Transmission of Vibroacoustic Energy through the Structure of a Car Body into the Protected Area and its Control

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Summary

The main aim of the paper is to investigate the conditions for the transmission of the dynamic loading (vibration and noise) from the vibration sources, mainly combustion engine, throughout the structures of the car body into the protected space (cabin) of the passenger's car. The solution of this problem includes the methodology of experiments for the given structure, using measuring equipment, measurements of vibration (or force) at the sources and on the transmission path of structural noise, as well as noise measurements in the protected area. The results of the vibration and noise measurements are compared to the modal analysis to obtain the possible resonance sources of the car body and/or assessing influence of the car safety parts (reinforced doors, body roof, deck) on the generated vibroacoustic energy transfer into the protected area of the passenger car. The measurements were made for the passenger car at rest at the most frequent operational rotational speed. Sound pressure level (noise) and mechanical vibration were measured and FFT analysis was used for the detection of vibroacoustic energy. In the first place, the low-frequency noise sources are specified and the direct effects on the human being are investigated. Finally, this paper suggests some measures which can have an impact on the reduction of the non required vibroacoustic energy in the passenger's cabin.

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1. Introduction

Isolation and structural damping constitute the two most widely applicable means for the control of vibration or structure-borne sound, particularly in the audio frequency range. Vibration isolation in essence involves the use of a resilient connection between a source of vibration and an item to be protected, so that this item vibrates less than it would, if a rigid connection was used. In a typical situation, the source consists of a vibration machine or structure and the item to be protected is a human being, environment, instruments, machines, and so on. Many salient features of vibration isolation can be analysed in terms of a simple model consisting of a rigid mass that is connected to a support via a vibroisolator (resilient element) and that is constrained to translate along a single axis [1, 2, 3]. More complex models are needed to address situations where the magnitude of excitation depends on the motions, where an additional vibroisolator-mass system is inserted between the primary one and the support, and/or at comparatively high frequencies where the vibroisolator mass plays a significant role or where the isolated items do not behave as rigid masses. Other complications arise because of non-uniaxial motions and nonlinearities.

2. Goals, object of investigation, instrumentation and methodology

2.1 Goals of the Study

The goals of the study were to investigate and analyze the transmission of the structure-borne vibroacoustic energy flow of a deterministic excitation on a subtle (2D) structure (cabin of vehicle). The study shows that the low frequency excitation wave can cause resonance state at higher frequencies, as well, which can be dangerous for the undesirable dynamic loading of structures and these

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structures can be relatively very noisy in respect to their surroundings. Frequency spectrum of the measured signal and Eigen (natural) frequencies of the Eigen modes were compared and analysed for the investigated objects, as well. This paper only mentions the energy flow throughout engine bed.

2.2 Object of investigation

The measurement structure-borne vibroacoustic energy flow was performed on the passenger car at defined operational conditions with the goal to determine the transmission and noise transmitted from the driving-gear as a vibration source through the surrounding components of the body car, mainly roof, doors, and deck [4, 5]. The main source was a gasoline powered internal combustion engine running closely to frequent speed 4 000 rpm together with the gear box. The passenger car was at stationary position.

2.3 Instrumentation

The deterministic signal was generated by an internal combustion engine and the consequent response on the body car was performed using the FFT analyzer PULSE Bruel & Kjaer platform. This portable analyzer represents the system which guarantees reliable measurement processes, analysis, and evaluation. The system consists of a piezoelectric accelerometer with a frequency range from 1 Hz to 10 kHz (amplitude ±10 %), modal hammer, and display and memory module. To identify the energy dominant Eigen frequencies more precisely, the fast Fourier transform (FFT) analysis was carried out using the FFT analyzer PULSE. The methodology presented in the article can also be applied to other excitation sources of low frequency vibration. Sensor mounting on the structural elements of the investigated objects must coincide with the ISO 5348 for accelerometers and with respect to past experience [6]. The goal is to ensure that the sensor (acceleration, force) correctly reproduces the motion of the analysed component without interfering with the response.

The sound pressure level was analyzed using Sound Level Meters Bruel & Kjaer 2250 platform.

Other than the frequency range, for the type of signal, it is also very important to select the appropriate type of averaging as well and number of averages per unit time as well as a suitable time window [18, 19].

