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TRANSFER OF VIBRATION ENERGY FROM THE WAGON WHEEL TO THE TANK BODY AND POSSIBILITIES OF ITS REDUCTION

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Abstract: *The reduction of railway wagons´ vibration must be taken into account during a wagon design using appropriate vibro-isolation in order to reduce the transmission of structural noise from its main source. The paper analyses the vibration transmission, procedures and vibro-acoustic conditions with a view to reducing this vibration generated between the wagon wheel, as a main source of vibro-sound energy, and the tank body when applying a suitable vibro-isolation. The modal analysis of the tank wagon was complemented by the analysis of gravitational impact of the wheel on the rail generating its vibration. The transmission of the wheel components´ vibration from the wheel through the primary suspension, and the placement of the tank through the sliders and the ball joint into the tank body, is investigated. The solution is also based on the theoretical analysis and methodology of transmission of vibration energy, as well as on the measurement of vibration at the entry and exit measurement points of the main components of the wagon chassis. FFT and modal analysis were used to obtain the vibration parameters. Finally, this article aims at proposing some effective measures that may contribute to reducing the unwanted transmission of structural noise and thus decreasing both the noise emission and immission which have a negative impact on the environment and human health.*

Keywords: Wagon, Vibration, Transmission loss, Auto-spectrum, Isolation.

1. Introduction

Efforts to reduce mechanical vibration and noise are aimed at reducing the acoustic power of the source itself, reducing the radiated sound energy of the source into the open space, and preventing the propagation of energy through the mechanical system, that is, the tank car body itself. Issues of reducing vibration and noise in general have received considerable attention in the developed countries of the world. The activity in this area focuses primarily on the primary reduction of vibration and noise (Musil, 2012), i.e. directly on eliminating the causes of significant vibro-sound energy of the technical components of the rolling stock systems. In this case, it is necessary to concentrate on the transmission of vibration, thus finding ways to reduce the increased dynamic response from the primary source, i.e. the contact of the wagon wheel with the rail, the tank body through the individual main wagon construction blocks such as the primary suspension of the bearing box and the horn-plate, sliders and rotary pan (rotating ball joint) connected to the main crossbar. If the superstructure is a tank, this energy is also transferred to the metal body of the tank, thereby increasing the sound energy emission area to the surrounding environment. The maximum noise emission values shall be achieved at the coincidence the natural (Eigen) frequencies of the tank's Eigen modes with the excitation frequencies. Thus, the individual components of the wagon, from the wheel to the superstructure itself, are characterized by natural frequencies which, when they are in coincidence with the excitation frequencies, cause the component to resonate and thereby increase the emission of noise emitted into the surrounding space (Žiaran et al., 2017). The reduction of vibro-sound energy generated by

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rolling the wheel on the rail, transmitted to the chassis and body structure, and radiated into the space as unwanted noise requires a theoretical, numerical and experimental analysis of the generation, transmission and emission of this energy into the surrounding space (Žiaran et al., 2019).

2. Goals, instrumentation and methodology

The aim of the experimental tests is to determine the Eigen (natural) frequencies of the Eigen modes and the corresponding magnitude of the dynamic loading of the tank wagon. The main aim is to find out the values of vibro-sound energy transmission loss from the source on the wheel to the tank body through the main construction blocks as are primary suspension of the bearing box, the slider and the rotating ball joint. The magnitude of the transmission loss for the monitored construction blocks is determined in the design of suitable vibro-isolation measures. The source of the excitation in the test was the impact hammer and the gravitational impact of the wheel and rail, measuring the time history of the vibration acceleration at the input and output of the investigate blocks (Fig. 1). To identify the energy dominant Eigen frequencies more precisely, the fast Fourier transform (FFT) analysis was carried out using the FFT analyser PULSE. Besides the frequency range, for the signal type, it is also very important to select suitable measuring points (Darula et al., 2011), the appropriate type of averaging, as well and a number of averages per unit time, as well as a suitable time window (Žiaran, 2013). Hanning's window with linear averaging and with 66.67 % overlap gives complete uniform weighting, which is useful for analysing generated signals. The FFT analysis results are presented via square-root of single-side amplitude spectrum. The methodology presented in the article can be also applied for measuring other vibration sources.

Fig. 1: Applying a modal hammer and gravitational impact.

