Active vibration control applied to a tram HVAC unit

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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 with open lid (the compressor can be seen behind the grille in the foreground).

Annoying noise and vibrations

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. 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. The dynamic structural behavior of the unit was further analyzed by means of various vibration measurements.

Control algorithm

To estimate the potential of active vibration reduction approaches, additional control forces are applied to the compressor by means of an electrodynamic shaker (see Fig. 2). The force applied by the shaker is measured with a force sensor at the excitation point. An accelerometer right next to the excitation point detects the vibrations on top of the shaker that are to be reduced.

Fig. 2: The electrodynamic shaker attached to the top of the compressor (with force sensor and accelerometer).

The shaker is controlled by a "modified error" algorithm as part of a digital adaptive internal model control running on an embedded industrial PC system.

The aim of the control algorithm is to generate a suitable signal to eliminate the vibrations of the compressor by minimizing the error signal measured by the accelerometer. Since the compressor is powered by an asynchronous machine in a fully enclosed casing, the engine speed is not available as a reference signal for a feed forward system. Therefore, a feedback configuration for the control system was chosen.

Figure 3 shows a block diagram of the algorithm. The disturbance signal $d(n)$ represents the measured vibration of the compressor generated by filtering an (unknown) disturbance source $x(n)$ with the primary path of the structure $P(z)$. The control path from the actuator to the sensor signals is represented by $S(z)$. The accelerometer signal $e(n)$ therefore denotes the superposition of the disturbance $d(n)$ and the control signal $y(n)$.

Since a reference signal is not available, a model of the secondary path is used for the estimation of the disturbance signal $\hat{d}(n)$ by subtracting the part of the error signal that results from the actuation. Thus, this method, called internal model control (IMC), removes the feedback effects from the control system.

The model $\hat{S}(z)$ is also used for the adaptation of the control filter $W(z)$ in a modified error configuration. Compared to the filtered-x LMS algorithm [1] one control filter $W(z)$ is used for the adaptation and another one for the generation of

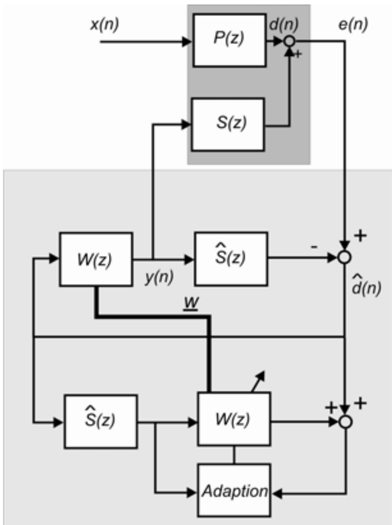

Fig. 3: Block diagram of the adaptive internal model control algorithm with modified error configuration.

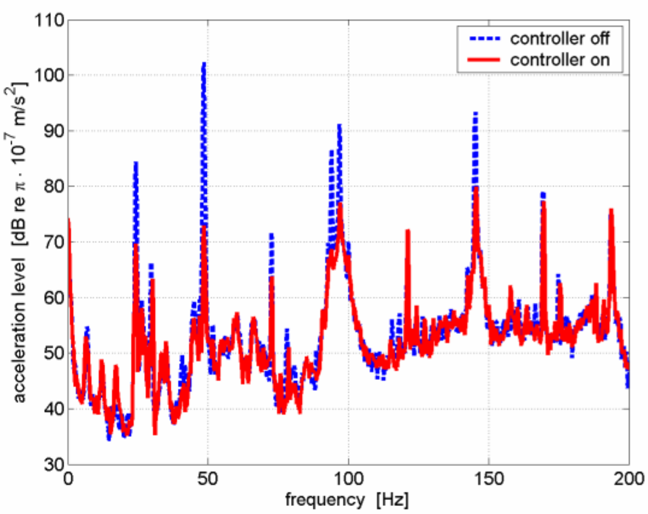

Fig. 5: Significant reduction of the acceleration level at the top of the compressor at 50 Hz and other frequencies.

the actuator signal *y(n)*. The filter weights are copied from the adaptation filter to the control filter. Due to this separation the secondary path and the control filter can be permuted and the error signal for the adaptation can be calculated directly at the filter output. An advantage of this arrangement is an improved speed of convergence [2].

The complete control system was implemented in Matlab/Simulink and tested on a PC/104 [3] based embedded PC system (see Fig. 4). The PC system mainly consists of a Pentium III 700 MHz processor with standard hardware components and an analog I/O card. The resolution of the I/O card is 16 bit for the input A/D converters and 12 bit for the output D/A converters.

Fig. 4: Embedded PC system (PC/104 [3]) used for control.

First results and next steps

The control configuration described above is able to significantly decrease the vibrations of the compressor. Figure 5 depicts the spectrum of the acceleration level on top of the compressor. As can be seen the acceleration level at 50 Hz is reduced by approx. 30 dB. At other frequencies the acceleration level is reduced by up to 15 dB. These vibration reductions require a maximum excitation force of approx. 40 N.

These first results demonstrate the potential of active vibration reduction measures. As a next step two different active vibration reduction devices will be developed and compared with each other. The first one is similar to a tuned vibration absorber that is attached at the compressor mounts. It consists of a leaf spring with piezoceramic patch actuators glued to its surface and masses attached to its ends [4]. The piezo patches cause the masses to vibrate, which in turn generates a vertical force at the center of the spring that counteracts the vibrations of the compressor. The second device is an active compressor mount based on piezoceramic stack actuators.

In both cases the effectiveness of these approaches should be verified by sound pressure measurements in-situ in the driver's cab of the tram while the tram is in operation. Microphone signals at the driver's cab can serve as additional controller inputs besides acceleration signals from the compressor.

Acknowledgment

The research presented in this paper was performed in the framework of the integrated European project "InMAR – Intelligent Materials for Active noise Reduction" (grant NMP2-CT-2003-501084 of the European Union) [5]. The financial support by this project is gratefully acknowledged.

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