2.4 Methodology of the measurement and determination of the transmission loss

The measurement of the transmission loss assumes that the material behaves in a linear way and that it has negligible mass compared with the mass loading. The method determines the impedance of the material when loaded by a mass providing a compression force equivalent to that found when the resilient material is pressed between the bed engine and chassis. This is done by measuring the transfer function of the mass-loaded material at all the required frequencies [3, 7, 8, 9]. The method uses a vibration excitation system below which the resilient material (element) is placed with the loading mass m on the bottom (Fig. 1). Two accelerometers measure the vibration on the generator of vibration (bed of combustion engine), a_1, and the vibration of the mass m (part of the car body), a_2. Assuming the test resilient material (e.g. silentblock) has negligible mass, the equation of motion is

$$m\ddot{x}_2 = -Z_{\mathrm{m}}(\dot{x}_2 - \dot{x}_1) \tag{1}$$

from which the mechanical impedance is

$$Z_{\mathrm{m}} = \frac{j\omega m}{\left[\dfrac{A_1}{A_2}\right] - 1} \tag{2}$$

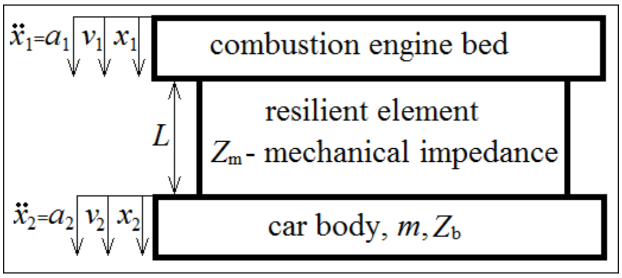

Figure 1. Theoretical model for determination of the transfer function and/or transmission loss.

When the part of the car body under the silentblock is considered only, the equation of motion is

$$Z_b\dot{x}_2 = -Z_{\mathrm{m}}(\dot{x}_2 - \dot{x}_1) \tag{3}$$

from which the required transmissibility of vibration or transfer function when the resilient material (silentblock) is loaded by the car body system (see Fig. 1) is then computed

$$T = \left|\frac{\dot{x}_2}{\dot{x}_1}\right| = \left|\frac{Z_m}{Z_b + Z_m}\right| \tag{4}$$

It is more suitable to directly determine the transmission loss D (in dB) of the resilient material which can be calculated from the formula

$$D = 10\lg a_1^2 / a_2^2 \text{ or } D = 10\lg \dot{x}_1^2 / \dot{x}_2^2 \tag{5}$$

Another method is to determine the *damping capacity* ψ of a vibrating system which can be defined as $\psi = \Delta E/E$, where ΔE represents the energy that is removed from the system per cycle and E denotes the vibration energy stored in the system. The *loss factor* η, defined by $\eta = \psi/2\pi$, represents the ratio of the energy removed per radian to the stored vibration energy. In most practical situations, the loss factors less than 0,2 and one may take the stored energy to be equal to the total vibration kinetic and potential energy. In general, the various measures of damping are related to each other by

$$\eta = \frac{\Delta E}{2\pi E} = \tan\phi \qquad (6)$$

The phase angle ϕ by which the displacement of the mass of the isolator lags the excitation force may also be used to characterise the system's damping. For example, at radian frequencies ω much below the system's natural frequency $\tan\phi = k_i/k_0$. Where k_i is the imaginary part of complex stiffness

$$k = k_0 + jk_i. \qquad (7)$$

The methodology presented in the section can also be applied to other sources of very low frequency acoustical vibration, as e.g. air-conditioning systems, boiler systems, large low-frequency Diesel engines, etc.

3. Results and discussion

3.1. Analysis of sound

As a result, the chassis of the vehicle was excited by a deterministic signal for a defined interval of time. From the analysis of the frequency spectra, a strong oscillating frequency was observed, which generated low frequency acoustic waves within the cabin (Fig. 2), where the one-third-octave band and FFT analysis in interior of the passenger car is carried out. Frequency analysis of the investigated low-frequency region with application of A- and Z- weightings as presented in Fig. 2 shows the frequency conformity of the results for constant speed. The utilization of the Z-weighting (i.e. no weighting) shows the exposition of the human being directly to this noise, regardless of the sensitivity of his/her ears.

