# RESEARCHES REGARDING MODAL ANALYSIS UTILIZATION AS A TOOL FOR FATIGUE AND STRUCTURAL CHANGE ASSESSMENT OF MECHANICAL STRUCTURES / *CERCETĂRI PRIVIND UTILIZAREA ANALIZEI MODALE CA INSTRUMENT PENTRU EVALUAREA STĂRII DE OBOSEALĂ ŞI A MODIFICĂRILOR STRUCTURALE ALE STRUCTURILOR MECANICE*

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***Abstract:*** *The paper presents theoretical and experimental researches on the use of experimental modal analysis as an instrument of appreciation of fatigue state, as well as to identify the structural changes of the complex mechanical structures. The application is carried out on a railway bogie frame, performed at SC Softronic Craiova, being in course to be tested for static and fatigue stress at INMA Bucharest. Researches are in full progress, in the article being presented the results of initial tests as well as those performed after 6 millions of stress cycles at fatigue.*

***Keywords:*** *modal analysis, eigenfrequencies, eigenmodes, fatigue, bogie frame.*

***Rezumat:*** *În lucrare sunt prezentate cercetari teoretice si experimentale privind utilizarea analizei modale experimentale ca instrument de apreciere a starii de oboseala, cat si pentru identificarea modificarilor structurale ale structurilor mecanice complexe. Aplicatia este realizata pe o rama de boghiu de cale ferata, realizat la SC Softronic Craiova, aflat la incercari la solicitari statice si oboseala la INMA Bucuresti. Cercetarile sunt in desfasurare, in articol fiind prezentate rezultatele privind incercarile initiale, precum si cele dupa 6 milioane de cicluri de solicitari la oboseala.*

***Cuvinte cheie:*** *anliza modală, frecvente proprii, moduri proprii, oboseala, rama de boghiu.*

## INTRODUCTION

Bogies are complex equipments with a vital role in the functioning of railway vehicles, having the role of carbody supporting, of ensuring the traction and braking forces, as well as of vibratory isolation of the carbody and transported loads. Taking into account the important role in the rolling stock security, the fact that all mechanical stresses from the rolling track are transmitted to the carbody through the bogie, as well as the long operating life of rolling stock, the present regulations require that at homologation, the bogies to be subjected to a complex set of static and fatigue tests.

For achieving a stand and a testing program it is taken into consideration that the strains at which the bogie is subjected replicate as accurately as possible the strains of normal functioning, without introducing additional or unrealistic constraints or degrees of freedom.

The bogies testing shall be made according to European standard EN 13749 *"Railway applications. Wheelsets and bogies. Methods of specifying the structural requirements of bogies frames"*.[6]

In practice there are standardised two static load cases, which are considered to be covering for the real situations encountered during an equipment lifetime:

- *exceptional static loads*, which may occur only rarely, over the lifetime of the bogie.The bogie structure is necessary to resist to these loadings, without cracks or deformations, that would affect the operation during tasks application;
- normal service loads, which represent those loads currently occurring during lifetime, the bogie having to withstand at service loads, without fatigue cracks.

The test at static loads is followed by fatigue testing, which is designed to confirm that the bogie frame is capable of withstanding to stresses due to operating loadings encountered throughout its whole life. The main loads acting are those responsible for the induction of mechanical stress in the whole structure of the bogie frame, namely: vertical forces, transversal forces and forces due to twist stresses. The dynamic loads are applied as follows:

- 6 million cycles with normal service loads;
- 2 million cycles with service loads increased by 20%;

## INTRODUCERE

Boghiurile sunt echipamente complexe cu rol vital in functionarea vehiculelor feroviare, avand rolul de purtator al carcasei, de asigurare a fortelor de tractiune si de franare, precum si de izolare vibratorie a carcasei si a incarcaturii transportate. Avand in vedere rolul important in securitatea materialului rulant, faptul ca toate solicitarile de la calea de rulare se transmit la carcasa prin intermediul boghiului, precum si durata mare de functionare a materialului rulant, normativele in vigoare impun ca la omologare, boghiurile sa fie supuse unui set complex de incercari la solicitari statice si oboseala.

La realizarea unui stand si a unui program de incercari se are in vedere ca solicitarile la care este supus boghiul sa reproduca cat mai fidel solicitarile din functionarea normala, fara a introduce constrangeri sau grade de libertate suplimentare sau nerealiste.

Incercarea boghiurilor pentru materialul feroviar se face conform standardului european *EN 13749 " Aplicații feroviare. Osii și boghiuri. Metode de specificare a cerințelor structurale ale ramelor de boghiuri"*.[6]

In practica sunt normate doua cazuri de incercare statica, care se considera a fi acoperitoare pentru situatiile reale intalnite pe durata de de viata a echipamentului:

- *incarcarea statica exceptionala*, care poate sa apara doar rar, pe durata de viata a boghiului. Structura boghiului este necesar sa reziste la aceste sarcini, fara fisuri sau deformatii, care ar afecta functionarea in timpul aplicarii sarcinilor;
- *incarcarea de exploatare*, care reprezinta acele sarcini care apar curent in timpul exploatarii, boghiul trebuind sa reziste sarcinilor functionale, fara aparitia de fisuri.

Incercarea la solicitari statice este urmata de incercarea la oboseala, care este destinata sa confirme ca rama boghiului este capabila sa reziste solicitarilor datorate sarcinilor de exploatare intalnite pe toata durata de viata a acestuia. Sarcinile principale care actioneaza sunt cele responsabile de inducerea solicitarilor mecanice in intreaga structura a ramei de boghiu si anume: fortele verticale, fortele transversale si fortele datorate solicitarilor de rasucire. Sarcinile dinamice se aplica dupa cum urmeaza:

- 6 milioane cicli cu sarcinile de exploatare;
- 2 milioane cicli cu sarcini de exploatare crescute cu 20%;

- 2 million cycles with service loads increased by 40%.

During the tests performing, in the bogie frame structure appear irreversible phenomena of material fatigue. Beside these, may also appear cracks or structure damages. Throughout the tests duration, it is performed the monitoring of the mechanical stress on the bogie frame structure.

At the present moment, there is in progress a contract for performing the static and fatigue tests at INMA Bucharest, for two railway bogies made at SC Softronic Craiova. With this occasion, within SC Softronic Craiova was developed and experimented a technology for monitoring of fatigue state and the structural integrity, by using the experimental modal analysis.

The paper presents the theoretical background of the developed technology, as well as the result of tests performed on a bogie frame which is in the initial stage of tests, as well as after the static tests and of the first stage of fatigue, after the 6 million exploitation cycles.

The research is in progress, and in a future article will be presented the final results of the research.

## MATHERIAL AND METHOD

### Theoretical basis of the experimental modal analysis [2]

Any mechanical system can be modeled by means of a discrete system consisting of 'n' material points of concentrated mass '$m_k$' connected by stiffness elastic elements '$k_k$' and damping elements of '$c_k$' constant. For this damped system subjected to the action of an external excitations system $\{Q(t)\}$, the equations of movement are:

- 2 milioane cicli cu sarcini de exploatare crescute cu 40%.

Pe durata efectuarii incercarilor, in structura ramei de boghiu apar fenomene ireversibile de oboseala a materialului. Pe langa acestea, mai pot sa apara fisuri sau cedari ale structurii. Pe toata durata incercarilor, se efectueaza monitorizarea tensiunilor mecanice din structura ramei de boghiu.

