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Vibroacoustical analysis of rail vehicle

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Abstract

The article deals with the vibroacoustical analysis of rail vehicle and is the extension of the previously presented work "Modelování podvozku kolejového vozidla s poddajným rámem". The vibroacoustical analysis uses the outcomes of the dynamical analysis of rail vehicle bogie i.e. surface velocities of the bogie frame to compute the acoustic power radiated by the bogie frame and forces acting in the bogie-body interface. The radiated power and the force spectra are then used as the excitation to the rail body model in the environment Auto-SEA to compute the interior acoustic quantities.

Keywords: vibroacoustic, vehicle dynamics, rail vehicle, bogie, SEA, acoustics

1. Introduction

As it was mentioned above this paper represents an extension of the work dedicated to the rail vehicle dynamics [4], especially to the response of the rail vehicle bogie to the kinematical excitation. In this further work the outcomes from the rail vehicle bogie simulation were taken and processed in the way that the acoustical quantities such as acoustic power radiated by the bogie frame and furthermore the force acting in the bogie-body interface were obtained. These outcomes itself constitute a sufficient fundament, which the optimization of the rail bogie vibroacoustics can be based on. Nevertheless since the rail bogie dynamical behaviour is of more important interest therefore changes in the bogie design would probably be very restricted because of this point of view, the rail body model was created and the vibroacoustic response inside the rail body was computed, so the successive vibroacoustic optimization could be applied directly on the rail body. See Fig. 2 for the general idea of the whole modelling process.

1.1. Short review of the investigated system

The investigated system consists of three subsystems (see Fig. 1.). Each one was modelled using different method. Subsystem 1 represents the bogie drive and was modelled as a Multi-Body system with 50 degrees of freedom. Subsystem 2, the bogie frame, was modelled using Finite Element Method and was connected with the bogie drive by means of the modal synthesis method. Subsystem 3 corresponds to the body of the vehicle and Statistical Energy Analysis was used to model the airborne and structureborne noise propagation through its structure into the passengers' compartment.

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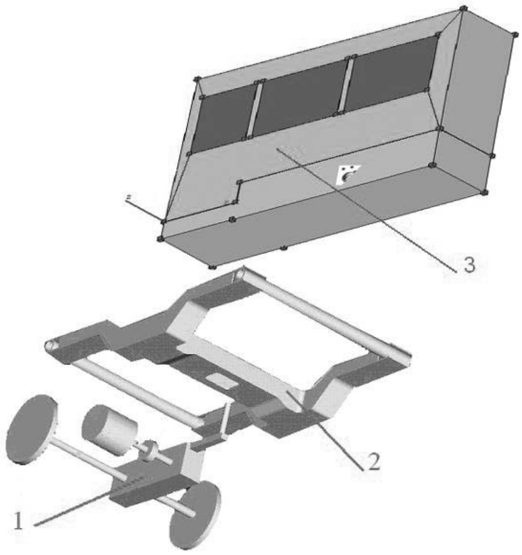

Fig. 1. Layout of the investigated system.

2. Acoustics

The calculation of the radiated acoustic power is a very complicated problem, but it is an advantage that in practice a large number of vibrating structures can be modelled as flat plates. As it was mentioned above the radiated power is an important value. The acoustic power calculation can be done in many ways. The following formula [2] is widely used in current engineering practice

$$W = \sigma \rho c a b \left\langle \overline{v^2} \right\rangle , \tag{1}$$

where ρ is the air density, c is speed of sound a,b are the plate dimensions, $\left\langle \overline{v^2} \right\rangle$ the spatially-averaged mean square velocity and finally σ is the radiation efficiency.

The use of the radiation efficiency makes the acoustic power calculation relatively easy, because it is not necessary to calculate the acoustic pressure and than the acoustic intensity and afterwards to integrate it over the space to get the acoustic power value. The radiation efficiency is than useful, but also very "dangerous" quantity and therefore we should be very cautious when using it. The outcomes can vary in the range of 10^3 if improperly used.

In the analysis the formula (1) and the "classical" approach using Reyligh's integral and integration of the acoustic intensity over hemisphere [3] was used to compute the acoustic power radiated by the bogie frame. To obtain the radiation efficiency of the bogie frame there were used three approaches. Two empirical sets of formulas given by Miadanik and Leppington [5] and so called average modal radiation efficiency broadly used in the Statistical Energy Analysis (SEA) [1]. The latter is computed in the way that radiation of all mode shapes appearing in the given frequency range are taken into account and by averaging them the single frequency dependent variable, average modal radiation efficiency, is computed.

