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THE INFLUENCE OF DAMPING LAYER ARRANGEMENT ON THE EFFICIENCY OF VIBRATION AND NOISE DAMPERS

VLIV USPOŘÁDÁNÍ TLUMÍCÍCH VRSTEV NA ÚČINNOST TLUMIČŮ VIBRACÍ A HLUKU

Abstrakt:
Článek pojednává o experimentálním výzkumu vlivu uspořádání tlumících vrstev tlumičů vibrací a hluku na jejich účinnost. Výzkum byl proveden na zjednodušeném modelu železničního kola a šesti variantách tlumičů s různým uspořádáním tlumících vrstev. Účinnost tlumičů byla posouzena na základě průměrných mobilit a průměrných autospekter akustického tlaku, jejich 1/3 oktávové analýzy a 1/3 oktávové analýzy akustického výkonu.

Summary:
The paper deals with experimental research on influence of arrangement of damping layers of vibration and noise dampers on their efficiency. The research was carried out on the simplified model of railway wheel and with six types of dampers with different arrangement of damping layers. The efficiency of the dampers was evaluated on the basis of averaged mobilities and averaged autospectra of acoustic pressure, on their 1/3 octave analysis and on 1/3 octave analysis of acoustic power.

1 Introduction

A research on vibration dampers efficiency was started few years ago at the Department of Mechanics of Technical University of Ostrava on demand of the manufacturer of railway wheels, because there is a permanent demand to reduce noise produced by wheels. A railway wheel was simulated by a simplified model - an annular plate of outer diameter 500 mm, inner diameter 40 mm and thickness 10 mm. It was clamped on a solid pin, which was fastened to the foundation. Six various dampers were tested with this model. All of them were plate dampers with the same both outer and inner diameters (500/60 mm). They were fastened to the model with the help of 16 screws equidistantly placed on the outer diameter using ring washer between the model and the damper. Specification of the used dampers is stated in the Table 1.

2 Measurement set-up

Following instruments have been used: dual channel analyzer BK2032, accelerometer BK4501, force transducer BK8200, charge preamplifiers BK2635, dynamic exciter LDS400, power amplifier PA100E, sound intensity probe with microphone pair BK 4181 with the spacer 8,5 mm. Vibrations of the model were excited using random noise signal in the frequency range from 0 to 1600 Hz. The exciter was connected to the model using a thin driving rod at a point distanced 10 mm from the outer diameter.

Vibration response was measured at 81 points on the model and at 16 points on the damper using the accelerometer. Vibration signal from the accelerometer was integrated in the dual channel analyzer to obtain velocity signal, which is more suitable for comparison with noise measurements. Frequency response functions (mobilities) were stored and used for further processing.

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Acoustical measurements were carried out at the distance of 0.5 m from the vibrating surface of the model in 17 measurement points. For all of the tested dampers, acoustic measurements were performed on the side of the model, on which damper was mounted. Sound pressure levels and acoustic intensity were measured.

Dampers ϕ 500/60						
No	1/1	1/2	2/1	2/2	2/3	3/1
Sect. view	2 5	2 10	1 2 2	1 5 2	1 10 2	5 2 5
steel	rubber					

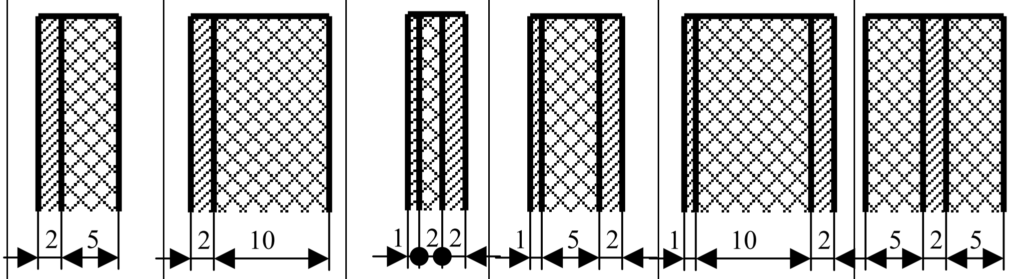

Table 1: Table of damper types

3 Measurement results

3.1 Modal properties

Damping ratios (in the form of fraction of critical damping) for all tested configurations are stated in the Table 2.

Averaged fraction of critical damping							
Damper	1/1	1/2	2/1	2/2	2/3	3/1	without
ξ [-]	0,0143	0,0180	0,0161	0,0170	0,0235	0,0138	0,0043

Table 2: Averaged values of fraction of critical damping in the frequency range 0 to 1600 Hz

3.2 Averaged mobilities

Frequency response functions (mobilities) were measured in 81 points on the model and in 16 points on the damper. Averaged mobilities were calculated from all the 81 (or 16) measurements, special program in C language was written to do that. Autospectra of sound pressure levels were measured at 17 points (at the same 16 points as vibration measurements + 1 point in the center of the plate). These measurements were averaged at the same manner as vibration measurements.

