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CALCULATION OF BANDWIDTH OF A TRANSDUCER FOR LATERAL FORCE ACTING IN WHEELSETS OF RAILWAY VEHICLES

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Abstract: *The aim of this paper is to estimate the bandwidth of the KUB transducer for measurement of the lateral forces that act upon the wheelset of rail vehicle. The solid 3D model of the transducer's body has been created and the natural frequencies of the transducer's body have been calculated by FEM analysis. The results obtained from modal analysis that enable estimation of the bandwidth of the KUB transducer are compared to the UIC recommendations in order to establish applicability of KUB transducer for the application in investigation of railway vehicles.*
Key words: *transducer, bandwidth, wheelset, rail vehicle*

INTRODUCTION

Bandwidth of a transducer is the main descriptor of its dynamic properties. It represents the frequency range in which amplitude-frequency characteristic of the transducer is constant, and phase-frequency characteristic of the transducer is linear. If frequency spectrum of an input signal belongs to the bandwidth of a transducer, then the corresponding output signal is proportional to the input signal, i.e. a transducer does not distort input signals with frequency spectra within the bandwidth of the transducer. Therefore, the bandwidth of a transducer determines input signals that may be processed by application of the transducer.

International regulations [1] for investigations of safety and comfort of railway vehicles demand that measurements of mechanical quantities during test rides should be performed with frequency of acquisition that has to be higher than 200 Hz. Besides mechanical quantities that may be measured by commercial measurement transducers (acceleration), these regulations request measurements of mechanical quantities that are to be measured by specialized devices, like it is the case with the lateral force which acts on wheelsets of investigated railway vehicle (the force will be denoted as H-force in the text that follows).

This paper presents estimation of the bandwidth of a transducer for measurement of H-force by numerical methods. The goal of the investigation is to determine if the transducer satisfies requirements for measurements of the horizontal forces that act on the wheelset of the rail vehicle.

DESCRIPTION OF THE TRANSDUCER

As a consequence of forces acting in the wheel/rail contact, the wheelset is exposed to lateral and longitudinal forces [2].

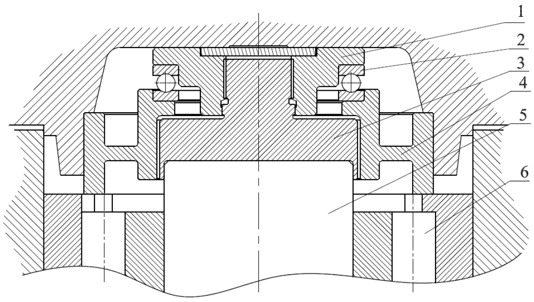

Figure 1: The force transducer KUB
1- screw nut, 2-axial bearing, 3-screw,
4-transducer's body, 5-axle, 6- bearing, 7-cover plate

Measurement of the lateral force, which acts on the wheelset of a rail vehicle, is performed by special transducer, named by its constructors KUB, shown in Figure 1. The transducer is mounted into the axle box, where the axial force, acting upon the wheelset, is transmitted from axle (5) to the transducer over screw nut (1) and axial bearing (2) to the transducer's body (4), as it is shown in Figure 1. Measurement of the force is based on the measurement of the strains of the transducer's body (4).

THEORY

There are several practical methods for calculation of bandwidth of transducers. The methods are divided to experimental and theoretical methods, and the latter are further divided into analytical and numerical methods.

Experimental methods [3] are based on measurements of the response of the transducer to the excitation signal with known and well-defined properties. The measured response is then used for calculation of amplitude-frequency and phase-frequency characteristics.

Theoretical methods [1, 4] for calculation of bandwidth of transducers are based on theory of systems.

The concept of analytical methods [4] is to determine analytical form of expression for the transfer function of the system $G(s)$ (where s represents complex frequency $s = j\omega$) based on differential equations that describe relations between input and output of the system, and then, to apply analytical methods to determine boundaries of the bandwidth of the transducer. The basic theoretical cases of transducers from the point of view of transfer functions are the first-order systems and the second-order systems. The first order systems are the systems that accumulate one kind of energy during measurement process (typical examples are thermometers that during measurements accumulate heat), and their transfer functions have form:

$$(1) \quad G(s) = G(0)\frac{1}{1+\tau s}$$

where τ represents response time of the system. The bandwidth of first-order systems is estimated to be between 0 and $1/10\tau$. Systems of the second order are the systems that store two kinds of energy during measurement process (typical examples are elastic dynamometers that store kinetic energy and potential energy of deformation during measurements), and their transfer functions have form:

$$(2) \quad G(s) = G(0)\frac{1}{1+2\xi\left(\frac{s}{\omega_0}\right)+\left(\frac{s}{\omega_0}\right)^2}$$

where ω_0 represents resonant frequency of the system and ξ represents damping factor of the system. The bandwidth of a second-order system may be estimated to be between 0 and $\omega_0/10$.

