EAN 2013
51th conference on experimental stress analysis

USING SPECTRAL MEASUREMENTS AND FRF IN UNDERSTANDING THE DYNAMIC PROPERTIES OF MACHINES

Vladimir Dekys[1, a], Alzbeta Sapietova[1, b] and O. Stevka[1, c]

[1]Department of Applied Mechanics,
Faculty of Mechanical Engineering, University of Zilina, Slovakia

[a]e-mail: vladimir.dekys@fstroj.uniza.sk, [b]e-mail: alzbeta.sapietova@fstroj.uniza.sk,
[c]e-mail: ondrej.stevka@fstroj.uniza.sk

Keywords: frequency response function, spectrum, rotor bar pass frequency, condition monitoring, EMA

Abstracts: The paper deals with understanding vibration source, transfer path and vibration of cover sheets by using tools of condition monitoring an experimental modal analysis, namely spectral analysis and frequency response functions. The measurements were selected specifically to an electromotor, a rotor, a shock absorber and the cover sheets and outputs measured were interpreted with regard to achieving the goal of reducing noise and vibration of machines. In this paper is presented using FRF functions as a useful tool for the interpretation of measured data.

1. Introduction

A problem of decreasing sound and vibration of the machine is publicly acceptable. The solution to this problem may be the goal of our global strategy for research, development and production. However, due to the specific tasks to be solved it can be difficult and complicated. These complications may be due to the absence of relationship or mathematical formulations between measurable sound and vibration parameters (for example RMS value of velocity vibration or acoustic pressure) and material, boundary and geometric conditions and parameters of machine. Therefore, the general task was formulated using partial tasks. Then the sub-tasks are addressed: reduction of potential sources of sound and vibration, modification of transfer paths and finally vibration and noise reduction of excitation parts.

This problem can be solved as a vibration based condition monitoring problem. The symptoms of many fault conditions are described in the publications [1-4].

The paper deals with interpretation of vibration measure on the real equipment with rotate parts in this paper. Test equipment is simplified to a driving electromotor and driven rotor (drum) with unbalance. The rotor is driven by a belt. Selected results of the measurements for the unit are presented.

2. The interpretation of the measurements

Condition monitoring can detect not only mechanical problems by using accelerometers, but also electrical problems using vibration signature analysis. The magnetic fluid creates flux which induces electromagnetic forces within a motor. The mechanical forces caused by imbalances, misalignment, resonance, for example, and the electromagnetic forces are captured in the bearings. These forces are then measured directly by force transducers placed on bearings housing, or indirectly by vibration transducers such as accelerometer, etc. [1].

Some of the problems which can be detected using vibration analysis are [2]:

- stator eccentricity, shorted laminations and loose iron,
- eccentric rotor (variable rotor gap),
- rotor problems (broken or cracked rotor bars or shorting rings, shorted rotor laminations, loose rotor bars, etc.,
- thermal bow induced by uneven localized heating of rotor,
- electric phasing problem due to loose or broken connectors,
- problems with synchronous motors,
- problems with DC motors,

- torque pulse problems.

  Rotor problems are detected by f_{RBPF}, rotor bar pass frequency:

$$f_{RBPF} = n_{RB} \cdot f_{RPM} , \qquad (1)$$

n_{RB} - number of rotor bars (or rotor slots with winding, and f_{RPM} – frequency of rotor.

The objects of our interest were the machines consist of an electric motor, drum and transfer belt. The two machines with different motors that denoted A and B were monitored at different speed of electric motor (RPM_e) in the range from 140 Hz to 220Hz. Both motors had 18 rotor bars. The high peaks in the acceleration spectrums were detected in the frequency ranges above 2000Hz. It was necessary to determine the cause of these spectral peaks.We assume that the cause of these peaks are the rotors problems or resonance. The measurements were taken at the selected running speeds of the electric motors to confirm or reject this assumption, when the accelerometers were placed at the bearings of motors.

