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STUDIES ON THE VALUATION OF THE WORKING SAFETY OF THE GANTRY CRANES USING VIBRATORY ANALYSIS

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Abstract: *For the calculation, the designing and the checking of the gantry crane's truss, the actual normative contain calculus notes and a succession of theoretical and experimental check absolutely necessary. Generally they are based on classical calculations of strength of materials, on the fundamental notions of the theory of elasticity but also on modern techniques by using specialized software (ANSYS, NASTRAN, FTOOL, ABAQUS etc.).*
The actual specially normative do not contain indications concerning the vibratory state of the cranes. Practice shows that the beginning of a work which gets out the normal parameters (changes of the optimum technical state) appear those vibratory tough.
The present paper deals with some aspects of great technical importance meaning, using the experimental modal analysis and an operational analysis which are useful to the periodical check of the gantry cranes..
Keywords: *oscillation, accelerometer, force transducer, frequency, Fourier transform*.

1. THE FRACTURE OF THE SUPPORT ROPE OF THE LOAD. EXPERIMENTAL DETERMINATIONS

One performed tests in order to determine the dynamic answer of the crane to changing loads due to the fracture of the drive cable during the horst of the weight. In Figure 1 is presented the connection scheme.

Figure 1: The connection scheme for the determination of the dynamic answer of the crane, due to the fracture of the drive cable

In order to simulate the fracture of the rope, between the force transducer and the crane's hook was placed a steel wire of diameter D = 3 mm, which fails at a force of about 1700 N. The force transducer was joint bound with the lifted weight.
The required devices for the registrations: acquisition system SPIDER 8 (12 bits resolution), signal conditioner NEXUS (linearity 0,01 %), signal conditioner type 2635, accelerometer Bruel & Kjaer type 4391 (linearity 2% - 4 pieces), force transducer U2B 10 kN (linearity 1 %).

One performed a number of lifting of the weight of 1444 N, with the carriage placed in different points of the crane's main beam (mainly at the middle of the beam span). One mentions that at the lift of the load of 1444 N, the force that acts in the drive rope represents the sum of the weight and of the inertial force due to the acceleration of the lifted load, so that the real force in the cable is higher than 1700 N. At this force the intejacent wire fail before detachment of the weight from the floor.

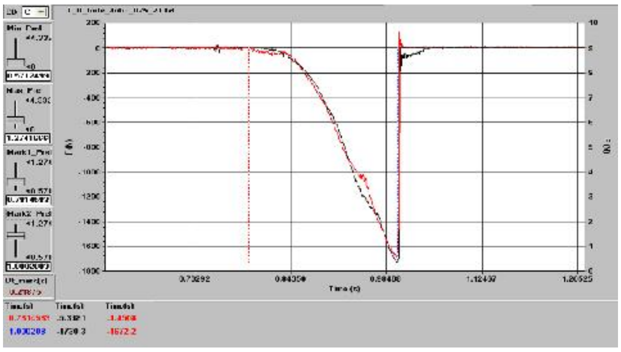

Figure 2: Internal forces at two operations for the fracture of the rope

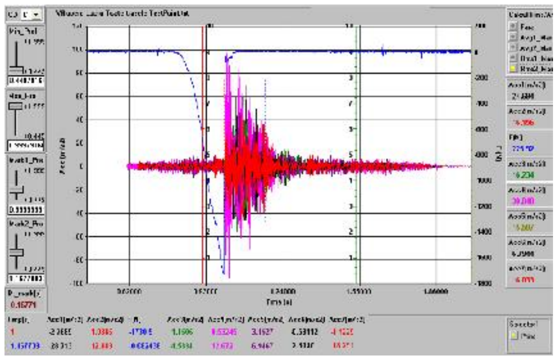

Figure 3: The answer in acceleration at the rope's fracture

Fmax. (N)	Acc1 (m/s2)	Acc2 (m/s2)	Acc3 (m/s2)	Acc4 (m/s2)	Acc5 (m/s2)	Acc6 (m/s2)	Acc7 (m/s2)
1730.5	24.69	16.15	16.23	30.84	15.59	6.79	16.03

The crane's carriage and so, the lifted weight, are located at the middle of the main beam's span. One performs two successively fractures of the cables. One observes that the obtained forces at the two operations have similar features – the fracture phenomenon is the same. The fracture force have values around 1730 N, Figure 2. In Figure 3 is presented the answer in acceleration of the crane structure. On the right side displays are recorded the effective values of the acceleration in the first 0,167 s after the failure of the drive cable (the load is placed at the middle of the main beam span), and in Table 1 are given the accelerations at the cable fracture. In the table, Acc1 … Acc7 refer to 7 successive points, placed at equal distances on the crane's main beam span.
One performed the spectral analysis of the answers in accelerations. From the analysis of the answer in frequency of the accelerations in Table 1 (due to the failure of the drive rope), one establishes that the vibrations of the main beam are broadband vibrations, the frequency spectrum of the vibratory answer being in the frequency field 0 … 400 Hz.