The sensitivity of the human ear at low frequencies is much lower; therefore also the measured results, weighted using the A- as well as C- or Z-weightings, are significantly different. Currently, there is a discussion about the evaluation of the low frequency noise of high sound pressure levels, since the A-weighting, which are most often used, do not reflect the correct influences on health and comfort of human beings, in general [10, 11, 12, 13].

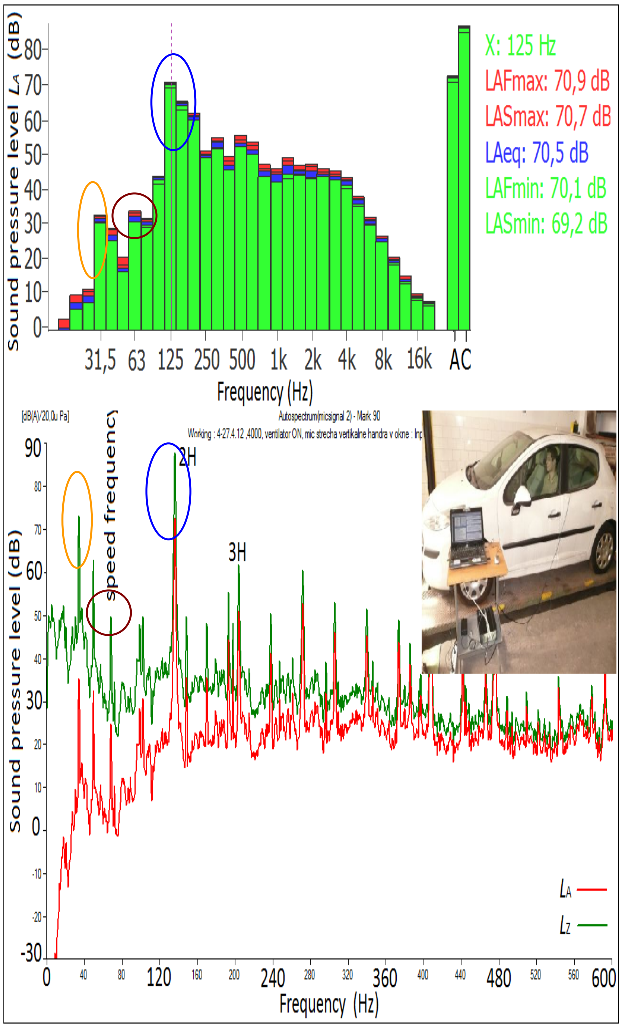

Figure 2. One-third-octave analysis and FFT analysis of a subtle car body at a given operational speed of the combustion engine using A- and Z- weightings.

If the one-third-octave band analysis and FFT analysis are compared, we can state that the determining contribution to A-weighting sound pressure level of the transmitted structural-borne noise to the car cabin has low frequency waves closely to 160 Hz. It is known that the sensitivity of the hearing mechanism highly depends on the frequency content of the received sound and at the middle frequency of 31,5 Hz it is 39,4 dB lower compared to the actual sound pressure level. It means that the passengers inside of the cabin are exposed to essentially higher low frequency sound energy than are the perceptive decibel values and it is confirmed by means of FFT analysis (Fig. 2).

The low-frequencies of the sound and infrasound of different vehicles, building equipments and noise which were analyzed as a whole can be perceived if they are sufficiently intense. In general, it has been found that if people are exposed to very low-frequencies of sound and infrasound, they may experience difficulties in performing mental work, as well as general sense of discomfort. As the intensity increases, dizziness, nervous fatigue, irritation, nausea, and headache may occur [11, 12].

3.2 Frequency and modal analysis

In terms of an internal acoustic environment, it is desirable to know the relationships between the excitation frequency and the revolutions of the motor and the Eigen frequencies of each individual chassis component. Using a modal hammer for modal analysis [14, 15] obtains the Eigen modes (shapes) for the body roof and deck of the vehicle, whose first three and first Eigen (natural) frequencies, respectively, can be seen in Fig. 3.