Numerical methods for calculation of bandwidth of a transducer [5-7] have the same theoretical basis as analytical methods. However, numerical methods rarely lead to calculation of transfer function of the system.

One of numerical methods for estimation of the bandwidth is calculation of natural frequencies of the transducer. Theory of systems shows that transfer function of any linear system may be decomposed into a sum of transfer characteristics of sub-systems of the first and the second order. Hence, a frequency characteristic of a system is constant in frequency range where frequency characteristics of all its sub-systems are constant. For elastic transducers, it means that bandwidth of the system is between zero and one-tenth of the lowest natural frequency that may be excited by input signal. Therefore, once all natural vibrations and corresponding frequencies are determined by procedure that is called modal analysis, it is easy to estimate the bandwidth of the transducer.

The considered transducer KUB is elastic and continual structure. With a purpose to perform modal analysis, commercial FEM software may be used. The concept of analysis is based on solving of dynamic equation of motion that describes free undamped vibrations. In the case of free vibrations, the externally applied loads in the equation are set to be zero, and the vibrations of the structure are determined by initial conditions. Solution of the free vibration problem with zero damping provides the most important dynamic properties of the structure, the natural frequencies and the respective mode shapes. Each mode shape represents a particular manner of motion of discrete elements obtained by FEM discretization of the structure. The number of vibration modes of a structure is infinite, and in FEM analysis, it is limited by number of degrees-of-freedom (DOF) of the model. However, existence of a certain vibration mode in a response of the considered structure depends on place, direction and frequency of excitation [8]. A certain vibration mode will be incited by excitation if:

- excitation force acts at place and in the directions where the modal shape largest deformations, and
- excitation frequency is close to natural frequency of the vibration mode.

In a real structure, the excitation of a certain vibration mode depends also on the damping within the structure.

The equation that governs the modal analysis of a finite element model is given by

(3) $$[m]\cdot\{\ddot{x}\}+[c]\cdot\{x\}=\{0\}$$

with the following notation: $[m]$-mass matrix, $[c]$-stiffness matrix, while $\{x\}$ and $\{\ddot{x}\}$ are vectors of nodal displacements and accelerations, respectively.

For an N-degree-of-freedom system, a possible solution can be assumed in the form:

(4) $$\{x\}_i=\{\phi\}_i\cdot\sin\left(\omega_i t-\alpha_i\right)$$

where: $\{\phi\}_i$ is the i-th modal shape or eigenmode with a corresponding natural circular frequency ω_i and phase angle α_i.

Substituting equation (4) in equation (3) and eliminating $\sin(\omega_i t\text{-}\alpha_i)$ one obtains matrix equation:

(5) $$\left([c]-\omega_i^2[m]\right)\cdot\{\phi\}_i=\{0\}$$

Equation (5) can be explicitly written as the system of N equations, given by equation (6):

(6) $$\begin{bmatrix} c_{11}-\omega^2 m_1 & c_{12} & \cdots & c_{1N} \\ c_{21} & c_{22}-\omega^2 m_2 & \cdots & c_{2N} \\ \vdots & \vdots & \cdots & \vdots \\ c_{N1} & c_{N2} & \cdots & c_{NN}-\omega^2 m_N \end{bmatrix}\cdot\begin{Bmatrix}\phi_1\\ \phi_2\\ \vdots\\ \phi_N\end{Bmatrix}=\begin{Bmatrix}0\\ 0\\ \vdots\\ 0\end{Bmatrix}$$

The objective of modal analysis is to calculate the values of ω_i and the corresponding vectors $\{\phi\}_i$ that satisfy equation (5). A non-trivial solution, i.e., a solution for which all ω_i and $\{\phi\}_i$ are non-zero, requires that the determinant of matrix in equation (7) is zero.

(7) $$\det\left([c]-\omega_i^2[m]\right)=0$$

Polynomial equation (7) is known as the characteristic equation of the system. Solutions ω_i (natural circular frequencies) of the characteristic equation are called eigenvalues. For each of the solutions ω_i (i=1,2,...,N) of the characteristic equation it is possible to solve equation (6) to determine $\{\phi\}_i$, which is called eigenvector. One eigenvector $\{\phi\}_i$ corresponds to each of the eigenvalues, that is, for a system with N-degrees-of-freedom there are N eigenvectors (each corresponding to a certain mode shape), and their corresponding eigenvalues. Trivial situation arises when this vector is zero, corresponding to the motion of the considered structure as a rigid body.