3. Dynamic transfer stiffness of vibration isolators

Dynamic transfer stiffness is the most appropriate parameter to characterize the vibro-acoustic transmission properties of the vibro-isolator for many practical applications. At low frequencies only elastic and damping forces are important; low frequency dynamic stiffness is only weakly dependent on frequency due to the material properties. In principle, the dynamic transfer stiffness of vibro-isolator is mainly dependent on the static preload and its perfect isolation from the source and the receiving structure (Žiaran et al., 2019). In other words, the following theory only applies if there is no flanking transmission between the vibration source and the receiver system. The system consists of three blocks: the wheel-rail, vibro-isolators on the path to the tank body and the receiving components of wagon (tank body) respectively, as is shown in Fig. 2.

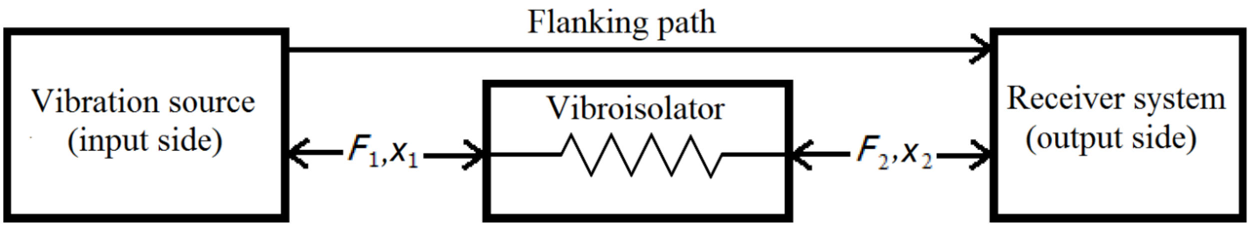

Fig. 2: Simplified block diagram of the vibration source – vibration isolation element – receiving structure and flanking path.

The blocked transfer stiffness is suitable for isolation component characterization in many practical cases. For the presented case, the damping force was not necessary to be considered. Assuming unidirectional vibration of a vibration isolation element (perpendicular to the contact surface), the isolation component equilibrium may be expressed by the following stiffness equations

$$F_1 = k_{1,1}x_1 + k_{1,2}x_2 \text{ and } F_2 = k_{2,1}x_1 + k_{2,2}x_2 \quad (1)$$

where $k_{1,1}$ and $k_{2,2}$ are driving contact stiffnesses occurring when the isolator is blocked at the opposite side (i.e. $x_2 = 0$, $x_1 = 0$, respectively) and $k_{1,2}$ and $k_{2,1}$ are blocked transfer stiffnesses, i.e. they denote the ratio between the force on the blocked side and the displacement on the driven side. For passive isolators $k_{1,2} = k_{2,1}$, because passive linear isolators are reciprocal. The matrix form of Eqs. (1) is

$$\boldsymbol{F} = [k]\boldsymbol{x} \quad (2)$$

with the dynamic stiffness matrices $[k] = \begin{bmatrix} k_{1,1}k_{1,2} \\ k_{2,1}k_{2,2} \end{bmatrix}$

For excitation of the receiving structure via isolator $k_{\mathrm{r}} = F_2/x_2$ (3)

where k_{r}**,** denotes the dynamic driving contact stiffness of the receiver system. From Eqs. (1) and Eq. (3) it follows that

$$F_2 = \frac{k_{2,1}}{1+k_{2,2}/k_{\mathrm{r}}}x_1 \quad (4)$$

Therefore, for a given source of displacement x_1 the force F_2 depends both on the isolator driving contact dynamic stiffness and on the receiver driving contact dynamic stiffness. However, if $|k_{2,2}| \leq 0{,}1|k_{\mathrm{r}}|$, then F_2 approximates the so-called blocking force to within 10 %, i.e.

$$F_2 \approx F_{2,\mathrm{blocking}} = k_{2,1}x_1 \quad (5)$$

Because vibration isolators are only effective between structures of relatively large dynamic stiffness on both sides of the vibration isolation element, Eq. (6) represents the desirable situation at the receiving end; therefore these conditions have to be met when setting up the vibration isolation components (Žiaran et al., 2016).