Comparing the excitation frequency spectra of the vehicles motor with Eigen frequencies of the body roof and deck Eigen modes, we may observe some similarities in excitation frequencies and Eigen frequencies. In this case, such similarity doesn't occur for the defined operational conditions, which is a positive result in terms of generated acoustic energy within the cabin of the vehicle. If the frequencies where to coincide, unwanted resonance would occur, increasing noise in the cabin of the vehicle. From the frequency spectra, where the basic rotational and its harmonic frequencies change with respect to speeds, assuming that the natural frequencies remain constant, it is possible to determine at which frequencies resonance will occur for each vehicle component.

Comparing the excitation frequency spectra of the vehicles motor with Eigen low frequencies of the body roof and deck of the first two Eigen modes 149 Hz and 231 Hz (for doors of the first two Eigen modes are 70 Hz and 139 Hz [4]), coincidence may be found between the excitation speed frequency and its harmonics and Eigen frequencies at approximately 4 000 rpm (Fig. 3). If coinciding frequencies exist in the operational frequencies of the vehicle, it is possible to modify each individual component in order to tune the Eigen frequencies during operation and minimize vibro-accoustic energy within the cabin [4, 5].

The discussed methodology can be applied for different subtle structures which are used not only in automobile industry, but also in nuclear power plants or other strategy plants and household facilities, provided that certain safety precautions are obeyed.

3.3 Energy flow throughout resilient material

A typical resilient material (element), used for vibro-isolation, consists of an elastic layer, as is a rubber for silentblock, between two rigid surfaces. The forces acting at the two sides of the resilient body stand in a firm relationship with the resulting accelerations and/or velocities at the terminal points. This relationship is frequency dependent and is usually expressed in terms of the direct and transfer impedances of each of the endpoints [1].