In momentul de fata se afla in derulare un contract pentru incercarea la solicitari statice si oboseala la INMA Bucuresti, a doua boghiuri de material feroviar realizate la SC Softronic Craiova. Cu aceasta ocazie, la SC Softronic Craiova, a fost elaborata si experimentata o tehnologie de monitorizare a starii de oboseala si a integritatii structurale, prin utilizarea analizei modale experimentale.

In articol se prezinta fundamentul teoretic al tehnologiei elaborate, precum si rezultatul incercarilor efectuate pe o rama de boghiu aflata in stadiul initial al incercarilor, precum si dupa efectuarea incercarilor statice si a primului stadiu de oboseala, dupa cele 6 milioane de cicli de exploatare.

Cercetarea se afla in derulare, iar intr-un articol viitor se vor prezenta rezultatele finale ale cercetarii.

## MATERIAL SI METODA

### Baza teoretica a analizei modale experimentale [2]

Orice sistem mecanic poate fi modelat printr-un sistem discret format din 'n' puncte materiale de masa concentrata '$m_k$' unite prin elemente elastice de rigiditate '$k_k$' si elemente de amortizare de constanta '$c_k$'. Pentru acest sistem amortizat supus actiunii unui sistem de excitatii exterioare $\{Q(t)\}$, ecuatiile de miscare sunt:

$$[M]\{\ddot{x}(t)\}+[C]\{\dot{x}(t)\}+[K]\{x(t)\}=\{Q(t)\} \quad [1]$$

The system response at external excitation is presented as a sum of 'n' modal contributions due to each separate degree of freedom:

Raspunsul sistemului la excitatia externa se prezinta sub forma unei sume de 'n' contributii modale datorate fiecarui grad de libertate separat:

$$\{X(\omega)\}=\sum_{k=1}^{N}\left[\frac{\{\psi^k\}\cdot\{\psi^k\}^T\cdot\{Q(\omega)\}}{a_k(-\mu_k+i(\omega-\nu_k))}+\frac{\{\overline{\psi^k}\}\cdot\{\overline{\psi^k}\}^T\cdot\{Q(\omega)\}}{\overline{a}_k(-\mu_k+i(\omega+\nu_k))}\right] \quad [2]$$

where:

- $\{\psi^k\}$ and $\{\overline{\psi^k}\}$ - its own vector of the order "k";
- $\mu_k$ - the "k" order damping ratio;
- $\nu_k$ - the "k" order damped natural frequency;
- $a_k$ and $\overline{a}_k$ - norming constants;
- $\omega$ - external excitation frequency.

In order to determine the eigenvectors from experimental data, in the paper it is presented the one point excitation method. It consists in the structure excitation in the successively points 'j' (j=1,2,…m) and simultaneous determination of the response (in accelerations or displacements) in points 'i' (i=1,2,…n). In practical applications, the eigenvectors are substituted with two modal constants $U_{ij}^k$ and $V_{ij}^k$ defined by following:

unde:

- $\{\psi^k\}$ and $\{\overline{\psi^k}\}$ - vectorul propriu de ordinul "k";
- $\mu_k$ - rata de amortizare de ordinul "k";
- $\nu_k$ - frecventa naturala amortizata de ordinul "k";
- $a_k$ si $\overline{a}_k$ - constante de normare;
- $\omega$ - frecventa excitatiei externe.

In vederea determinarii vectorilor proprii din date experimentale, in articol se prezinta metoda de excitare a structurii intr-un singur punct. Aplicarea metodei consta in excitare succesiva a structurii in punctele 'j' (j=1,2,...m) si determinarea simultana a raspunsului (in acceleratii sau deplasari) in punctele 'i' (i=1,2,...n). In aplicatiile practice vectorii proprii sunt inlocuiti cu doua constante modale $U_{ij}^k$ si $V_{ij}^k$ definite prin relatia:

$$\frac{\psi_i^k\cdot\psi_j^k}{a_k}=U_{ij}^k+i\cdot V_{ij}^k \text{ and / şi } \frac{\overline{\psi}_i^k\cdot\overline{\psi}_j^k}{\overline{a}_k}=U_{ij}^k-i\cdot V_{ij}^k \quad [3]$$

It can be introduced the structure admittance, or compliance, by ratio between displacement response and force excitation. Having in mind the (2) relation, the structure admittance can be writen as:

Se introduce marimea admitanta, sau complianta, a sistemului, definita ca raportul dintre raspunsul in deplasare si forta de excitare. Tinand cont de relatia (2) admitanta sistemului se scrie ca:

$$\alpha_{ij}(\omega)=\sum_{k=1}^{n}\frac{U_{ij}^k+i\cdot V_{ij}^k}{-\mu_k+i\cdot(\omega-\nu_k)}+\sum_{k=1}^{n}\frac{U_{ij}^k-i\cdot V_{ij}^k}{-\mu_k+i\cdot(\omega+\nu_k)} \quad [4]$$

The last relation, with $i=1,2,...n$; $j=1,2,...m$, defines the frequency response functions of the mechanical system.
In the above approximations made to achieve the mathematical model, the concept of discrete system with mass concentrated in 'n' material points was used. For the discrete system closely approximate the real system, should be necessary that 'n' be very high, practically ($n \to \infty$). In practice, this thing cannot be possible due to both physical arguments and limitations imposed by the measurement, excitation and computing equipment. In applications the frequency range is limited to a reasonable value established by the major frequencies of analyzed equipment and purpose of the application. In these conditions the sum of equations (4) is reduced to a few components, further noted with 'n' too.

The contributions of the lower and higher vibration modes are included in some correction factors named "lower modal admittance " $-\dfrac{1}{M_{ij}^{'}\omega^2}$, for lower modes, and "residual flexibility " $S_{ij}^{'}$, for upper modes.

The system admittance will have the expression:

Ultima relatie, cu $i=1,2,...n$; $j=1,2,...m$, defineste setul functiilor de raspuns in frecventa ale sistemului mecanic.
In aproximatiile facute la realizarea modelului matematic a fost utilizat conceptul de sistem discret cu masa concentrata in 'n' puncte materiale. Pentru ca sistemul discret sa aproximeze fidel sistemul real, trebuie ca 'n' sa fie foarte mare, practic trebuie ca ($n \to \infty$). In practica acest lucru nu este posibil atat din considerente legate de tehnicile de excitare si de masurare a raspunsului, cat si din considerente legate de tehnica de calcul utilizata si de timpul necesar pentru prelucrari. In aplicatii domeniul frecventelor de lucru este limitat la o valoare rezonabila stabilita in functie de frecventele majore ale echipamentului analizat, cat si de scopul aplicatiei. In aceste conditii suma din relatia (4) se reduce la cateva componente, notate in continuare tot cu 'n'.

Contributia modurilor inferioare si superioare se include in doi factori de corectie denumiti "admitanta modala inferioara" $-\dfrac{1}{M_{ij}^{'}\omega^2}$,pentru modurile inferioare, respectiv "flexibilitate reziduala", $S_{ij}^{'}$, pentru modurile superioare.

