig and a force sensor between the interface and the bottom plate.

A free stroke (without the mass) of the output piston of approximately 1 mm at a frequency of 1 Hz was measured. After putting the mass on top of the output piston the amplitude reduced to approximately 0.8 mm. Presuming 40 µm as the free stroke of the actuator an effective transmission ratio of 1:25 (geometrical transmission ratio as mentioned: 1:32) can be calculated.

The fundamental frequency of this system consisting of the mass and the interface is 18 Hz. At higher frequencies up to 200 Hz the interface exerts a force of approximately 35 N on both the bottom plate and the mass. In addition the phase between actuator voltage and interface force is constantly zero (Fig. 10), which is an important criterion for the integration of the interface into a control system.

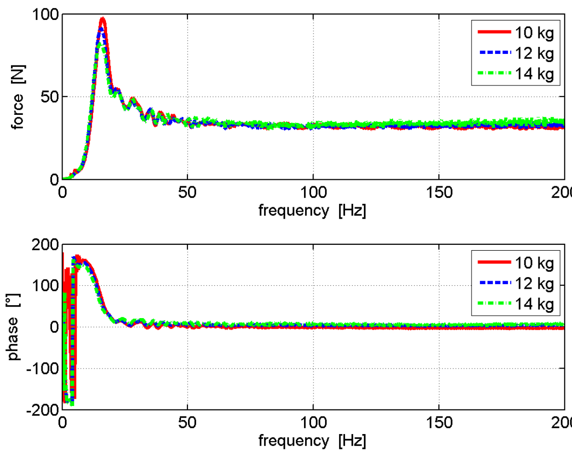

Fig. 10: Amplitude and phase response of the interface's force to an actuator voltage for various masses put on the output piston.

8 Control of the active interface

For the control of the active interface two different control strategies are implemented as depicted in Fig. 11. The first control strategy is a feedback of the bottom plate's acceleration $a_{\text{bottom plate}}$. By this means both the mass forces (voltage signal U_a) and, by integrating the acceleration signal, the damping forces (voltage signal U_v) of the active interface system can be "virtually" varied. This strategy acts as an active damping device and possesses advantages in the resonances of the system.

To reduce the vibrations outside of the resonance frequencies a feed forward control is implemented. For this pur-

[1] CeramTec 32.4×7×7 mm^3; ceramic type: SP 505

pose the acceleration signal of the bottom plate is lead through a module for the identification of the excitation frequency and its phase (voltage signal U_c). A combination of all voltage signals $U_{interface}$ is applied to the actuators and results in a correlating force, which is induced in the system (see Fig. 11).

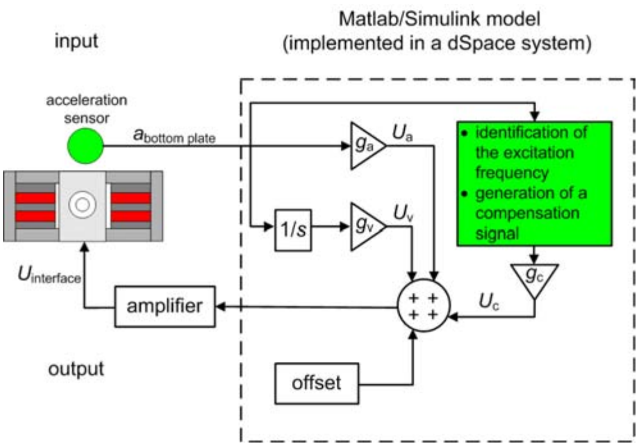

Fig. 11: Control circuit of the elastomer transmission.

9 Measurements results

To achieve a statically defined mounting, three active interfaces are placed beneath the compressor as illustrated in Fig. 12. Next to each mount an acceleration sensor is attached to the unit's bottom plate in order to feed back the acceleration signal and to monitor the vibration behavior of the bottom plate as well as the control success.