In Figs. 1 - 2 are acceleration spectrums for motor A. In Fig. 1 are the harmonic frequencies of 146 Hz (RPM_e) and in Fig. 2 are harmonic frequencies of 199 Hz (RPM_e). If there is a problem of the rotor bars, then the increasing of RPM implies a shift of the peaks in the analyzed frequency range. The peaks around 2750 Hz are not moved significantly when changing engine speed and they expressed resonance. Because the distance side bands around the resonance peak corresponding revolutions of the drum (RPM_d), the drum is exciter of resonance. The high peak at 14xRPM_e is present around 2750Hz. It is no rotor problem, but resonance problem around at 14xRPM. The motor drives a drum in machine. Drum RPM (RPM_d) is sideband around harmonic of motor RPM. We assume the resonance is caused by the drum.

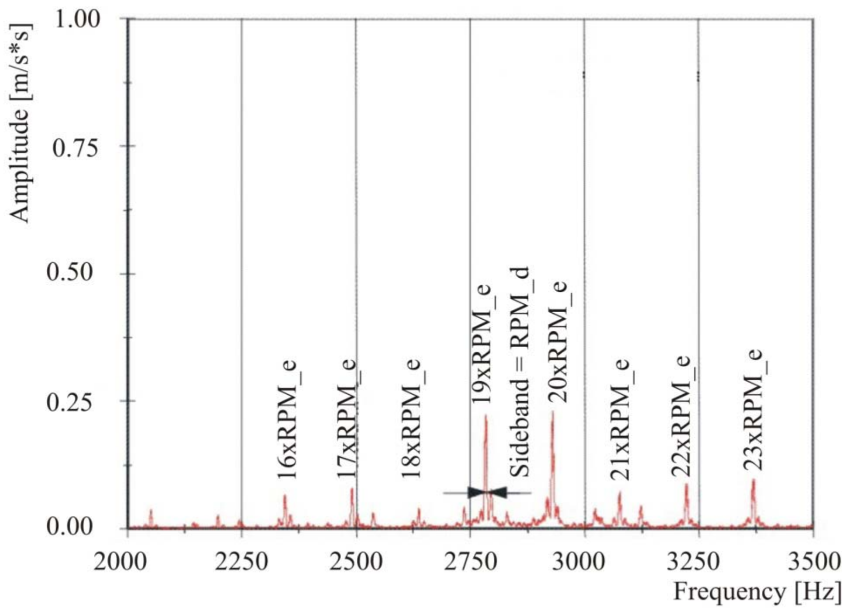