Finally four outcomes of the radiated acoustic power (three on the basis of the formula (1) and fourth based on the "classical" approach) are computed for comparison.

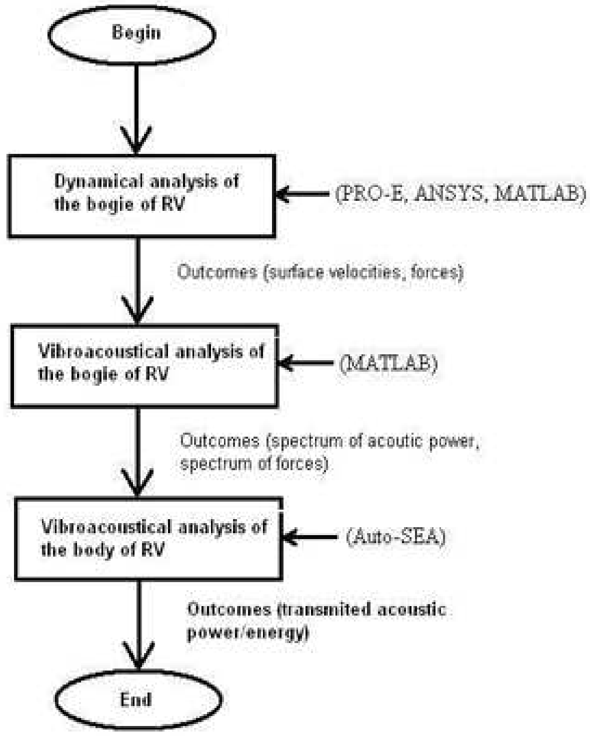

Fig. 2. Layout of the complex analysis.

2. Bogie frame vibroacoustics

To acquire the velocity field of the bogie frame surface and furthermore the time dependent relation of the displacements of the nodes situated at the places of the secondary suspension, which will enable to calculate the force excitation of the body floor, was the first step to the vibroacoustical analysis of the rail vehicle (RV) bogie. The velocity field was used to calculate the acoustic power radiated by the bogie frame surface and the displacements of the nodes to calculate forces acting on the body floor.

2.1. Vibroacoustical analysis at steady harmonic response

The dynamic analysis at the steady harmonic excitation was used to acquire the acoustic power spectrum radiated by the bogie frame, which was needed for the excitation of the RV body in the AutoSEA environment.

The dynamic model was excited by the kinematic displacement of 1.5 mm from 5Hz to 3000 Hz with the step of 5 Hz.

The vector of the velocities in the vertical direction of the frame surface nodes was rearranged into the matrix, which represents the surface velocity field of the bogie frame. This field was furthermore used for the acoustic power radiation calculation. Both approaches for the acoustic power calculation mentioned in the previous chapter, i.e. formula (1) and the acoustic intensity integration over the hemisphere were used. The average modal radiation coefficient and furthermore other two conventional approaches published by Maidanik and Leppington was used in the formula (1).

Fig. 3 shows the average radiation efficiency and other two approaches to the radiation efficiency calculation given by Maidanik and Leppington.

All three approaches quite correspond. Only the calculation according to Leppington differs from the others, but only to the frequency of 100 Hz, which is in practice a negligible error.

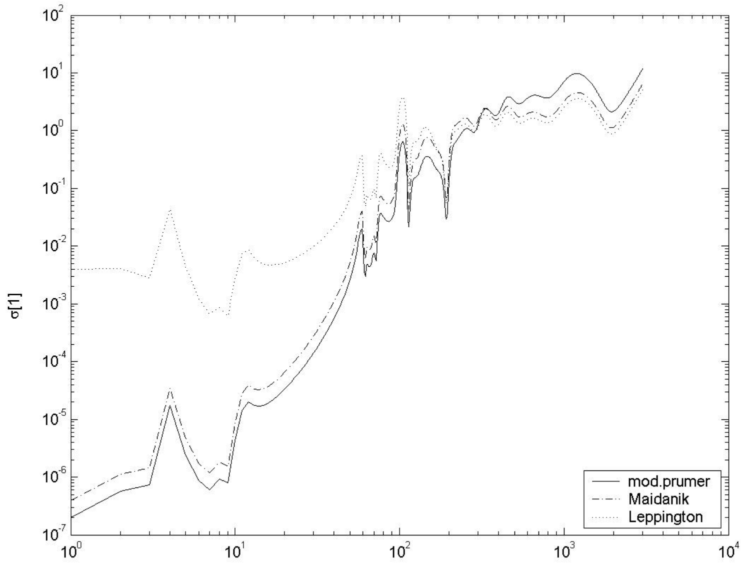

Fig. 3. Radiation coefficients as function of frequency.