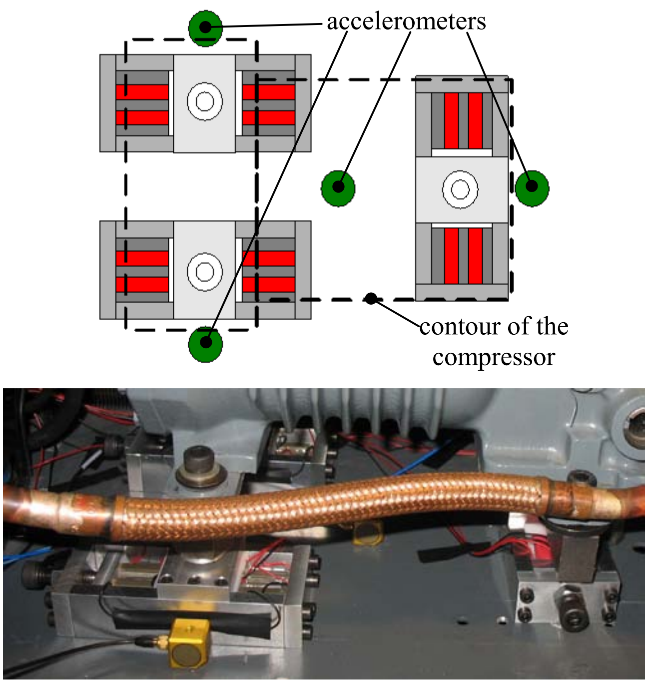

Fig. 12: Active interfaces mounted beneath the compressor (top: schematic view from above, bottom: photograph).

This way a broadband reduction of the acceleration level as well as a reduction of the acceleration level at single frequencies up to 14 dB is achieved (see Fig. 13).

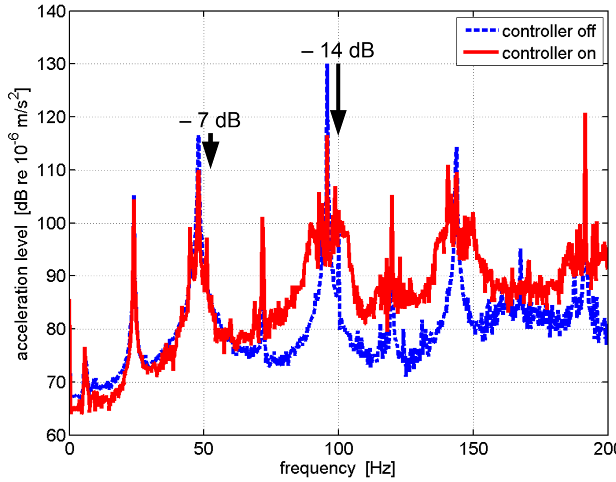

Fig. 13: Reduction of the acceleration level beneath the compressor.

10 Comparison with an active tuned vibration absorber

For the same purpose and based on the same analysis of the compressor's vibration behavior as described in Section 3 an active tuned vibration absorber was designed. The design, which was inspired by [5], and the testing of this active tuned vibration absorber are described in detail in [2] and [3]. The vibration absorber consists of two discrete masses attached to the ends of two cantilevered beams (see Fig. 14) and is tuned to 50 Hz, which is very close to the frequency of the highest vibration level of the compressor.

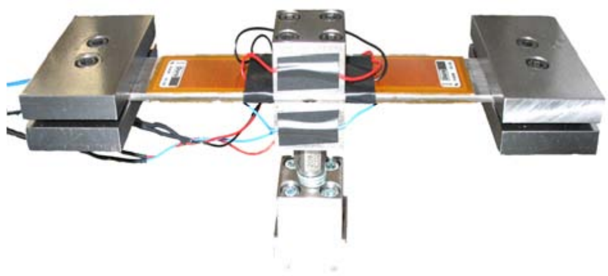

Fig. 14: Prototype of an active tuned vibration absorber.

This fundamental frequency can be varied within a certain range by an acceleration feedback control system using the acceleration of the discrete masses as input to the voltage signal applied to the piezoelectric patch actuators attached to the cantilevered beams (see Fig. 14), thus virtually adapting the mass forces of the passive absorber. In addition, the acceleration at the mounting point can be used as the input to another control system such that the absorber behaves as a vibration compensator at higher frequencies, using the inertia of the masses at the end of the cantilevered beams to generate a force at the mounting points.