Fig. 1 The acceleration spectrum of motor A, at 149Hz

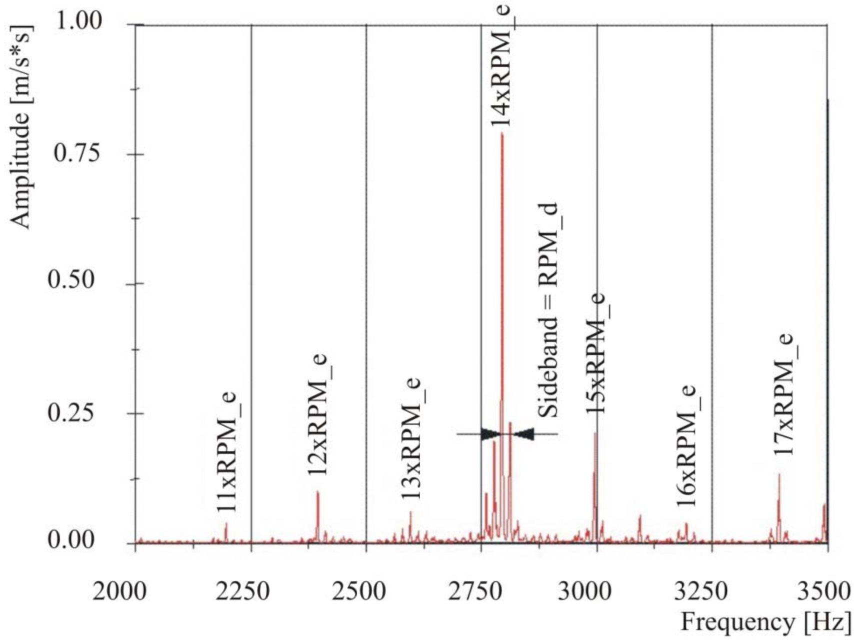

Fig. 2 The acceleration spectrum of motor A, at 199 Hz

In Figs. 3 - 4 are acceleration spectrums for motor B. In Fig. 3 are the harmonic frequencies of 146 Hz RPM and in Fig. 4 are harmonic frequencies of 221.5 Hz RPM. In this case the increasing of RPM implies a shift of the peaks at RBPF = 18xRPM in the analyzed frequency range. The peak at 18xRPM is caused by rotor motor problem.

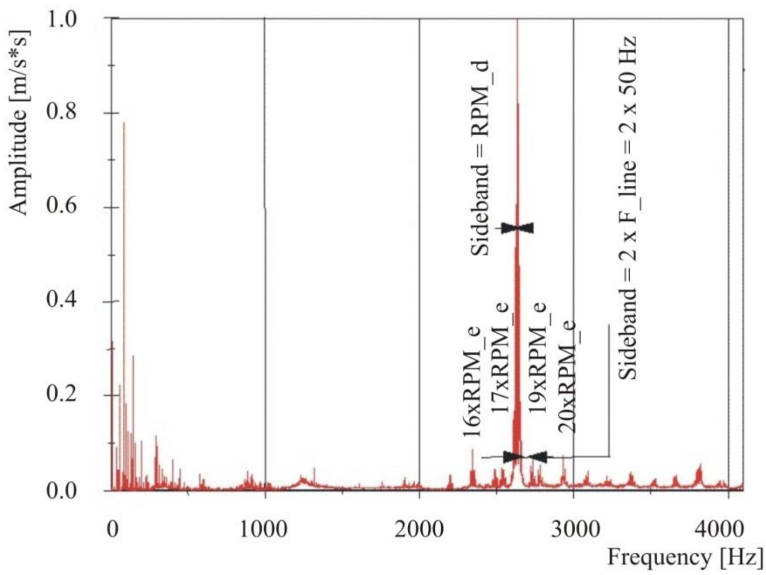

Fig. 3 The acceleration spectrum of motor B, at 146 Hz

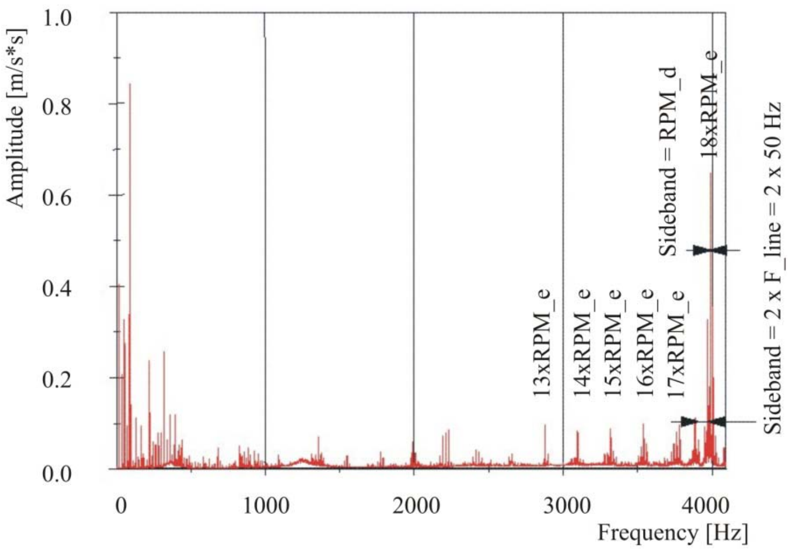

Fig. 4 The acceleration spectrum of motor B, at 221 Hz

The measurement on the bearing of drum in the axial direction is presented in Fig. 5. The peaks in this spectrum can be interpreted as a result of the events:

a. The most likely events:
   - 1x$f_{RPM,\ DRUM}$ - unbalance,
   - 1x$f_{RPM,\ DRUM}$ and 1x$f_{RPM,\ MOTOR}$, - belt / pulley misalignment,

b. The possible events:
   - 1x$f_{RPM,\ DRUM}$ with harmonics - mechanical looseness,
   - 1xf_{BELT} with harmonics - worn, loose or mismatched belt, and

$$f_{BELT} = \frac{\pi\, f_{RPM,PULLEY}\, \phi_{PITCH\ DIAMETER\ PULLEY}}{L_{BELT\ LENGTH}} \tag{2}$$

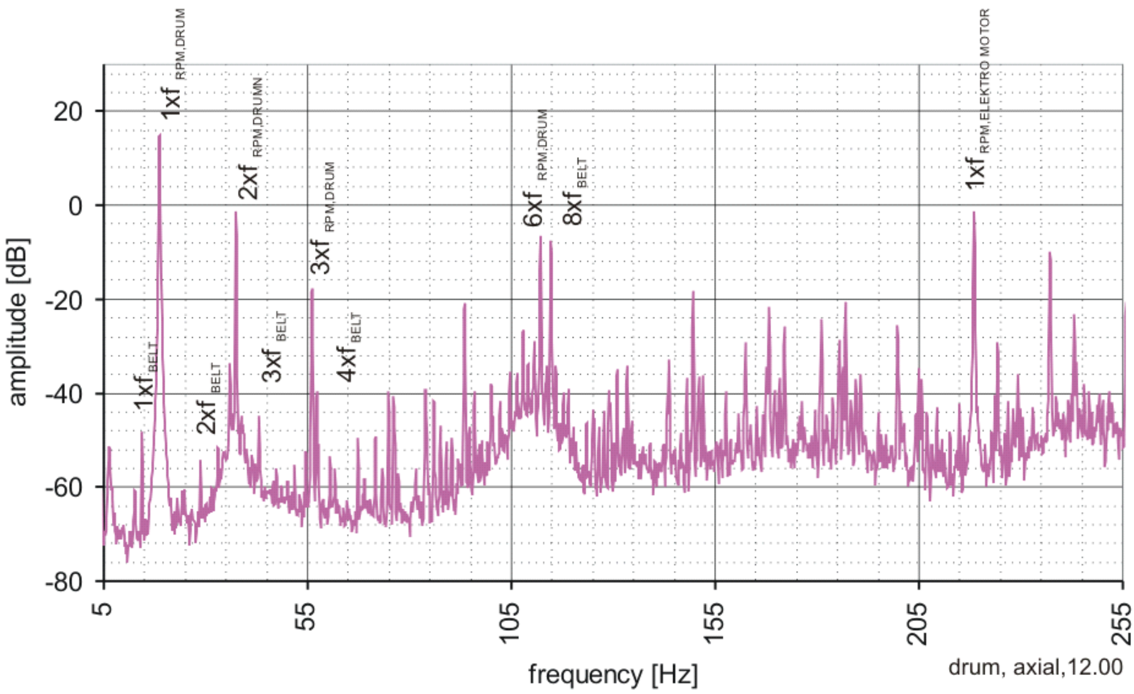

Fig. 5 The frequency spectrum of acceleration on the drum in axial direction

The high peak at RPM of the drum is symptom of unbalance. If exciter of resonance of the motor A is drum, Figs. 1-2, then unbalance is a source of an excitation.