Admitanta sistemului va avea expresia:

$$\alpha_{ij}(\omega)=\frac{-1}{M_{ij}^{'}\cdot\omega^2}+\sum_{k=1}^{n}\left(\frac{U_{ij}^{k}+i\cdot V_{ij}^{k}}{-\mu_k+i\cdot(\omega-\nu_k)}+\frac{U_{ij}^{k}-i\cdot V_{ij}^{k}}{-\mu_k+i\cdot(\omega+\nu_k)}\right)+S_{ij}^{'} \qquad [5]$$

An eigenmode is defined by a set of modal parameters, which are intrinsic characteristics of the system, independent of the external conditions: $\{\psi^k\}, \mu_k, \nu_k, a_k, \quad k=1,2,...,n$, or by a combination of the modal parameters and modal constants, which depends on the external conditions:

Un mod propriu este definit printr-un set de parametri modali, care sunt caracteristici intrinseci ale sistemului, independenti de conditiile externe: $\{\psi^k\}, \mu_k, \nu_k, a_k$, sau printr-o combinatie de parametri modali si de constante modale, dependente de conditiile externe de excitare.

$$\mu_k, \nu_k, U_{ij}^{k}, V_{ij}^{k}, -\frac{1}{M_{ij}^{'}}, S_{ij}^{'}, \quad k=1,2,...,n$$

Modal analysis consists in determination of the modal parameters from experimental tests carried out on the equipment brought in a controlled vibration state and simultaneously measurement of the applied excitation and structure response. The controlled vibration state can be achieved by using one of the following low-level one-point excitation methods: the relaxed step force, sinusoidal or large band steady-state vibration excitation or impact force method.The impact force excitation method is very good for modal analysis of bogie frame.

Aplicarea analizei modale consta in determinarea parametrilor modali pe baza incercarilor experimentale efectuate pe echipamentul adus intr-o stare controlata de vibratii, cu determinarea simultana a excitatiei si a raspunsului. Starea de vibratie poate fi realizata prin una dintre urmatoarele metode de excitare de nivel energetic coborat: treapta relaxata, excitatie sinusoidala stationara sau de banda larga, excitare cu impuls de forta. Metoda de excitare cu impuls de forta este recomandata pentru analiza modala a ramei de boghiu.

**Package programs for modal analysis**

Based on the above presented theoretical background it had been achieved a package of calculation programs with major orientation for modal analysis of mechanical structures. The package is realised under TestPoint programming medium, has a modular conception, including the following programs:

- ***'ModalAch'*** is a module to control the excitation and system response during the test.
- ***'IdModal'*** is a module for calculating the frequency response functions and modal parameters. The frequency response functions are calculated using a selective length of data and some pondering windows. For modal parameter identification there are used some sophisticate linear and non-linear regressive procedures.
- ***'ModalForm'***, is a module for eigenfrequencies assessment, vibration eigenmode calculating and for graphical animation of the structure in its eigenmode.

The program reads data from files achieved with 'IdModal' program.
The programs description will be made during the presentation of tests on the bogie frame.

**Pachet de programe pentru analiza modala**

Pe baza celor prezentate anterior a fost realizat un set de programe de calcul cu orientare majora catre analiza modala a structurilor mecanice. Pachetul de programe este realizat sub mediul de programare TestPoint, are o conceptie modulara, cuprinzand urmatoarele programe:

- ***'ModalAch',*** este un modul pentru controlul achizitiei datelor reprezentand excitatia si raspunsul sistemului.
- ***'IdModal',*** este un modul pentru calculul functiilor de raspuns in frecventa si a parametrilor modali. Functiile de raspuns in frecventa sunt calculate utilizand o lungime selectiva a datelor, precum si diverse ferestre de ponderare. Pentru identificarea parametrilor modali se utilizeaza proceduri de regresie liniara si neliniara.
- ***'ModalForm',*** este un modul pentru evaluarea frecventelor proprii, a formelor proprii de vibratie si pentru animatia grafica a structurii, in modurile de vibratie.

Programul citeste datele continute in fisierele realizate cu programul 'IdModal'.
Descrierea programelor va fi realizata pe masura prezentarii experimentarilor pe rama de boghiu.

**Measuring equipment „Softronic Data Acquisition System”**

The equipment is a portable construction type "diplomat", having incorporated the following elements:
- DAQ acquisition interface type - USB-30A16 (16 analog channels, 500 kHz sampling, 16 bit resolution);
- support plate for 16 amplifier modules with galvanic isolation type SCMB;
- external transducers for amplification modules type SCMB;
- piezoelectric acceleration transducers type 353B32, powered with amplifier modules type SCM5B48;
- impact hammer with full strain gauge powered with amplifier modules type SCM5B39.

*Main technical characteristics:*
- analog imputs: 16;
- digital I / O: 24;
- analog outputs: 4;
- input voltage: ±10V;
- protection against the continuously applied voltage by amplifier modules SCMB: 240 $V_{RMS}$;
- maximum sampling rate: 500 kHz;
- resolution: 14 bit for both analog inputs / outputs.

**Echipament de masura „Softronic Data Acquisition System”**

Echipamentul este o constructie portabila, de tip „diplomat”, si are inglobate urmatoarele elemente:
- interfata de achizitie tip µDaq – USB- 30A16 (16 canale analogice, 500 kHz esantionare, 16 bit rezolutie);
- placa suport pentru 16 module de amplificare cu izolare galvanica de tip SCMB;
- traductoare externe si module de amplificare cu izolare galvanica de tip SCMB;
- traductoare piezoelectrice de acceleratie tip 353B32, alimentate prin module de amplificare tip SCM5B48;
- ciocan de impact cu marci tensometrice alimentat prin modul de amplificare tip SCM5B39.

*Principale caracteristici tehnice:*
- intrari analogice: 16;
- I / O digitale: 24;
- iesiri analogice: 4;
- domeniul tensiunilor de masura: ±10V;
- protectie la tensiuni aplicate continuu la intrarile analogice, prin module tip SCMB: 240 VRMS;
- frecventa maxima de esantionare: 500 kHz;
- rezolutia: 14 biti pentru intrarile si iesirile analogice.

**Fig. 1** - Measuring equipment „Softronic Data Acquisition System” /
*Echipament de masura „Softronic Data Acquisition System”*

Experiments were performed in the Laboratory of Dynamic Testing of INMA Bucharest, bogie frame being suspended in crane hook by four inextensibile straps as representation in Fig.2.

It was applied the same procedure for modal identification in the initial stage of testing and after the first stage of fatigue, after the 6 million fatigue cycles.

Accelerometers were mounted, successively in vertical and horizontal directions, in measurement points P1(Acc1) ... P6(Acc6), considered to be representative for frame dynamics. Excitation was applied successively in the same measurement points using an impact hammer of about 3.5 kg, fitted with a rubber pad to protect the frame and increase pulse duration (fig. 3).

There were simultaneously measured the excitation force and the response accelerations at a sampling frequency of 10 kHz. In Figure 3 is presented an example of record obtained at the structure excitation in the point P1 and the response measurement in the points P1 (Acc1) … P6(Acc6).

In the displays at the bottom part are transmitted the instantaneous values of the characteristics at the moments selected by cursors. Association of routes with the displays is done through color. The force impulse duration is approx. 5ms.

Experimentarile au fost efectuate in Laboratorul de Incercari Dinamice al INMA Bucuresti, rama de boghiu fiind suspendata in carligul macaralei prin intermediul a patru chingi inextensibile, conform reprezentarii din fig. 2.

A fost aplicata aceiasi procedura de identificare modala in stadiul initial al incercarilor si dupa efectuarea primului stadiu de oboseala, dupa cele 6 milioane cicli.