4. Analysis and evaluation results

The transmission of vibration energy through selected main components (construction blocks) of the chassis of the wagon depends on their design, the material used and the method of application (attachment). The transmission loss in the frequency range up to 3.2 kHz from the modal hammer and gravitational wheel impact sources to the rail, through the main components up to the tank body, was processed separately for the primary suspension of the bearing box and the horn-plate connected to the main crossbeam, the slider mounted on the springs and on the main crossbeam and the rotary pan rigidly connected to the main crossbeam and movably through the ball joint with the tank body (Žiaran et al., research report 2019).

By comparing the frequency-amplitude characteristics, in decibels, for the individual construction blocks, one can see significant differences in the attenuation of the energy transmission of the vibration in dependence on the frequency (Žiaran et al., 2020).

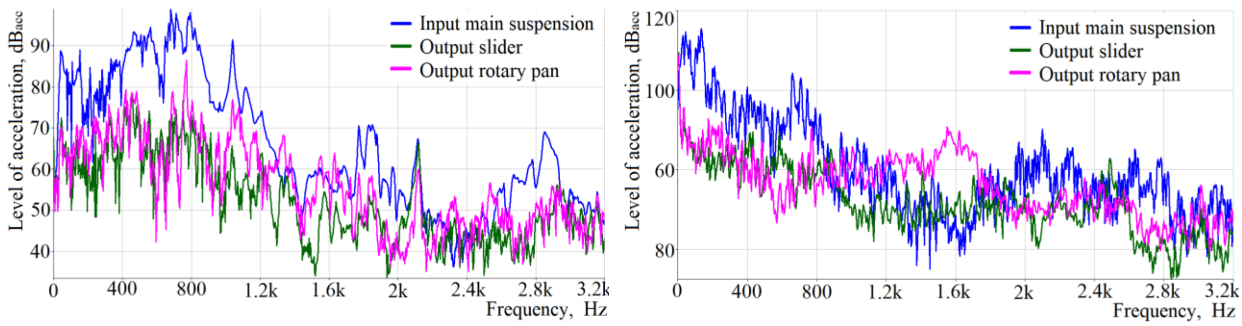

Fig. 3: Frequency distribution of transmission loss by impact hammer (left) and gravity (right).

Transmission loss through all chassis construction blocks, i.e. from the wagon wheel (input) to the tank body (output), was also analysed (Fig. 3). It does not analyse the effect of a braking system attached to the tank body and rigidly connected to the chassis, which forms a vibro-sound bridge and reduces the resulting transmission attenuation. Research in this area is ongoing. From the auto-spectra of Fig. 3, it is evident that the transmission loss is frequency dependent and its value, for a frequency range up to 3.2 kHz, using a modal hammer (17.1 dB) and a gravity impact (16.4 dB) is approximately the same. However, for the frequency range up to 1 kHz, the transmission attenuation is greater (18.5 dB) than the modal hammer (17.6 dB). The reason for the increase in the blocking force is the inertia force at the location of the rotating ball joint vibro-isolator, which is supported both by the aforementioned theory, but also by the elimination of higher braking system frequencies. Results with a greater transmission attenuation difference were also obtained with the analysis of the ball joint block alone (Žiaran et al., 2020). Thus, noise emissions decrease as the weight of the load increases. Measurement of the braking system indicates that this system has a significant effect on reducing transmission loss in a gravitational impact to 3.2 kHz. This is also confirmed by the frequency spectrum at the location of the ball joint, where in the frequency range from 1.2 kHz to 1.8 kHz the transmission attenuation is negative (-10 dB) (Fig. 3-right). To reduce the vibro-sound energy of the tanker, the braking system must also be vibro-isolated.

5. Discussion and conclusions

Noise emission reduction of the tank body can be ensured by the vibro-isolation of the contact surfaces of the tank body with the chassis, i.e. in the places of the primary suspension, rotary pan and slider. The source of excitation of the tank body is also the braking system suspended at the bottom of the tank body and firmly connected to the main crossbar of the chassis. This vibro-acoustic bridge remains open in terms of energy transfer and should be given greater attention in subsequent analyses and measurements. It should be borne in mind that an energy-weak excitation is sufficient to excite in the natural frequencies of the Eigen modes. Sufficient reduction of the noise emission of the tank body can also be ensured by anti-vibration paintwork, which, however, increase the weight of the wagon. The research of other alternatives and possibilities of solution will be based on the evaluation of results from already realized two other experimental measurements and especially measurements during the movement of the prototype tank wagon.

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