As the range of the paper is limited, not all of the above mentioned results will be presented here. Two types of dampers are used for illustration of the results – 1/2 and 2/2 (Fig.1).

3.3 Third octave analysis

For better comparison of vibration and noise measurements, one third octave analysis was performed both for mobilities and sound pressure level measurements. The averaged functions were used for this analysis. In addition, total values in the entire frequency range were calculated. The results are shown at the Fig.2.

4 Conclusions

From the point of view of vibration and acoustic emission suppression, following general conclusions can be dedicated from the measurement results:

a) In the explored range of the ratio of the thickness of the damping layer (rubber) to the thickness of the supporting layer (steel), increasing of this ratio leads to increasing of the damper efficiency.

b) Dampers equipped with the covering steel layer of the thickness 1 mm are considerably more efficient than dampers with the damping layer of the same thickness only.

c) The arrangement of damping layers on both sides of the supporting steel layer is disadvantageous from the point of view of dampers efficiency in comparison with the damping layer placed on one side of the supporting layer.

d) The size of acoustic emission is considerably influenced by dampers vibration.

Reviewer: Prof. Ing. Karel Bailotti, CSc.

Acknowledgements

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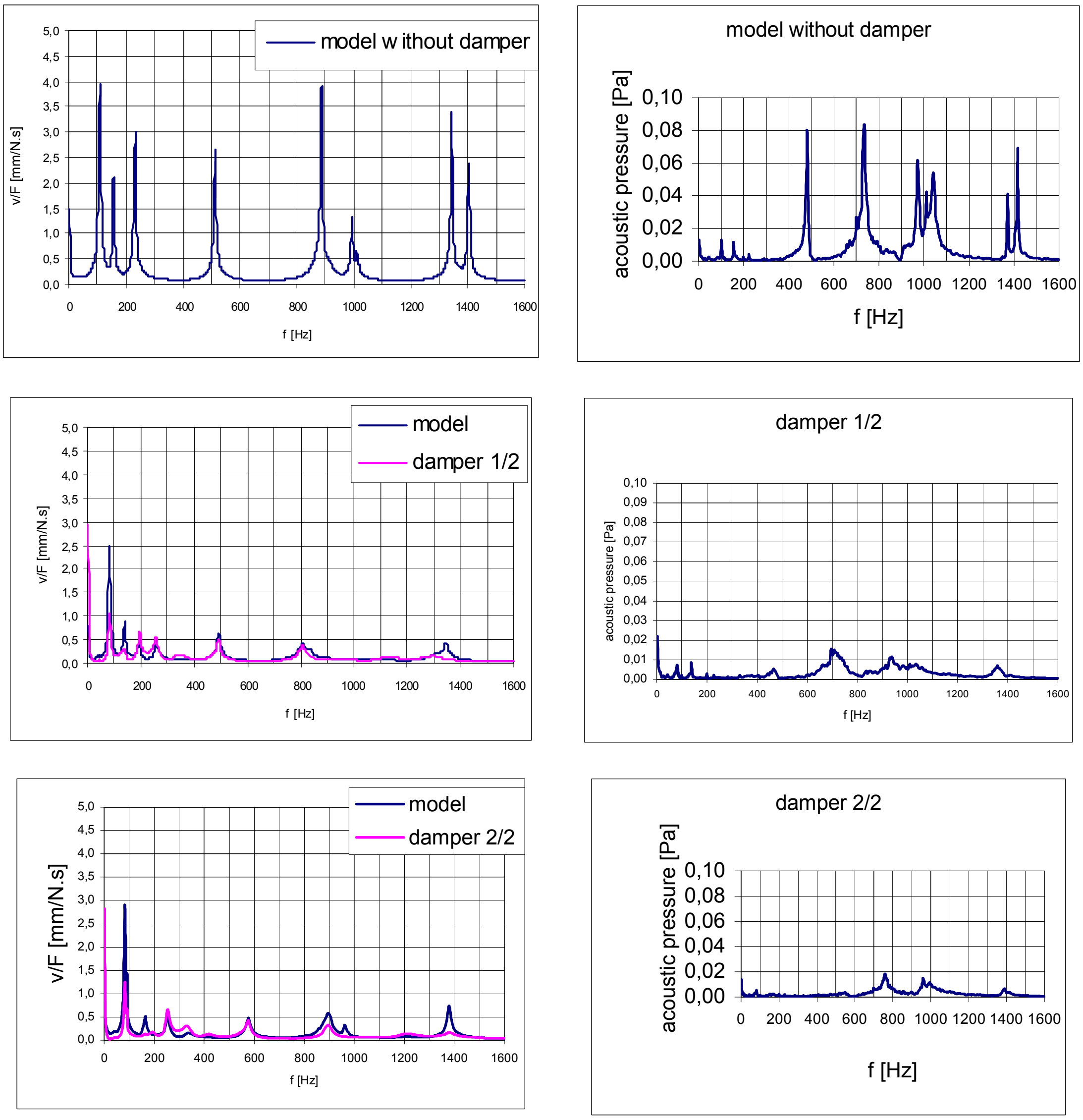