Fig. 4 represents the acoustic power as a function of frequency. The values of the acoustic power were similarly to the radiation coefficient calculation calculated using the average radiation efficiency („modprumer"), conventional formulas given by Maidanik („Maidanik"), Leppington („Leppington") and furthermore using the acoustic intensity integration over the hemisphere („Wfar"). The last mentioned approach was assumed to be the most credible, because the average radiation efficiency and the conventional approaches include certain simplifications and averaging. Satisfying coincidence is disturbed only to the frequency of 100 Hz, which is negligible.

2.2. Vibroacoustical analysis at excitation by track irregularities and obstacle crossing

The dynamic analysis at excitation by track irregularities and obstacle crossing gave the frame surface velocity field. In such a response there is a lot of frequencies. From above it is clear that structures don't radiate sound similarly at each frequency. It was therefore necessary to decompose the multi-frequency response and calculate the radiated acoustic power at each frequency. The overall acoustic power was a simple superposition of these partial acoustic values.

Because the decomposition to all frequency components and successive acoustic power calculation was too much time consuming the simple technique, which replaces the original signal by the altered signal consisting of the frequency components, which mainly contribute to the acoustic power radiation, was developed.

Regretfully there is no room to present this technique, therefore this chapter confine itself only in presenting the outcomes from this analysis.

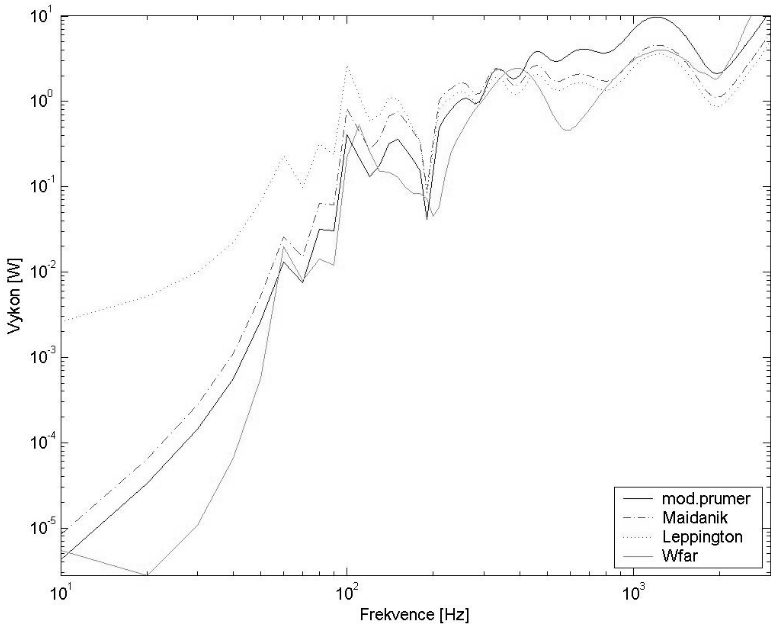

Fig. 4. Acoustic power as a function of frequency.

By the application of the “altered signal technique” on the dynamic response calculated for the case of the track irregularities excitation and obstacle crossing and its processing by the vibroacoustical analysis we obtain a single value of the acoustic power, which is the estimate of the acoustic radiation occurring during the ride on the real track.

Tab. 1 gives the estimate of the acoustic power radiation of the RV frame in the case of the rail junction crossing at 80 km/h.

The acoustic power was again obtained by using several approaches: the integration of the acoustic intensity, the average radiation efficiency and the conventional formulas given by Maidanik and Leppington. The outcomes of the Maidanik and Leppington vary from the outcomes calculated using the average rad. efficiency and the integration of the acoustic intensity, which is assumed to be more accurate.

Approach	*Acoustic power [W]*
Rad. efficiency	2,1
Integration of ac. int.	1,9
Maidanik	1,15
Leppington	0.9

Tab. 1. Radiated acoustic power at rail junction crossing.

3. Body of RV

The vibroacoustical model of the RV body was created in the AutoSEA environment. The AutoSEA is based on the Statistical Energy Analysis, which is currently one of the most effective tools for the vibroacoustical analyses in practice. Its advantages can be used in the modelling of the systems characterized by the large number of connected structures, which also contains acoustic cavities. Simplicity of the energetic equations with relatively low number of unknowns (125 in the disputed case) enables to solve these equations very quickly. Despite the fast building and solution of a model it is necessary to devote longer time to the model adjusting.