Accelerometrele au fost montate, succesiv pe directiile verticala si orizontala, in punctele de masura P1(Acc1) ... P6(Acc6), considerate a fi reprezentative pentru dinamica ramei. Excitatia a fost aplicata succesiv in aceleasi puncte de masura, cu un ciocan de impact de cca. 3,5 kg, prevazut cu tampon de cauciuc pentru protejarea ramei si marirea duratei impulsului (fig. 3).

Au fost masurate simultan forta de excitare si acceleratiile de raspuns la o frecventa de esantionare de 10 kHz. In fig.3 este prezentat un exemplu de inregistrare obtinuta la excitarea structurii in punctul P1 si masurarea raspunsului in punctele P1(Acc1) … P6(Acc6).

In afişările din partea inferioara sunt transmise valorile instantanee al caracteristicilor la momentele selectate prin cursoare. Asocierea traseelor cu display-urile se face prin culoare. Durata impulsului de forta este de cca. 5ms.


**Fig. 2** - Experiments for modal identification of a bogie frame /
*Experimentari pentru identificarea modala a unei rame de boghiu*

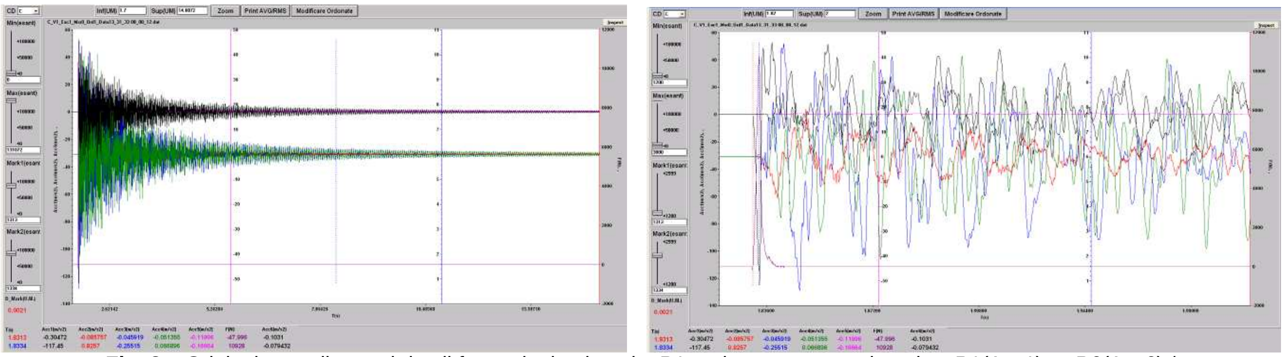

**Fig. 3** – Original recording and detail for excitation in point P1 and measurement in points P1(Acc1) … P6(Acc6) /
*Inregistrare originala si detaliu pentru excitare in punctul P1 si masurare in punctele P1(Acc1) … P6(Acc6)*

**Modal identification -** The paper presents only the tests result carried out on the vertical direction.

For each of the excitation points are succesively selected the measurement points P1...P6 and was determined the frequency response function (FRF), as the ratio between the Fourier transform of acceleration response and the Fourier transform of the excitation force. For the modal identification was adopted a model of system with generalized viscous damping. For such a system, near to a resonance frequency, the imaginary part of the FRF shows a maximum or a minimum. The real part crosses through zero presenting a maximum and a minimum on both sides of the resonance frequency. The representation in polar coordinates of FRF enrolls on a circle.

In Figure 4 are represented the frequency response functions in Cartesian and polar coordinates, for the case of excitation in point P1 and response measurement in point P1. It can be noticed that in the frequency range 0 ... 360 Hz the excitation force presents the spectral consistency, and the bogie frame has a number of the least 18 resonant frequencies.

Taking into account the above considerations, it is limited the frequency range between 0 and 230 Hz, comprising 15 resonant frequencies.

The modal identification is performed using successive complex procedures of linear and nonlinear regression. If the vibration modes are multiple and some are closely as frequency, we proceed to a partial identification on groups of close modes. As the identification, the modes are stored and in the final stage it is proceeded at identification by nonlinear regression of all modes from the frequency range of interest. In Figure 5 are presented a partial identification panel and the final identification panel, over all 15 modes of interest. In both panels are presented in overlay mode the theoretically determined paths (continuous line) and experimentally determined paths

**Identificarea modala -** In lucrare se prezinta doar rezultatul incercarilor efectuate pe directia verticala.

Pentru fiecare din punctele de excitatie se selecteaza succesiv punctul de masura P1...P6 si se determina functia de raspuns in frecventa (FRF), ca raport dintre transformata Fourier a raspunsului in acceleratie si transformata Fourier a fortei de excitare. Pentru identificarea modala a fost adoptat un model de sistem cu amortizare vascoasa generalizata. Pentru un asemenea sistem, in apropierea unei frecvente de rezonanta, partea imaginara a FRF prezinta un maxim sau un minim. Partea reala trece prin zero, prezentand un maxim si minim un de o parte si de alta a frecventei de rezonanta. Reprezentarea in coordonate polare a FRF se inscrie pe traiectoria unui cerc.

In fig.4 sunt reprezentate functiile de raspuns in frecventa in coordonate carteziene si polare, pentru cazul excitarii in punctul P1 si masurarii raspunsului in punctul P1. Se observa ca in domeniul de frecventa 0...360 Hz forta de excitare prezinta consistenta spectrala, iar rama de boghiu prezinta un numar de cel putin 18 frecvente de rezonanta.

Avand in vedere considerentele prezentate anterior, se limiteaza domeniul de frecventa intre 0 si 230 Hz, care cuprinde 15 frecvente de rezonanta.

Identificarea modala se realizeaza utilizand proceduri complexe, succesive, de regresie liniara si neliniara. Daca modurile de vibratie sunt multiple, iar unele sunt apropiate ca frecventa, se procedeaza la o identificare partiala, pe grupe de moduri apropiate. Pe masura identificarii, modurile sunt stocate iar in etapa finala se procedeaza la identificarea prin regresie neliniara a tuturor modurilor din domeniul de frecventa de interes. In fig.5 este prezentat un panel de identificare modala partiala si panelul final de identificare, peste toate cele 15 moduri de vibratie de interes. In ambele panele sunt prezentate suprapus trasele determinate teoretic (linie continua) si trasele determinate (dashed lines). The real parts are represented with red, and the imaginary parts with blue. The fact that theoretically determined paths are overlapping over the experimentally determined paths, highlights that the system model was correctly chosen and a fair identification of modal parameters.

The modal parameters are stored in a data file that has a number of lines equal to *Number of Excitation Points* x *Number of Measuring Points* x *Number of Modes*. For this case the file contains 540 lines.

experimental (linie intrerupta). Partile reale sunt reprezentate cu trase rosii, iar partile imaginare cu trase albastre. Faptul ca trasele determinate teoretic se suprapun practic peste trasele determinate experimental, evidentiaza un model de sistem corect ales si o fidela identificare a parametrilor modali.

Parametrii modali sunt stocati intr-un fisier de date care are un numar de linii egal cu *Numar Puncte Excitare* x *Numar Puncte Masura* x *Numar Moduri*. Pentru cazul de fata fişierul contine 540 linii.