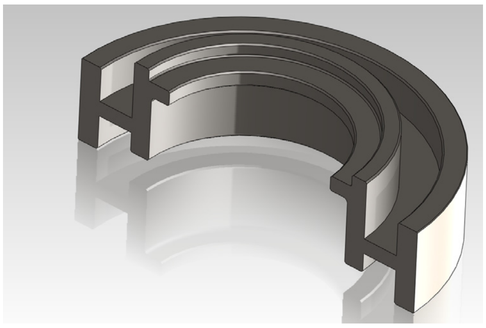

Fig. 2: Solid model of the transducer's body

ANALYSIS

The solid model of transducer body has been created in software package Solid Works, and the calculation of the natural frequencies of transducer's body has been performed using the software package Ansys.

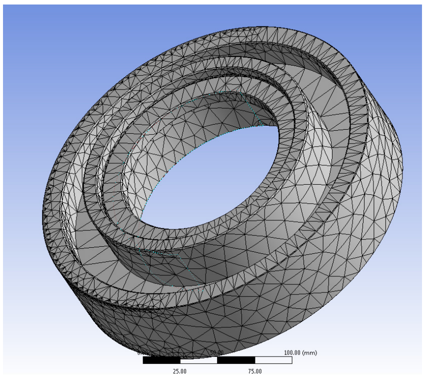

Fig. 3: Model of the transducer's body divided into mesh

The model of the transducer's body has been divided into linear 3D finite elements with tetrahedral shape with 12 DOF (3 DOF per tetrahedral node). Discretization parameters are shown in Table 1, and they were selected according to the request that only natural frequencies lower than 5 kHz were calculated.

Elements	18723
Nodes	34660
DOF	103980
Length of tetrahedral edge	0,707 mm

Table 1: Discretization parameters of the FEM model of the transducer's body

The transducer's body was modelled as being made from structural steel.

The boundary conditions strongly influence natural frequencies and modal shapes. Therefore, it is very important to perform FEM calculations of natural vibrations with boundary conditions that are describing the real contacts between bodies as close as it is possible.

The cover plate, marked with (7) in Fig.1, is fixed with screw nuts to the axle box and is pressing the transducer's body from the top side. Therefore, at the top side of the transducer, denoted as "Surface 1" in the Fig. 4, boundary conditions are adopted so that displacements in x, y and z direction are equal to 0.

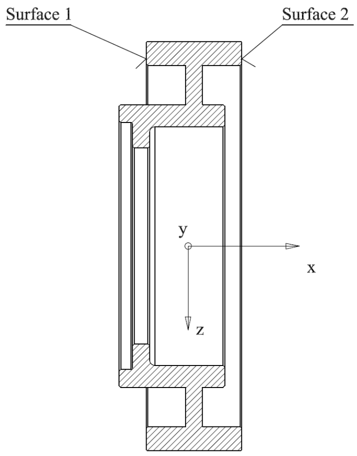

Fig.4: Cross-section of the transducer's body

Bottom side of the transducer's body is supported by the axial bearing. Therefore, at the bottom side of the outer ring of transducer's body, donoted as “Surface 2” in Fig. 4, are adopted boundary conditions so that longitudinal diplacements (along x axes) are equal to zero and the displacements of the surface in lateral directions (in yz plane) are not limited.

RESULTS

For the described model and the boundary conditions, the lowest natural frequencies are: f_{n1}=4248.5 Hz, f_{n2}=4557.2 Hz, f_{n3}=4579.6 Hz, f_{n4}=4859.6 Hz and f_{n5}=4860.6 Hz. The modal shapes of the three vibration modes with lowest natural frequencies are shown in Figures 5-7.

The upper limit of the bandwidth of the transducer may be estimated as:

$$(4)\ \Delta f = \frac{\omega_1}{10} = \frac{4248.5}{10} \approx 425 \text{ Hz},$$

and the bandwidth of the transducer may be estimated to be:

$$(5)\quad BW = 0\text{-}400 \text{ Hz}$$

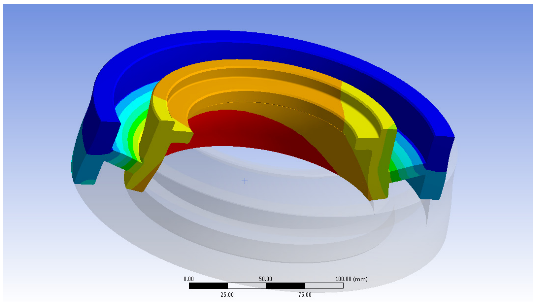

Fig. 5: The first modal shape with frequency f_{n1} = 4248 Hz

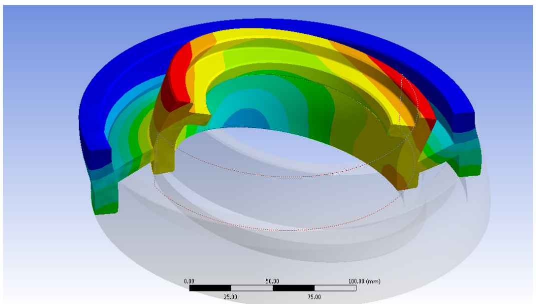

Fig.6: The second modal shape with frequency f_{n2} = 4557 Hz

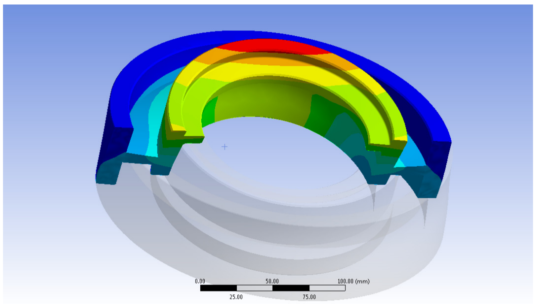

Fig. 7: The third modal shape with frequency f_{n3} = 4579 Hz

CONCLUSION

Based on the natural frequencies of its body was estimated the bandwidth of the KUB transducer for measurements of H-forces. The obtained results have shown that the transducer is suitable for measurements of the forces with frequencies up to 400 Hz. For the signals with higher frequencies is expected distortion.

Future work should be focused on the experimental investigations of the response spectra of the transducer by modal testing. Considering that the transducer is exposed to the dynamic excitations, the transient analysis should be performed in order to estimate dynamic characteristics and transfer function of the transducer.

REFERENCES

[1] *UIC 518: Testing and approval of railway vehicles from the point of their dynamic behaviour-Safety-Track fatigue-Ride quality*, International Union of Railways, 3rd edition, October 2005.

[2] D. Atmadzhova**,** *Influence of wheel and rail profiles on railway vehicle dynamics,* First International Conference on Road and Rail Infrastructure CETRA 2010, 17-18 May, Opatija, Croatia, 2010

[3] D.J. Ewins, *Modal Testing-theory, practice and application*, Research Studies Press, Baldock, Hertfordshire (England), 2000

[4] N.M.M. Maia and J.M.M. Silva, *Theoretical and Experimental Modal Analysis*, Research Studies Press, Baldock, Hertfordshire (England), 1998

[5] R.D. Cook, D.S. Malkus, M.E. Plesha and R.J. Witt, *Concepts and Applications of Finite Element Analysis*, John Wiley and sons, New York (USA), 2002

[6] Ying-Jun Li, Jun Zhang, Zhen-Yuan Jia, Min Qian, Hanguang Li, "*Research on force-sensing element's spatial arrangement of piezoelectric six-component force/torque sensor*", Mechanical Systems and Signal Processing, Volume 23, Issue 8, pp. 2687-2698, 2009

[7] A. Abdullah, A. Pa, *Correct Prediction of the Vibration Behaviour of a High Power Ultrasonic Transducer by FEM Simulation*, Int. J. Adv. Manuf. Technol., Vol. 39, pp. 21-28, 2008

[8] S. Ćirić Kostić, M. Ognjanović, *Excitation of Modal Vibration in Gear Housing Walls*, FME Transactions, Vol. 34, No.1, pp. 21-28, 2006.

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ИЗЧИСЛЯВАНЕ НА ЧЕСТОТНИЯТ ДИАПАЗОН НА ПРЕОБРАЗОВАТЕЛ ЗА РЕГИСТРИРАНЕ НА СТРАНИЧНА СИЛА ДЕЙСТВАЩА НА КОЛООСИ ОТ ЖЕЛЕЗОПЪТНИ ВОЗИЛА

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СЪРБИЯ

Ключови думи: *преобразовател, широчина на честотната лента, колоос, железопътно возило.*

Резюме: *Целта на настоящата статия е да се определи честотния диапазон на KUB преобразувател за измерване на странични сили, които действат върху колоос на железопътно возило. Създаден е SOLID 3D модел на тялото на преобразователя и чрез FEM анализ са изчислени реалните честоти на тялото на преобразователя. Сравняват се резултатите, получени от модалния анализ, които позволяват оценка на широчина на честотният диапазон на KUB преобразовател с препоръките на UIC, с цел да се създаде приложимост на KUB преобразувателя за прилагане в изследване на железопътни превозни средства.*