2. THE MODAL EXPERIMENTAL ANALYSIS USED FOR THE VALUATION OF THE TECHNICAL STATE OF THE CRANES [1]

In Experimental Modal Analysis (EMA) is the elaboration process of the mathematical model of a structure, based on experimental data obtained by measurement. These were performed on a structure brought in a controlled vibratory state. The structure is excited in precisely defined conditions (know conditions). One gets the evolution laws of the excitation and the vibratory answer; one identifies a minimum number of parameters which describe the eigenmodes of oscillation: eigenfrequencies, damping constants, modal forms.
The set of modal parameters defines the modal pattern which is associated uniquely to the real system; thus one makes the theoretical valuation of the structure's answer to different excitation conditions:
- ground applied excitation (earthquake);
- excitation in different points (load lifting, carriage motion);
- distributed excitation on the structure (wind).

The answer is given in accelerations, stresses displacement. The modal forms are geometrical diagrams of the prevailing motions of the structure to its eigenfrequencies. By analyzing the modal forms are emphasizes the weak points and the loosening or failure areas of the studied structure.
The Experimental Modal Analysis is applied together with the theoretical analysis with help of the Finite Elements Method. The two analysis work with similarly mathematical model, but that of FEM is obtained by the discretization of the real structure. Depending on the used software (ANSYS, NASTRAN, ABAQUS, etc.), on the complexity of the analyzed structure and on the user's skill, one achieves theoretical results which can describe the behavior of the real structure.

Figure 4: Scheme for the execution of the modal identification tests

Figure 5: Test execution – detail of the excitation

The assembling diagram is shown in Figure 4, the crane being mounted rigidly on the ground (welded joint). The accelerometers were mounted rigidly on the crane's structure (3 accelerometers Bruel & Kjaer, type 4391 and the impact hammer Bruel & Kjaer, type 8206, connected at the input of the conditioner NEXUS 2692 – A – 0I4, with a recording time of about 3 s). One performed the determination of the eigenfrequencies and of t5he oscillation eigenforms of the gantry crane structure.

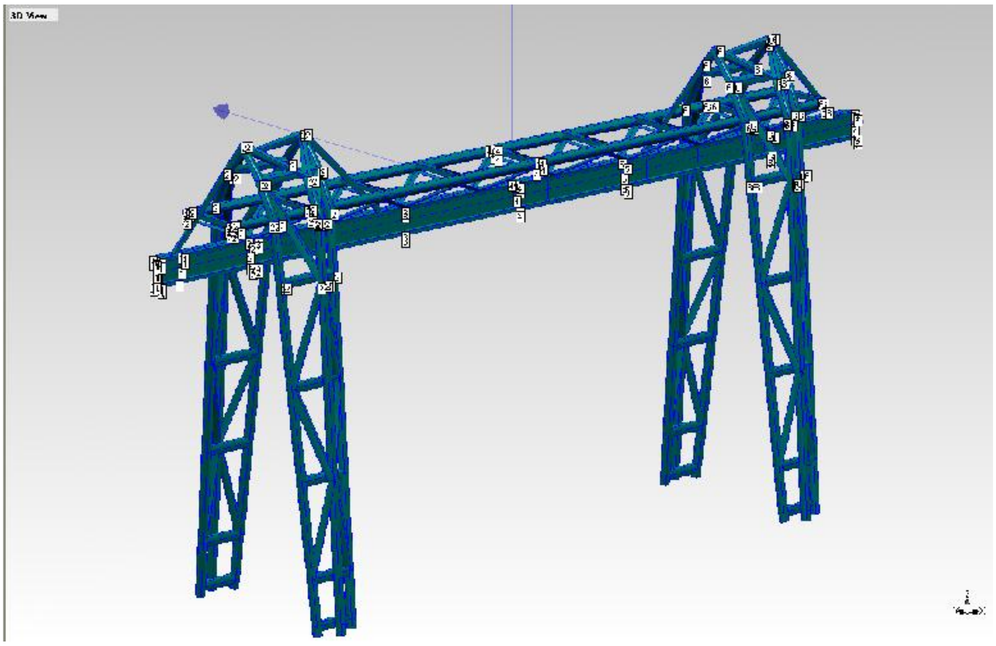

Figure 6.:The location of the measurement points on the crane's structure

The graphical structure of the crane is that presented in Figure 6, the location of the measurement points on the structure maintaining the same. In order to process the experimental data one achieved software which allows the analysis of the recorded data in time and frequency fields. One performed also a graphical view of the crane's structure dynamics both for the recorded data and for the processed data, in time and frequency field. Both the excitation signal and the vibratory answer signal are recorded. One proceeds to the determination of the frequency answer functions (FRF) as a ratio of the Fourier transforms of the answer (answer accelerations) and the Fourier transform of the input (exciting force). One draws graphs which contain functions of frequency answer and the first 10 modes of the crane's oscillation.

3. CONCLUSIONS

- The modal analysis represents a useful tool for the achievement of the eigenfrequencies and of the modal forms.

- The recognition of the eigenfrequencies is very important; the location of some structural resonances in excitation frequencies field, leads to undesired amplification of the vibratory answer.
- For a given structure, the displacement of the resonances emphasizes the material fatigue, cracks in strength structure, loosening of the assembling elements.
- The analysis of the modal forms (eigenfrequencies) may emphasize the weak points and the loosening or failure areas of the structure.
- From the above presented study, one established that the damped oscillations of the lifted load drive rope, have a changing eigenfrequency, in the field 7,8 Hz … 10,6 Hz; these are outside the eigenfrequencies and don't lead to resonance.
- In case of the drive rope failure, broadband oscillations (0 … 400 Hz) occur; these may lead to resonance in a high frequency field. We conclude by mentioning that these oscillations are transitory and do not represent a risk for the safety and the stability of the crane.

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