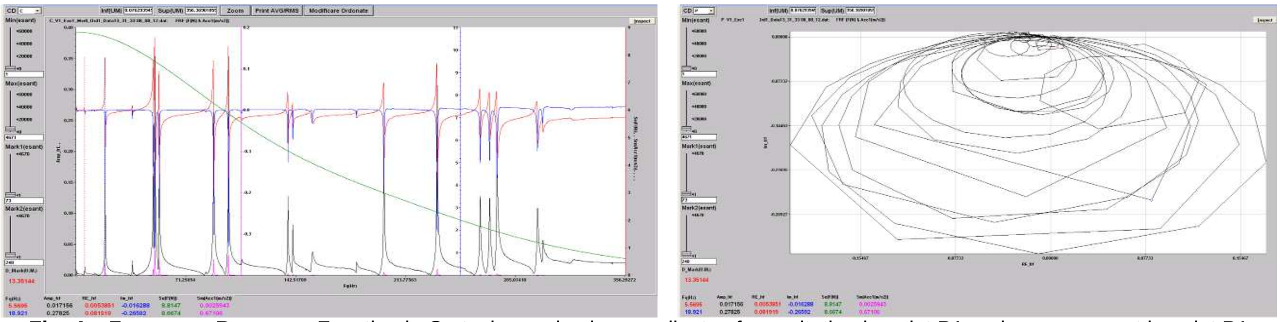

**Fig. 4** – Frequency Response Function in Cartesian and polar coordinates for excitation in point P1 and measurement in point P1
*Reprezentarea FRF in coordonate Cartesian si polare pentru cazul excitarii in punctul P1 si masurarii raspunsului in punctul P1*

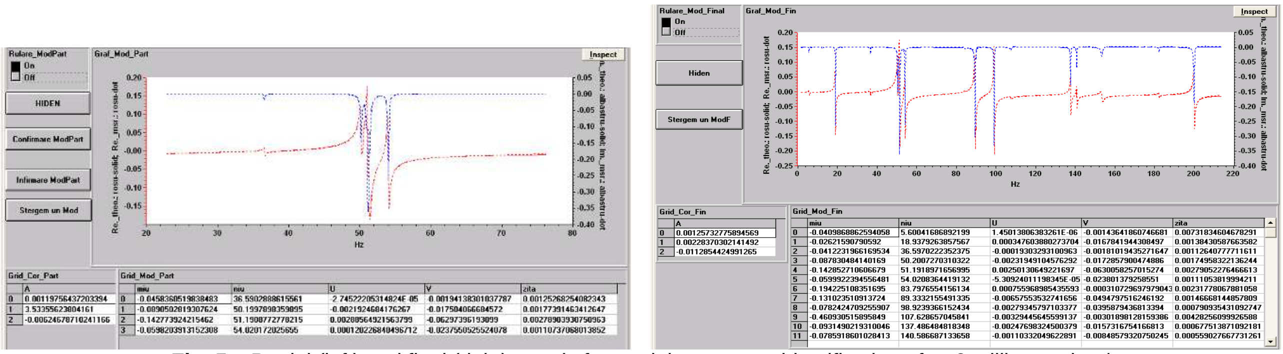

**Fig. 5** – Partial (left) and final (right) panels for modal parameter identification after 6 million cycles /
*Panel partial (stanga) si final (dreapta) pentru identificarea parametrilor modali dupa 6 milioane ciclii*

## RESULTS

### Using modal analysis to validate the constructive concept and the structural integrity

From the previous analysis it follows that the elastic system consisting of the bogie frame, suspension cable and crane is characterized by the existence of 15 eigenfrequencies in the range of 0...230 Hz. From these, not all are eigenfrequencies of bogie frame. For their identification it is necessary to analyze the system eigeshapes and to eliminate the modes of rigid body.

The analysis is done with the 'ModalForm' module which, in the first stage performs the three dimensional graphical representation of the analyzed structure, with location of points where the vibratory response was measured. For representation are used the Euler angles, allowing the structure rotation with proper emphasizing of the oscillation forms.

It is read the file of modal parameters and determined the eigenfrequencies. In an eigenmode are determined the oscillation amplitude and phase of the points response.

The module performs the structure animation in their eigenmodes, by overlapping of the deformed state, due to the oscillation, over the undeformed state of the structure. To highlight the oscillation modes, the deformed state can be amplified by a factor of amplification *,,Amp''*, common to all structure points. The module allows keeping the paths footprint in order to achieve an intuitive visual representation of the oscillation forms. In fig. 6 is represented the elastic system in the first 14 vibration eigenmodes.

## REZULTATE

### Utilizarea analizei modale pentru validarea conceptiei constructive si a integritatii structurale

Din analiza anterioara rezulta ca sistemul elastic format din rama de boghiu, cablu de suspensie si macara este caracterizat de existenta a 15 frecvente proprii in domeniul 0...230 Hz. Dintre acestea nu toate sunt frecvente proprii ale ramei de boghiu. Pentru identificarea lor este necesara analiza formelor proprii de vibratie ale sistemului si eliminarea modurilor de corp rigid.

Analiza se face cu modulul ***'ModalForm'*** care, in prima etapa, realizeaza reprezentarea grafica tridemensionala a structurii analizate, cu localizarea punctelor in care a fost masurat raspunsul vibratoriu. Pentru reprezentare se utilizeaza unghiurile lui Euler, care permit rotirea structurii cu evidentierea corecta a formelor de oscilatie.

Se citeste fisierul parametrilor modali si se determina frecventele proprii. Intr-un mod propriu se determina amplitudinea si faza oscilatiei punctelor de raspuns.

Modulul realizeaza animatia structurii in modurile proprii, prin suprapunerea starii deformate, datorata oscilatiei, peste starea nedeformata a structurii. Pentru evidentierea modurilor de oscilatie, starea deformata poate fi amplificata cu un coeficient de amplificare, *,,Amp''*, comun pentru toate punctele structurii. Modulul permite mentinerea urmei traselor in vederea unei reprezentari vizuale intuitive a formelor de oscilatie. In fig. 6 este reprezentat sistemul elastic in primele 14 moduri proprii de vibratie.