The total mass of each absorber is 1.7 kg. The final design (see Fig. 14) is quite simple and very robust. The absorber mounts rigidly connect the compressor feet to the HVAC unit's housing, which is an important safety issue (see Fig. 15). The fundamental frequency of the vibration absorbers can be varied in the range –12 Hz/+3 Hz by applying an appropriate voltage signal to the patch actuators. In the higher frequency range beyond the fundamental frequency up to about 200 Hz the vibration absorbers can act as vibration compensators and can generate forces of approximately 11 N each.

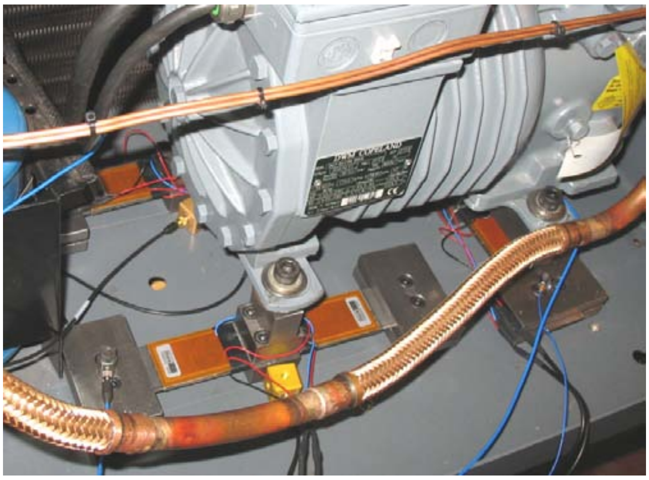

Fig. 15: Active tuned vibration absorbers mounted beneath the compressor.

The acceleration levels of the HVAC unit's housing directly beneath the compressor are depicted in Fig. 16. The blue and red lines show the acceleration levels with the controller being switched off and on, respectively. Two effects can be seen: At 48 Hz the acceleration levels are reduced by 15 dB by means of the passive vibration absorber whose fundamental frequency is actively tuned to match exactly the frequency of the highest peak.

At 119 Hz and 191 Hz the vibration absorbers act as vibration compensators, thus reducing the acceleration levels by 10 dB. At these frequencies an appropriate anti-phase sinusoidal voltage signal is applied to the piezo actuators. The required amplitudes and phase shifts are determined automatically by a digital control system. The peaks at 119 Hz and 191 Hz were chosen arbitrarily in order to demonstrate the potential of the compensator effect – other or more peaks could be reduced as well.

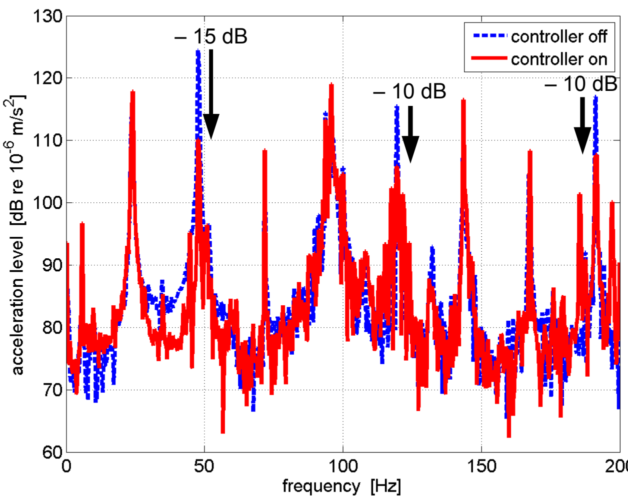

Fig. 16: Significant reduction of the acceleration level beneath the compressor at 48 Hz and other frequencies.

11 Summary and conclusions

Both concepts, i.e., the active interface based on four piezoelectric stacks and an elastomer on the one hand and the active tuned vibration absorber on the other hand, are capable of effectively reducing the vibrations tranmitted from the compressor to the HVAC unit's housing. Whereas the active interface is able to produce broadband effects and to reduce the vibration level in a wide frequency range beyond its fundamental frequency (approx. 18 Hz), the active tuned vibration absorber can reduce the peak at its fundamental frequency (approx. 50 Hz) and the peaks in the frequency range beyond that, but only in a narrowband way.

The effectiveness of both concepts should be verified by sound pressure measurements in-situ in the driver's cab of the tram while the tram is in operation. Both concepts could be integrated into an automatically controlled network of active interfaces or active tuned vibration absorbers such that the mutual influence of several active devices can be compensated automatically.

Acknowledgments

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