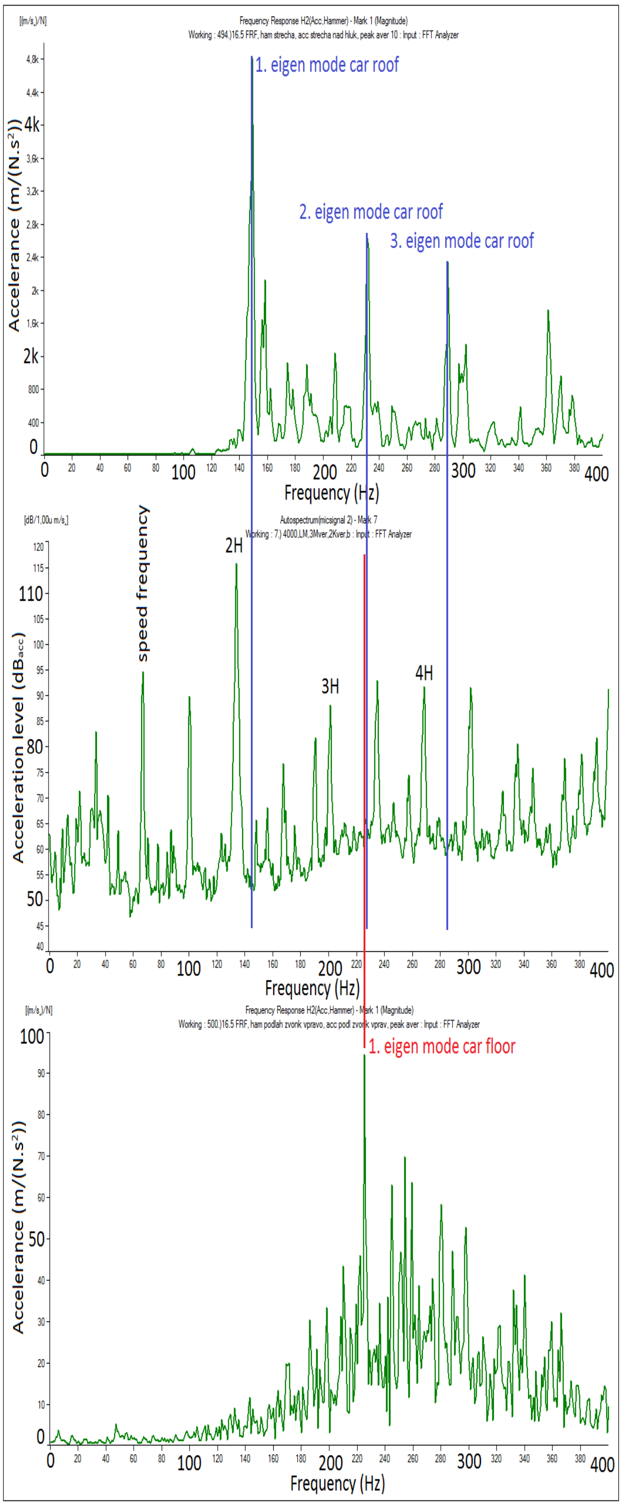

Figure 3. FFT analysis of the car body beyond the silentblock (in the middle) at a given rpm of the combustion engine, and Eigen frequencies of the roof (up) and deck (down) of the same subtle structure as a result of the modal analysis.

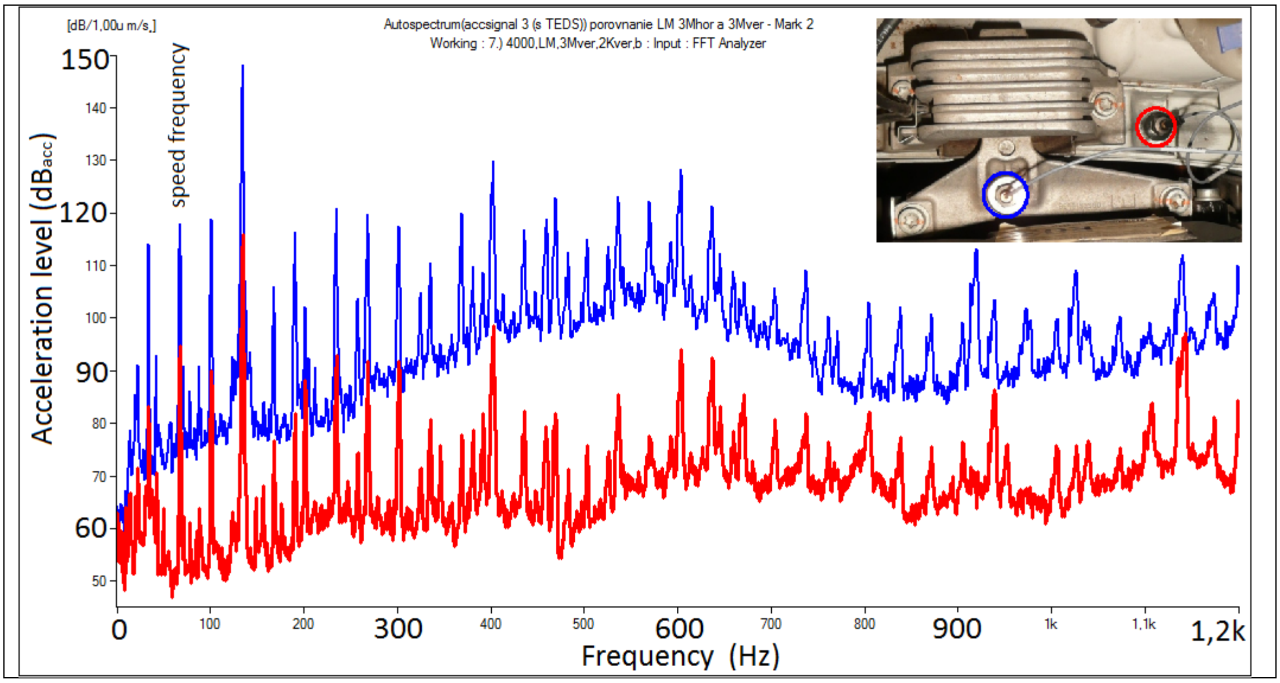

Figure 4. Frequency spectra for silentblock of the bed engine in front of the silentblock (up) and beyond the silentblock (down).

Once the force-velocity relationship has been determinates, either by computation or by measurement, knowledge of the end velocities of the mounted resilient element is sufficient for evaluation of the transmission energy flow throughout the resilient material. Due to inevitable internal losses in the resilient element, the input energy flow must be larger than the output energy flow, as it is confirmed on the engine bed silentblock (Fig. 4). Determining frequency components transmitting throughout the silentblock are speed frequency and its harmonics. The maximal attenuation of the silentblock is in frequency interval from 300 Hz to 800 Hz. From the FFT analysis in front of and beyond resilient element, the attenuation for each dominant frequency can be determined.