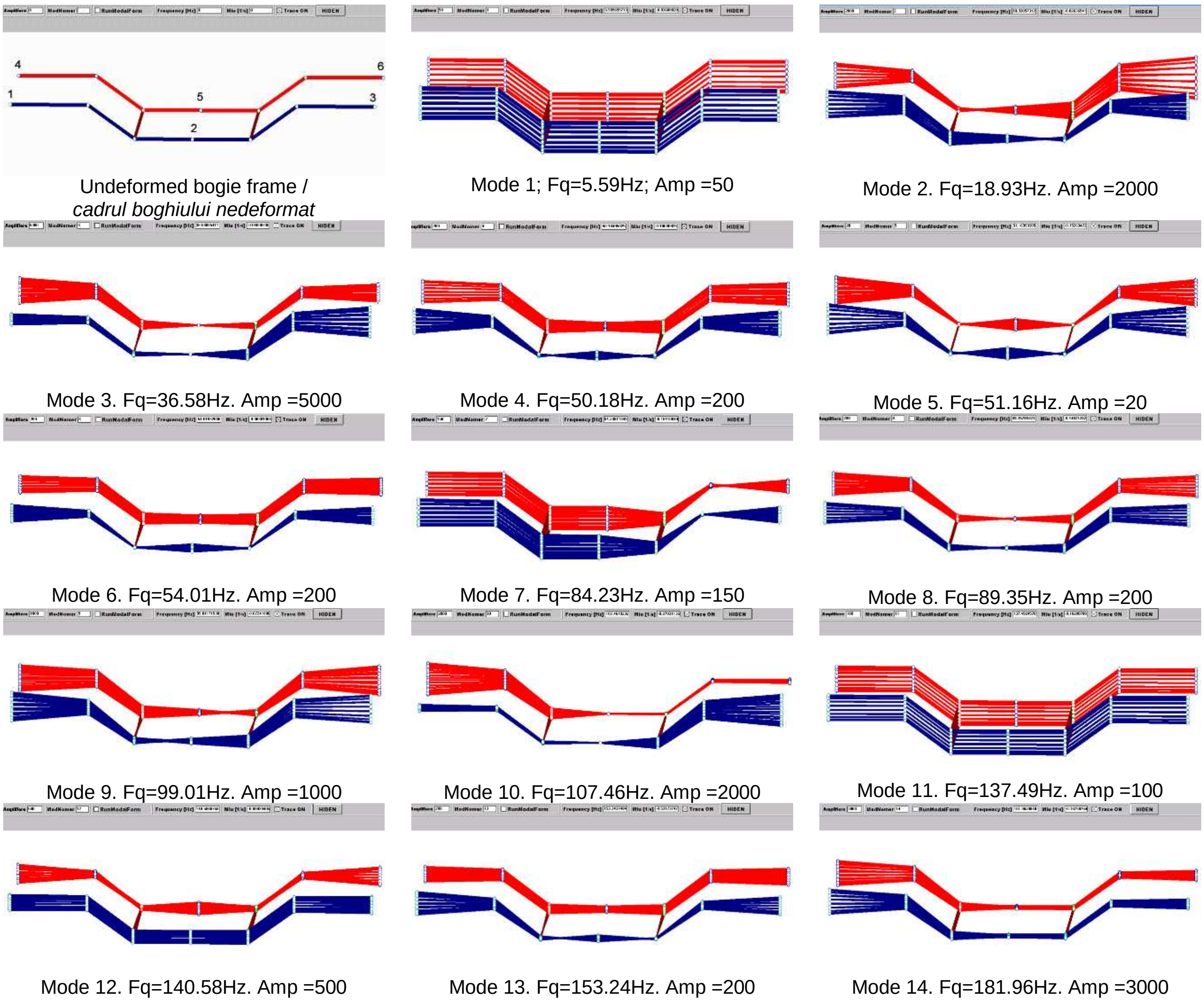


**Fig. 6** – Bogie frame in their first 14 vibration eigenmodes /
*Rama de boghiu in primele 14 moduri proprii de vibratie*

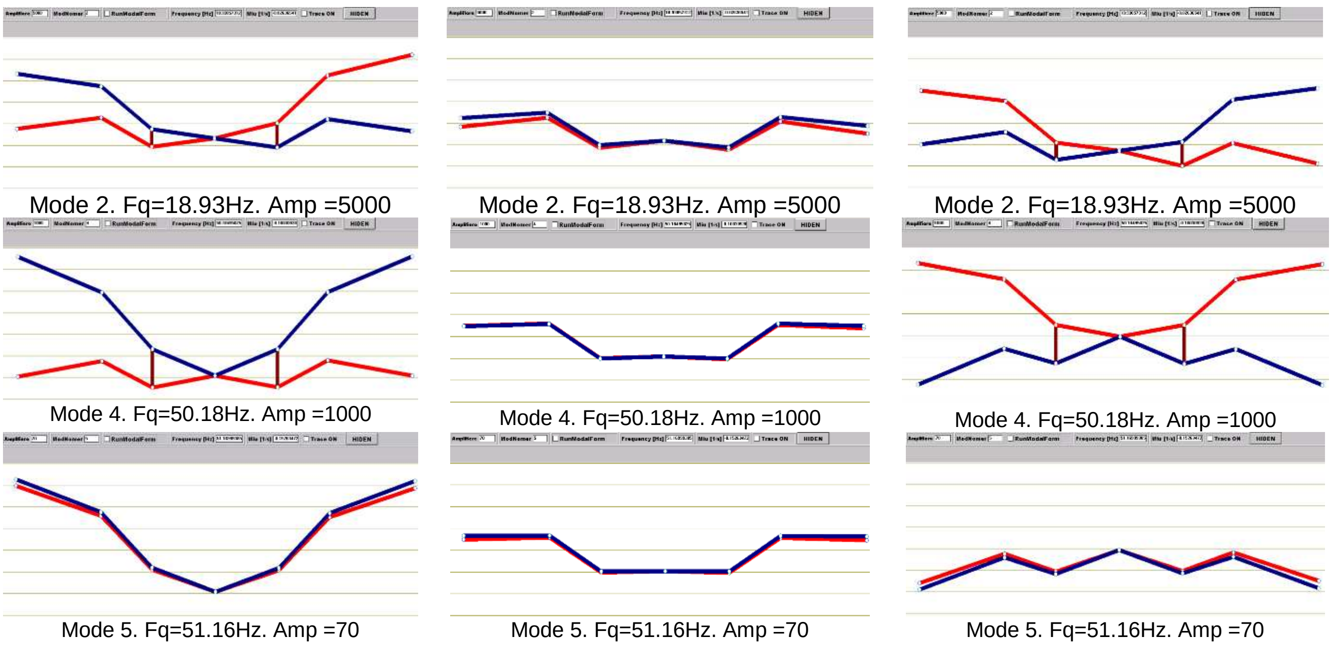


**Fig. 7** – Stages of bogie frame in the 2-nd, 4-th and 5-th vibration eigenmodes /
*Secvente ale ramei de boghiu in modurile proprii de vibratie 2, 4 si 5*

From analysis of representations in Figures 6 and 7 are obtained the following results:

- Mode 1, at the frequency Fq=5.59Hz, is due to the elasticity of cable and crane beam and represents a vertical oscillation of the frame, rigid body oscillation;
- In the mode 2, at the frequency Fq=18.93Hz the oscillation is achieved due to elasticity of the transverse frames, the longerons behaving like rigid;
- In the mode 3, at the frequency Fq=36.58Hz, over the oscillation due to elasticity of the transversal frames overlaps the oscillation of longerons in its own fundamental mode, by emphasizing some small asymmetry of the elasticity of materials of which were manufactured the two longerons;
- In the mode 4, at the frequency Fq=50.18Hz, the longeron marked by points 1,2,3 (blue) oscillates in the fundamental mode of vibration driving the longeron marked by the points 4,5,6 (red) that presents a rigid body motion. It is strongly emphasized the constructive asymmetry of the two longerons, the one marked by the points 1,2,3 (blue) presenting a higher elasticity than the longeron marked by the points 4,5,6 (red);
- In the mode 5, at the frequency Fq=51.16Hz, the longerons oscillate in phase, in the fundamental vibration mode;
- Mode 6 is identical in form with the oscillation mode 4;
- Modes 7 and 8, at frequencies of 84.23Hz si 89.35Hz, are due to elasticity and asimetry of the connecting cables of the frame to the crane hook, representing rigid body oscillations of the the bogie frame. It makes the observation that, at the modal identification test carried out at the beginning of the static tests, clamping the frame in the crane was done using straps, whilst at test performed after 6 millions of fatigue cycles, the clamping was made using steel cables, attached in the same points as the straps;
- Mode 9 is identical in form with mode 2, with changing the line of symmetry of the oscillation;
- Starting with the mode 10 can be found the previous forms of oscillation with changing of oscillations between the longerons, due to the asymmetry of the sheets elasticity of which these are made.