The belt is a sensitive indicator of vibration in the mechanical system or even it appears as an amplifier of vibration. In Fig. 6 is depicted a frequency response function (FRF) of belt specifically the amplitude and the phase function. The resonance frequencies are indicated by two symptoms the peaks of amplitude at this frequencies and phase shift around 180deg at this frequencies too. If the motor run up (from 0 Hz to 200 Hz) or coast down, then it transits through belt natural frequencies with temporary increasing sound and vibration of equipment and belt can be exciter the resonances in these transition states.

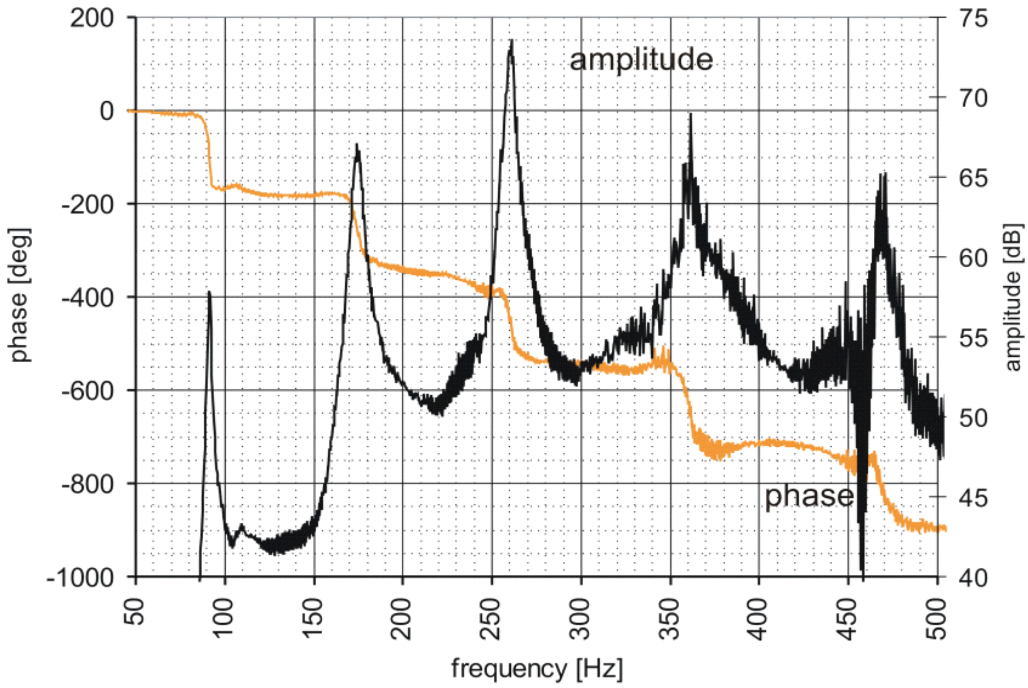

Fig. 6 The FRF of the belt

The motor and the drum are suspended in a cabinet with cover sheets by the springs and shock absorbers. The vibration of motor and drum is transmitted through these elements in the sheets of the cabinets. FRF of shock absorber also characterizes the transmission path. In Fig. 7 a natural frequency of absorber is around 250 Hz (peak at amplitude FRF = FRF_H1_dB_e-y, phase shift at phase FRF around 180deg = Phase_e-y, zero value of imaginary part FRF = FRF_Imag_e-y, extrem value of real part FRF = FRF_Real_e-y). A signal, the source is inside the cabinet and which transmits from the inside to cover sheet trough the shock absorber may be amplified in the frequency range around this natural frequency. A coherence function (Coh_e-y) takes the value equal to the unit, which corresponds to the linear model of the analysed system.

If the device is in operation and we have to do the measurements with sensors then the measured spectrum is depicted in Fig. 8. (The sensor is not placed near sources of vibration inside the cabinet, but only on the sheets. Such the measure may be the result of vibration condition monitoring.) The acceleration spectrum measured on the sheet (a dark blue lower curve) is depicted at the bottom of Fig. 8, amplitude FRF (a light blue upper smooth curve) is depicted at the top and phase FRF (a red smooth diagonal curve) is depicted too. We can see that the background of vibration spectrum and peak of acceleration are significantly determined by modal properties of sheet. Acceleration spectrum was measured during operation of the machine and FRF was measured at stop stage. An accelerometer was placed in both measurements at the same location.