Figure 1: Averaged mobilities (left column)
and averaged autospectra of acoustic pressure level (right column)

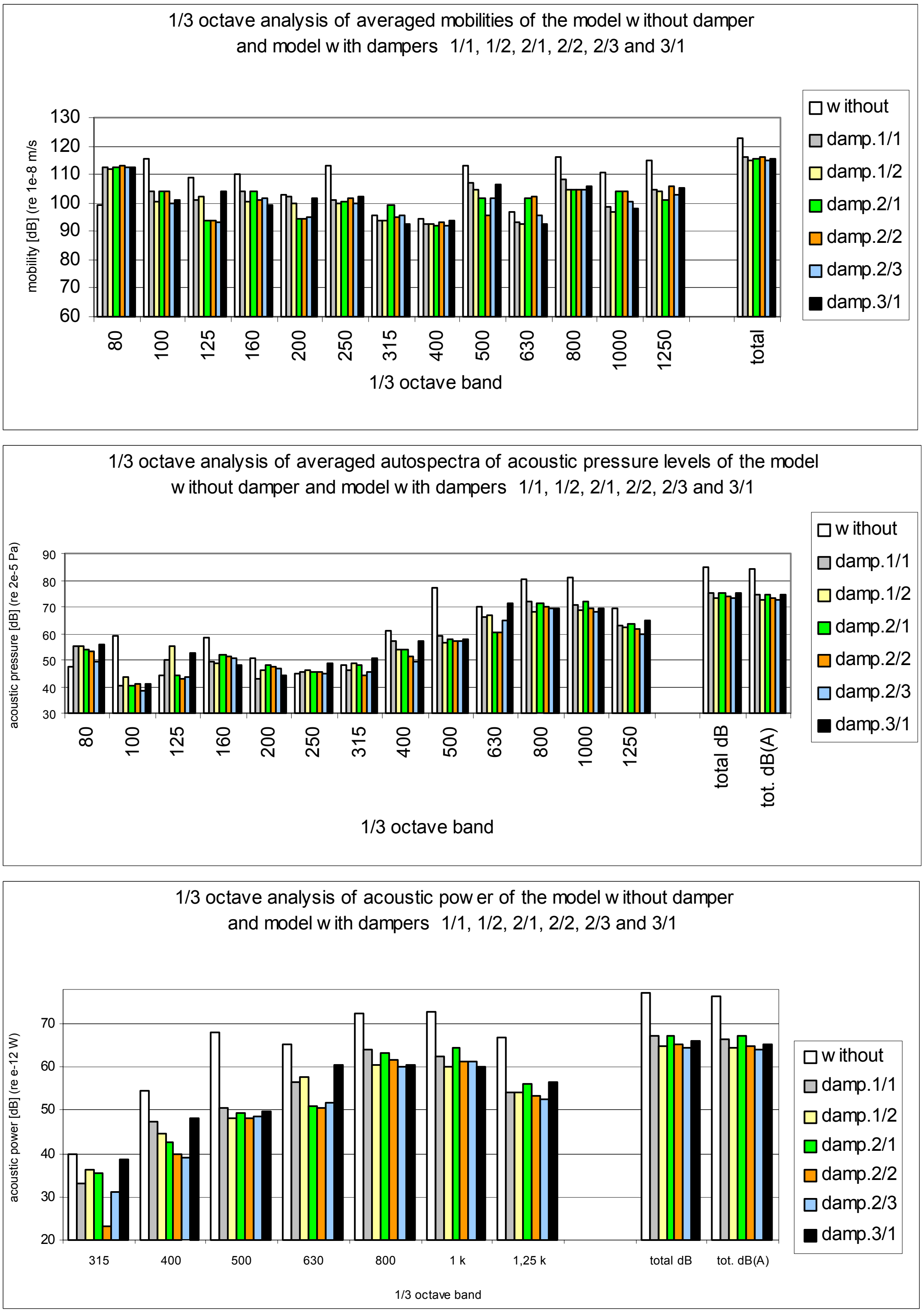

Figure 2: One third octave analysis of the averaged mobilities,
of averaged acoustic pressure levels and of acoustic power