From the above it follows that the modal analysis can be successfully used for:

- constructive concept validation, because the existence of some structure eigenfrequencies in an area where exist external exciter frequencies or due to natural operating conditions, is dangerous;
- validation of manufacturing technology, because it can be highlighted possible inhomogeneities in the distribution of the mechanical characteristics or eventual cracks;
- identification of weak areas of the structure, these being the zones showing high inflections of certain vibration modes.

**Using modal analysis for highlighting the material fatigue phenomenon**

During testing at static loads, the mechanical stresses on the bogie frame structure were monitored, in a total of 44 measurement points resulting from a preliminary finite element analysis.

Periodically at one million cycles, were made measurements of the same mechanical stresses, in order to identify any fatigue, weakening or failure phenomena of the bogie frame structure. It was noted that during the fatigue tests have not manifested growing phenomena of mechanical stresses on the bogie frame structure.

At the end of the 6 million cycles of fatigue were performed dimensional measurements, with a laser station, confirming that the frame structure does not present remanent deformations.

Din analiza reprezentarilor din figurile 6 si 7 se desprind urmatoarele rezultate:

- Modul 1, la frecventa Fq=5.59Hz, este datorat elasticitatii cablului si grinzii macaralei si reprezinta o oscilatie pe verticala a ramei, oscilatie de corp rigid;
- In modul 2, la frecventa Fq=18.93Hz oscilatia se realizeaza datorita elasticitatii cadrelor transversale, lonjeroanele comportandu-se ca rigide;
- In modul 3, la frecventa Fq=36.58Hz, peste oscilatia datorita elasticitatii cadrelor transversale se suprapune oscilatia lonjeroanelor in modul propriu fundamental, cu evidentierea unor mici asimetrii ale elasticitatii materialelor din care au fost confectionale cele doua lonjeroane;
- In modul 4, la frecventa Fq=50.18Hz, lonjeronul marcat de punctele 1,2,3 (albastru) oscileaza in modul fundamental de vibratie antrenand lonjeronul marcat de punctele 4,5,6 (rosu) care prezinta miscare de corp rigid. Este evidentiata puternic nesimetria constructiva a celor doua lonjeroane, cel marcat de punctele 1,2,3 (albastru) prezentand o elasticitate mai mare decat lonjeronul marcat de punctele 4,5,6 (rosu);
- In modul 5, la frecventa Fq=51.16Hz, lonjeroanele oscileaza in faza, in modul fundamental de vibratie;
- Modul 6 este identic ca forma cu modul 4 de oscilatie;
- Modurile 7 si 8, la frecventele de 84.23Hz si 89.35Hz, se datoreaza elasticitatii si nesimetriei cablurilor de legatura ale ramei in carligul macaralei, reprezentand oscilatii de corp rigid ale ramei de boghiu. Se face observatia ca, la testul de identificare modala efectuat la inceperea incercarilor statice si de oboseala, prinderea ramei in macara s-a realizat utilizand chingi, in timp ce la testul efectuat dupa 6 milioane de cicli de oboseala, prinderea s-a realizat utilizand cabluri de otel, prinse in aceleasi puncte ca si chingile;
- Modul 9 este identic cu modul 2, cu schimbarea liniei de simetrie a oscilatiei;
- Incepand cu modul 10 se regasesc formele de oscilatie anterioare cu schimbarea oscilatiilor intre lonjeroane, datorita asimetriei elasticitatii tablelor din care sunt confectionate acestea.

Din cele prezentate rezulta ca analiza modala se poate utiliza cu succes pentru:

- validarea conceptiei constructive, deoarece existenta unor frecvente proprii ale structurii, intr-o zona in care exista frecvente excitatoare externe sau datorate conditiilor naturale de functionare, este periculoasa;
- validarea tehnologiei de fabricatie, deoarece pot fi puse in evidenta eventuale neomogenitati in distributia caracteristicilor mecanice sau eventuale fisuri;
- identificarea zonelor slabe ale structurii, acestea fiind zonele ce prezinta inflexiuni mari la anumite moduri de vibratie.

**Utilizarea analizei modale pentru evidentierea fenomenului de oboseala a materialului**

Pe durata incercarilor la solicitari statice, au fost monitorizate tensiunile mecanice din structura ramei de boghiu, intr-un numar de 44 puncte de masura rezultate dintr-o analiza preliminara cu elemente finite.

Periodic la cate un milion de cicli, au fost efectuate masuratori ale acelorasi tensiuni mecanice, in vederea depistarii eventualelor fenomene de oboseala, slabiri sau cedari ale structurii ramei de boghiu. S-a constata ca pe durata incercarilor la oboseala nu s-au manifestat fenomene de crestere a tensiunilor mecanice pe structura ramei de boghiu.

La sfarsitul celor 6 milioane de cicli la oboseala au fost efectuate masuratori dimensionale, cu o statie laser, confirmandu-se faptul ca structura ramei nu prezinta deformatii remanente.

Also, was performed the ultrasonic control. No cracks were found of welds or of the frame.

As previously mentioned, the bogie frame was subjected to modal identification tests, at the beginning of static tests and after completion of the 6 million cycles of fatigue tests, applied at a frequency of 3.5 Hz.

In the Figure 8 is shown the final panel of the modal identification of bogie frame at the identification test made at the beginning of static tests, for the structure excitation in point P1 and response measurement in the same point, P1.

In the Table 1 are presented the eigenfrequencies of the bogie frame determined at the beginning of the tests and after 6 million cycles of fatigue tests.

Deasemenea, a fost efectuat controlul ultrasonic. Nu au fost depistate fisuri ale sudurilor sau ramei.

Asa cum a fost mentionat anterior, rama de boghiu a fost supusa la teste de identificare modala, la inceputul incercarilor la solicitari statice, precum si dupa finalizarea celor 6 milioane de cicli de solicitari la oboseala, aplicate la o frecventa de de incercare de 3,5 Hz.

In fig.8 este prezentat panelul final al identificarii modale a ramei de boghiu la incercarea efectuata la inceperea incercarilor statice, pentru excitarea structurii in punctul P1 si masurarii raspunsului in acelasi punct, P1.

In Tabelul 1 sunt prezentate frecventele proprii ale ramei de boghiu determinate la inceperea incercarilor si dupa 6 milioane de cicli de incercari la oboseala.