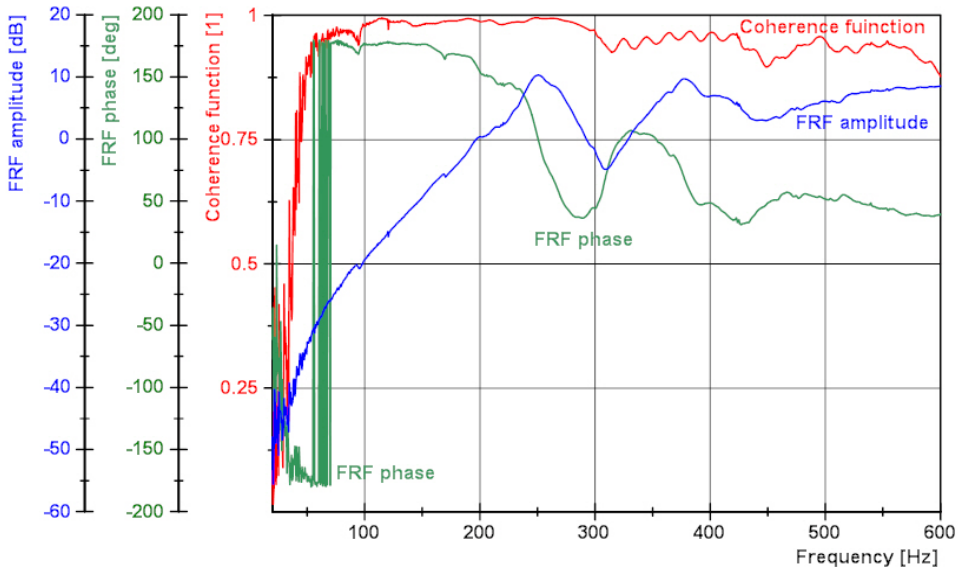

Fig. 7 The FRF of shock absorber, natural frequency at around 250Hz

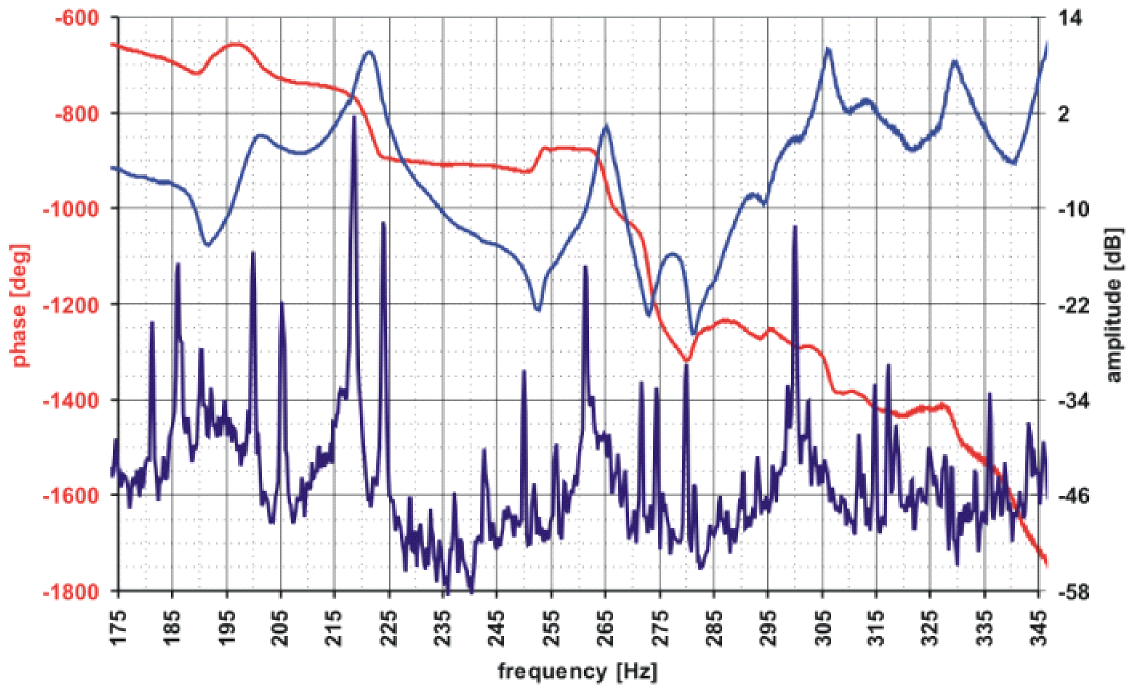

Fig. 8 The comparison of acceleration spectrum and amplitude FRF on the sheet of cabinet

4. Conclusion

The paper discusses the interpretation of vibration spectrum and FRF on the base of measure results. We deal with the problems of:

1. The vibration sources based on resonance in system, electric problem of electromotor and rotor unbalance. This case was presented to the impact of variable speed electric motors on the resonance frequency and rotor bar pass frequency.

2. The propagation of vibration from source to cover sheet panels of machine trough transfer path containing shock absorber and belt.

The impact of the belt vibration machine respectively the transmission path between two machine elements that are associated belt. It was presented in the case of resonance during run up or coast down of the machine for example as a result of the presence of natural frequencies belt. It was presented in the case of FRF function shock absorber and its selective filtering properties of the excitation frequency.

3. The measure of vibration on the panels of cabinet because these panels act as sources of undesirable sound.

The natural frequencies determine a shape of amplitude spectrum. The FRF´s (and modal shapes) are useful for update FEM model of sheet panel for next step – optimisation – decreasing of sound and vibration level.

A numerical simulation, optimization and stress analysis will be a next step. We will base upon the experiences and research works from our Department of applied mechanics, Faculty of mechanical engineering and our co-workers e.g. [4-8].