On the assumption of pure translational motion in the direction of its axis, the resilient material (e.g. silentblock, see Fig. 1) is characterized by four impedances Z_{11}, Z_{22}, Z_{12}, Z_{21}, where

$$Z_{ij} = \left.\frac{F_i}{v_j}\right|_{v_i=0} \quad \text{where } i, j = 1,2 \tag{8}$$

The mechanical impedances Z_m are complex (see Sec.2.4), and frequency dependent quantities as is shown in Fig. 4 and 5. The impedance can be determined by means of the force measurement [15]. Frequency spectrum of the acceleration measured in front of and beyond the silentblock of the bed engine for the defined frequency interval is shown in Fig. 4.

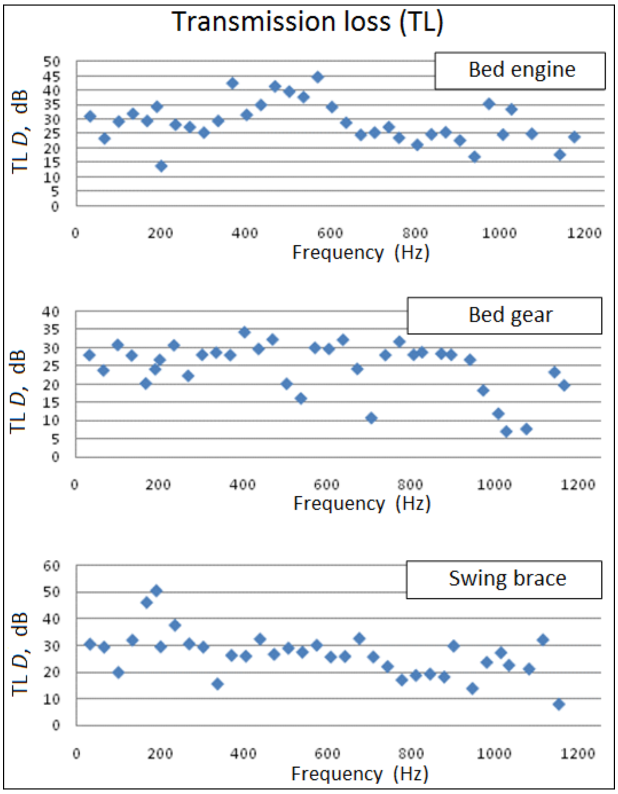

Figure 5. The transmission loss as a function of frequency of the engine mounting.

As each structure and components of mechanical systems, also the resilient material has Eigen (natural) frequencies which are depending on the material properties and characteristic dimension – thickness of the resilient element. The natural frequencies and mode shapes of an element with both ends built in can be calculated by

$$f_n = nc_L/2L \tag{9}$$

where c_L represents the phase speed of the longitudinal wave, L is the thickness of the resilient element and n represents the order of the mode. The thickness of the resilient element according to the frequency spectra is determined using formula (9).

4. Conclusions

The theory and given methods can be used for the calculation of the vibration transmissibility of a resilient material which are used in mechanical and civil engineering. The method is applicable to all materials which behave in a linear way. The methods can be applied for two or more layers forming a sheet. The accuracy (repeatability) of the measured transmissibility is expected to be about 10 %. This will depend on the linearity of the sample, the accuracy of the mechanical impedance value Z_m or other part of body and other measurement parameters as test condition (e.g. humidity, temperature, shape of the sample) and the properties of the test material (e.g. ageing, non-linearity).
The goal of the paper was to carry out the frequency analysis of low frequency vibration and modal analysis for the passenger car which generate low frequency waves that were transmitted to cabin of a car and to analyze the problem so that the problem can be eliminated.
The low frequency waves generated by periodic excitation from rotating machines and/or mechanical systems with reciprocating motion has negative effects on the mechanical system itself and on the human being, as well. The problem of low frequency vibration sources, transmission and their influence on machines, structures and humans is currently of great interest. Especially, their effects on human being which cause damage of physical health and create mental problems when a person is exposed to these frequencies for a longer time as is shown in Refs. [9, 25, 30, 31]. Internal noise has an effect on the comfort of the driver and passenger and affects the safety and reliability of the drive.

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