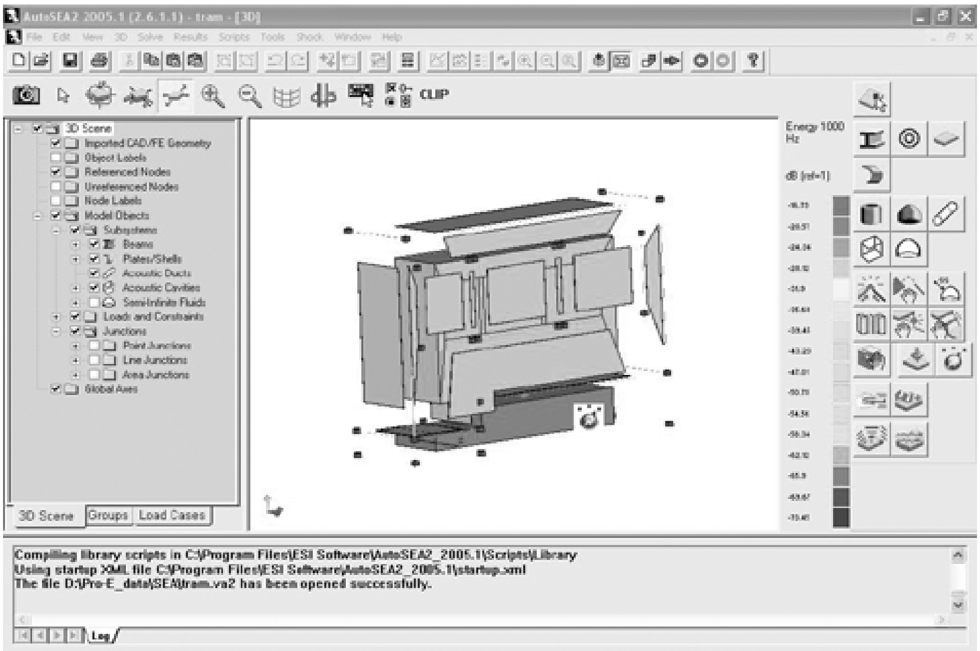

Fig. 5. AutoSEA model overwiev.

Fig. 5 shows the model of the RV body in the AutoSEA environment. Regarding the symmetry only the quarter of the body was modelled. The interior cavity, which creates the passengers' space, is of interest. Thanks to the SEA principle the details of the interior arrangement or the places where the acoustic quantities are taken are not important. The acoustic power in the passengers' space is presented on the Fig. 6. The excitation from the bogie frame acoustic radiation was firstly taken, then the excitation from the floor vibrations and finally both together. Fig. 6 shows that the structural excitation of the floor is dominant mainly in the lower frequency range. Fig. 7 compares the input acoustic power in the bogie space to the power transmitted into the passengers' interior. It is apparent that the ability of the structure to transfer acoustic energy decreases with increasing frequency. It should be also noted, that SEA outcomes are not valid at low frequencies, where only a few resonant modes of the subsystems appear. In this case the reliability of the outcomes starts at the frequencies 150-200 Hz (due to the large subsystems) contrary to the common lower limit of 300 Hz in the automotive industry.

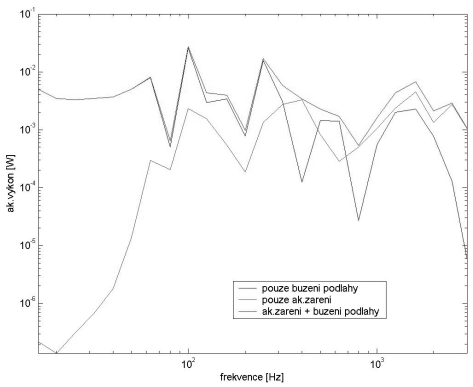

Fig. 6. Acoustic power in passengers' space.

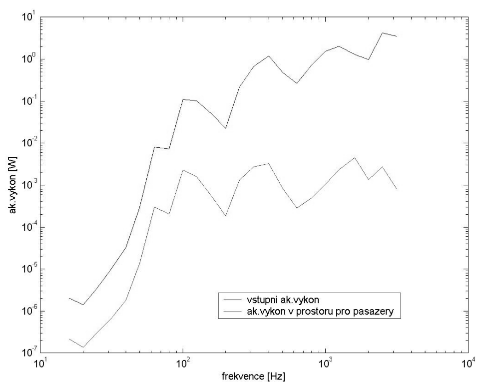

Fig. 7. Comparison of input and transferred acoustic power.