Acknowledgement

The work has been supported by the grant project VEGA 1/1000/12 and 1/1259/12.

This research project is partially supported by the EU Structural Funds, Operational Programme Research and Development, 22nd measure Transfer of knowledge and technology from research and development into practice: Project Title: Development of an optimum technology for the analysis of limit states of structural elements in contact, ITMS 26220220118

Support research activities in Slovakia / Project is co-financed by the EU

References

[1] R.K. Mobley, Root Cause Failure Analysis, Newnes, 1999.

[2] J.E. Berry, Intensive vibration diagnostics, SKF Condition Monitoring, Zaltbommel, Charlotte, 1985.

[3] C.W. de Silva, Vibration. Fundamentals and Practice, CRC Press LLC, 2000.

[4] C. Sheffer, P. Girhard, Practical Machinery Vibration Analysis and Predictive Maintenance. Newnes, 2004.

[5] M. Saga, M. Vasko, Stress Sensitivity Analysis of the Beam and Shell Finite Elements. Communications - Scientific Letters of the University of Zilina. 11 (2009) 5–12.

[6] M. Saga, P. Kopas, M. Vasko, Some Computational Aspects of Vehicle Shell Frames Optimization Subjected to Fatigue Life. Communications - Scientific Letters of the University of Zilina. 12 (2010) 73–79.

[7] M. Vasko, M. Saga, Solution of Mechanical Systems with Uncertainty Parameters using IFEA, Communications - Scientific Letters of the University of Zilina. 11 (2009) 19–27.

[8] M. Zmindak, P. Novak, J. Mesko, Numerical simulation of ARC welding processes with metalurgical transformations. Metalurgija (Metalurgy). 49 (2010) 595 – 599.