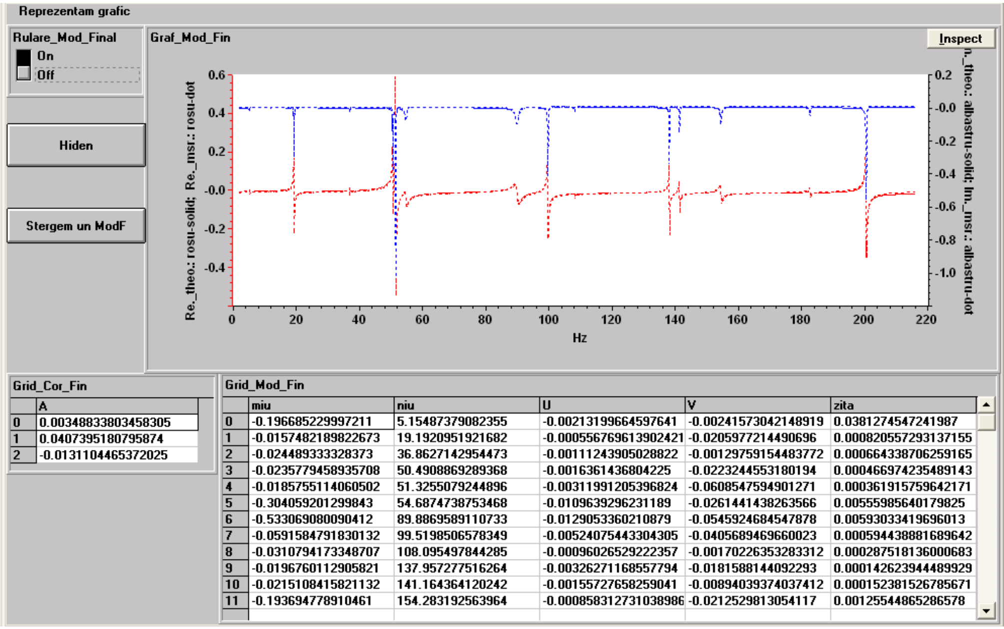


**Fig. 8** – Modal parameter identification for excitation in point P1 and response measuring in point P1 at start of the tests
*Identificarea parametrilor modali pentru cazul excitarii in punctul P1 si masurarii raspunsului in punctul P1 la începerea incercarilor*

**Table 1 / *Tabelul 1***

The bogie frame eigenfrequencies at start of tests and after 6 million of fatigue cycles /
*Frecventele proprii ale ramei de boghiu la inceperea incercarilor si dupa 6 milioane de cicli de incercare la oboseala*

| **Exc.Point / *Punct Excitare*** | **Meas.Point / *Punct Măsură*** | **Mode No. / *Număr Mod*** | **Frequency / *Frecvența* (Hz)** | **Frequency / *Frecvența* (Hz)** |
|---|---|---|---|---|
| | | | **0 million cycles** | **6 million cycles** |
| 1 | 1 | 1 | 5.154874 / *5,154874* | 5.60202 / *5,60202* |
| 1 | 1 | 2 | 19.1921 / *19,1921* | 18.93807 / *18,93807* |
| 1 | 1 | 3 | 36.86271 / *36,86271* | 36.59926 / *36,59926* |
| 1 | 1 | 4 | 50.49089 / *50,49089* | 50.20117 / *50,20117* |
| 1 | 1 | 5 | 51.32551 / *51,32551* | 51.19232 / *51,19232* |
| 1 | 1 | 6 | 54.68747 / *54,68747* | 54.02111 / *54,02111* |
| 1 | 1 | 7 | - | 83.76763 / *83,76763* |
| 1 | 1 | 8 | 89.88696 / *89,88696* | 89.33374 / *89,33374* |
| 1 | 1 | 9 | 99.51985 / *99,51985* | 98.9242 / *98,9242* |
| 1 | 1 | 10 | 108.0955 / *108,0955* | 107.7028 / *107,7028* |
| 1 | 1 | 11 | 137.9573 / *137,9573* | 137.4873 / *137,4873* |
| 1 | 1 | 12 | 141.1644 / *141,1644* | 140.5881 / *140,5881* |
| 1 | 1 | 13 | 154.2832 / *154,2832* | 153.3011 / *153,3011* |
| 1 | 1 | 14 | 182.4826 / *182,4826* | 181.9938 / *181,9938* |
| 1 | 1 | 15 | 200.2565 / *200,2565* | 199.7497 / *199,7497* |

From eigenfrequencies analysis determined at the beginning of the tests and after 6 million of cycles it is found that:

Din analiza frecventele proprii determinate la inceperea incercarilor si dupa cele 6 milioane de cicli se constata ca:

- At the start of the test does not appear the own frequency of 83.76Hz. The explanation is in fact that at the two tests the attachment of frame in the crane hook was different, by belt straps at the beginning and steel cables at the end. The modes from the frequencies of 84.23Hz and 89.35Hz represent rigid body oscillations of the bogie frame;
- Generally it has not found a significant difference between the eigenfrequencies determined at the start of the tests and after the 6 million of cycles.

Following the tests was found that:

- the analysis of mechanical stresses, dimensional measurements and ultrasound checking did not show the revealed structure modifications of bogie frame;
- the modal analysis applied at start of the static tests and after the 6 million cycles of fatigue loads did not reveal the notable movement of eigenfrequencies or the appearance of additional eigenmodes.

It is necessary to continue the application of modal identification tests after performing the 2 million cycles with loads amplified with factors of 1.2, respectively 2 millions of cycles with loads amplified by factors of 1.4, compared with nominal loads at fatigue.

## CONCLUSIONS

1. The experimental modal analysis can be successfully used to validate the structural conception and structural integrity of complex expensive mechanical structures, which may present high risks in operation.
2. Experimental modal analysis can be successfully used to detect early fatigue phenomena of the material.
3. In order to obtain conclusive results on the use of experimental modal analysis, for earlier detection of material fatigue phenomenon, it is necessary to continue the tests of modal identification after performing the 2 millions of cycles with loads amplified by 1.2, respectively 2 millions of cycles with loads amplified by 1.4, comparatively with nominal fatigue loads, according to European standard EN 13749.

**Acknowledgement:** The work is performed under the contract PN II-PCCA, no. 192/2012.

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- La inceperea incercarilor nu apare frecventa proprie de 83.76Hz. Explicatia consta in faptul ca la cele doua incercari prinderea ramei in carligul macaralei a fost diferita, prin chingi la inceput si cabluri de otel, la final. Modurile de la frecventele de 84,23Hz si 89,35Hz reprezinta oscilatii de corp rigid ale ramei de boghiu;
- In general nu se constata o diferenta notabila intre frecventele proprii determinate la inceperea incercarilor si dupa cele 6 milioane de cicli.

In urma incercarilor s-a constatat ca:

- analiza tensiunilor mecanice, masuratorile dimensionale si verificarea ultrasonica nu au evidentiat modificari ale structurii ramei de boghiu;
- analiza modala aplicata la inceperea incercarilor statice si dupa cele 6 milioane de cicli de solicitari la oboseala nu a evidentiat deplasarea notabila a frecventelor proprii sau aparitia unor moduri proprii suplimentare.

Se impune necesitatea continuarii aplicarii incercarilor de identificare si dupa efectuarea celor 2 milioane de cicli cu sarcini amplificate cu factori de 1,2, respectiv 2 milioane de cicli cu sarcini amplificate cu factori de 1,4, comparativ cu sarcinile nominale la oboseala.

## CONCLUZII

1. Analiza modala experimentala poate fi utilizata cu succes pentru validarea conceptiei constructive si a integritatii structurale a structurilor mecanice complexe, costisitoare sau care pot prezenta risc mare in functionare.
2. Analiza modala experimentala poate fi utilizata pentru depistarea din timp a fenomenului de oboseala a materialului.
3. In vederea obtinerii unor rezultate concludente privind utilizarea analizei modale experimentale, pentru depistarea din timp a fenomenului de oboseala a materialului, este necesara continuarea aplicarii incercarilor de identificare modala si dupa efectuarea celor 2 milioane de cicli cu sarcini amplificate cu factori de 1,2, respectiv 2 milioane de cicli cu sarcini amplificate cu factori de 1,4, comparativ cu sarcinile nominale la oboseala, conform standardului european EN 13749.

**Precizare**: Lucrarea este realizata in cadrul contractului PN II-PCCA, nr. 192/2012.

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