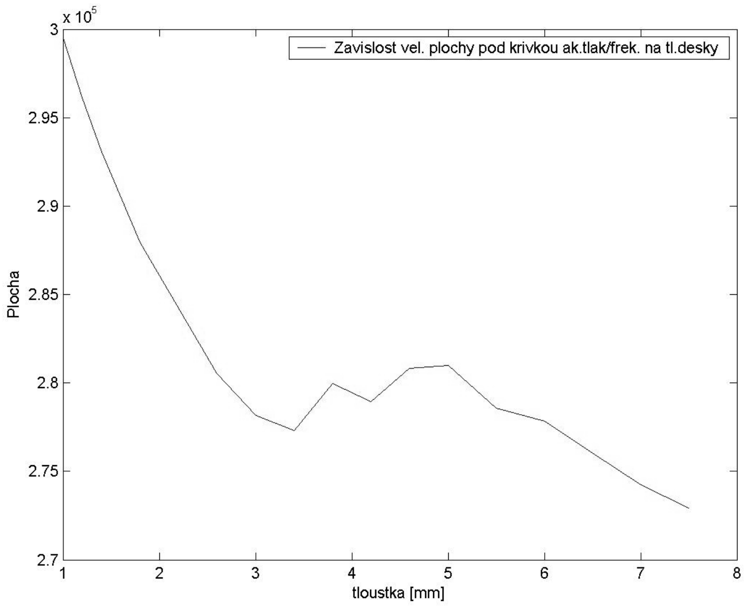

Fig. 8. Relation between the floor thickness and the acoustic power in RV body interior.

4. Conclusion

The presented work deals with the vibroacoustical analysis of rail vehicle. The complex mathematical model of the RV bogie including the flexible bogie frame, which was connected to the bogie drive using the Modal Synthesis Method, was built up. The bogie drives were created as discrete systems with 50 degrees of freedom. The modal analysis of this model was performed and furthermore the dynamical analysis, where responses to the following kinematical excitations were calculated:

-harmonic excitation

-excitation by the pulse simulating rail junction crossing

-excitation by the track irregularities

Because the vibroacoustical response was the main goal of the work, the most important outcomes of the dynamical analysis were: the velocity field of the frame surface and the time dependent forces acting in the locality of the secondary springs. The velocity field of the bogie frame was used to calculate the radiated acoustic power of the frame surface. This calculation was performed in four ways. The displacements of the nodes in the location of the secondary springs were used for the calculation of the forces acting on the body floor.

The vibroacoustical response of the bogie (the radiated acoustic power and the forces acting on the body floor) was used as the excitation in the RV body analysis. The vibroacoustic model of the vehicle body was created using the AutoSEA, which is based on the Statistical Energy Analysis. The SEA approach was used because it appears to be the most appropriate way for the vehicle body vibroacoustical analysis during the development stage. The acoustic pressure or the acoustic power transmitted into the RV body interior were the outcomes of the RV body analysis in AutoSEA. The sensitivity analysis to the variation of certain parameters defining the floor subsystem, which is the main path of the acoustic energy transmission, was also performed.

Furthermore the new technique for the calculation of the acoustic power radiated during the ride on track or during the crossing over the rails junction was developed. This technique includes some idealisations and simplifications and therefore the outcomes represent the estimate of the acoustic power radiation. These outcomes can be fully used for the successive optimization or for the assessment and comparison of the different types of bogies.

The whole methodology of the vibroacoustical model development can be divided into the following steps:

- development of the complex dynamic model of RV bogie using the Modal Synthesis Method
- dynamic analysis of the RV bogie in order to obtain the bogie frame velocity field and the forces acting on the RV body floor
- vibroacoustical analysis of the RV bogie
- vibroacoustical analysis of the RV body

The most important contribution of this work to the RV mechanics is the possibility to asses the vibroacoustical behaviour of the rail vehicle bogie and its propagation into the interior without the need of the excitation parameters measuring and the direct calculation of body floor vibrations. For the vibroacoustical analysis only the physical parameters of the RV bogie, body and the interface between them are needed. To asses the vibroacoustical response to the special excitations such as rail irregularities and rail junction crossing the track and the junction characteristics are necessary.

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