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.

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%.

Resumat: În lucrare sunt prezentate cercetări teoretice și experimentale privind utilizarea analizei modale experimentale ca instrument de apreciere a stării de oboseală, cat și pentru identificarea modificărilor 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. Cercetările sunt in desfasurare, in articol fiind prezentate rezultatele privind incercările initiale, precum si cele dupa 6 milioane de cicluri de solicitari la oboseala.

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

INTRODUCERE

Boghiurile sunt echipamente complexe cu rol vital in functionarea vehiculelor feroviere, avand rolul de portutor al carcasei, de asigurare a forțelor de tractiune si de franare, precum si de izolare vibratoare a carcasei si a incarcaturii transportate. Avand in vedere rolul important in securitatea materialului rulant, factul ca toate solicitariile de la calea de rulare se transmit la carcașa 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 solicitariile la care este supus boghiul sa reproduca cat mai fidel solicitariile din functionarea normala, fara a introduce constrangeri sau grade de libertate suplimentare sau nerealiste.

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

în practica sunt normate doua cazuri de incarcare statica, care se considera fi a accep or icaii pentru situațiile reale intâlnite pe durata de de viata a echipamentului:

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

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

- 6 milioane cicli cu sarcinile de exploatare;
- 2 milioane cicli cu sarcinile de exploatare crescuta cu 20%.
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 \( m_i \), connected by stiffness elastic elements \( k_{ij} \) and damping elements of \( c_{ij} \) constant. For this damped system subjected to the action of an external excitations system \( \{Q(t)\} \), the equations of movement are:

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

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

\[
[X(\omega)] = \sum_{i=1}^{n} \left\{ \psi^{i}(\omega) \cdot \frac{\psi^{i}(\omega)}{\mu} \cdot [Q(\omega)] \cdot \frac{\psi^{i}(\omega)}{\mu} \cdot [Q(\omega)] \cdot \frac{\psi^{i}(\omega)}{\mu} \cdot [Q(\omega)] \right\}
\]

where:
- \( \{\psi^{i}(\omega)\} \) - its own vector of the order "k";
- \( \mu_k \) - the "k" order damping ratio;
- \( v_k \) - the "k" order damped natural frequency;
- \( a_k \) and \( \alpha_k \) - 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 excitation of the successively \( j \) \((j=1,2,...m)\) and simultaneous determination of the response (in accelerations or displacements) in points \( j \) \((j=1,2,...n)\). In practical applications, the eigenvectors are substituted with two modal constants \( U_{ij} \) and \( V_{ij} \) defined by following:

\[
\psi^{i}(\omega) = \frac{U_{ij} + i \cdot V_{ij}}{\alpha_k} \quad \text{unde:}
\]
- \( \{\psi^{i}(\omega)\} \) - vectorul propriu de ordinul "k";
- \( \mu_k \) - rata de amortizare de ordinul "k";
- \( v_k \) - frecventa naturala amortizata de ordinul "k";
- \( a_k \) si \( \alpha_k \) - constante de normare;
- \( \omega \) - frecventa excitatiei externe.

In vederea determinarii vectoriilor proprii din date experimentale, in articol se prezinta metoda de a excita structurii intr-un singur punct. Aplicarea metodelor consta in a efectua succesiv a structurii in punctele \( j \) \((j=1,2,...m)\) si a determinarii simultane a raspunsului (in accelerati sau deplasari) in punctele \( i \) \((i=1,2,...n)\). In aplicatii practice vectorii proprii sunt inlocuiti cu doua constante modale \( U_{ij} \) si \( V_{ij} \) definite prin relatia:

\[
\psi^{i}(\omega) = \frac{U_{ij} + i \cdot V_{ij}}{\alpha_k}
\]

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 written as:

\[
\alpha(\omega) = \sum_{i=1}^{n} \frac{U_{ij} + i \cdot V_{ij}}{\mu_k + i \cdot (\omega - v_k)} + \sum_{j=1}^{m} \frac{U_{ij} - i \cdot V_{ij}}{\mu_k + i \cdot (\omega + v_k)}
\]
The last relation, with $i=1,2,...,\infty$, $j=1,2,...,n$, 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” $\frac{-1}{M_{ij} \omega^2}$, for lower modes, and “residual flexibility” $S'_{ij}$, for upper modes.

The system admittance will have the expression:

$$a_i(a) = \frac{-1}{M_{ij} \omega^2} \sum_{j=1}^{\infty} \left( \frac{U_i' + i V_i}{\mu_i + i(\omega - \nu_i)} - \frac{U_i' - i V_i}{\mu_i + i(\omega + \nu_i)} \right) + S_{ij}$$

An eigenmode is defined by a set of modal parameters, which are intrinsic characteristics of the system, independent of the external conditions: $[\omega_{ij}, \mu_{ij}, V_{ij}, a_{ij}]$, $k=1,2,...,J$, or by a combination of the modal parameters and modal constants, which depends on the external conditions:

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

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.

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.

Ultima relatie, cu $i=1,2,...,\infty$, $j=1,2,...,m$, defineste setul functiilor de rapsun 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 considerinte legate de tehnici de excitare si de masurare a rapsunului, cat si din considerinte legate de tehnica de calcul utilizata si de timpul necesar pentru prelucrari. In aplicatiile domeniului frecventelor de rapsun 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’.

Contribution to modular analysis and superimposed to include in the factors of correlation denoted “admittance modala inferioara” $\frac{-1}{M_{ij} \omega^2}$ pentru modurile inferioare, respective “flexibilitate reziduala”, $S'_{ij}$ pentru modurile superioare.

Admitanta sistemului va avea expresia:

$$a_i(a) = \frac{-1}{M_{ij} \omega^2} \sum_{j=1}^{\infty} \left( \frac{U_i' + i V_i}{\mu_i + i(\omega - \nu_i)} - \frac{U_i' - i V_i}{\mu_i + i(\omega + \nu_i)} \right) + S_{ij}$$

Un mod proprie este definit printr-un set de parametri modali, care sunt caracteristici intrinseci ale sistemului, independente de conditiile externe: $[\omega_{ij}, \mu_{ij}, V_{ij}, a_{ij}]$, sau printr-o combinatie de parametri modali si de constante modale, dependent de conditiile externe de excitare.

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 rapsunului. 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.

Pachet de programe pentru analiza modala

Pe baza celor prezente 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 mediuul de programare TestPoint, are o conceptie modulara, cuprinzand urmatoarele programe:

- **ModalAch**, este un modul pentru controlul achizitiiei datelor reprezentand excitatia si rapsunul sistemului.
- **IdModal**, este un modul pentru calculul functiilor de rapsun in frecventa si a parametrilor modali. Funcțiile de rapsun 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 continue in fisierile realizate cu programul ‘IdModal’. Descriserea programelor va fi realizata pe masura prezentarii experientarilor 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 inputs: 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.

![Fig. 1 - Measuring equipment „Softronic Data Acquisition System”](image)

Experiments were performed in the Laboratory of Dynamic Testing of INMA Bucharest, bogie frame being suspended in crane hook by four inextensible 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.
Modal identification - The paper presents only the tests result carried out on the vertical direction. For each of the excitation points are successively 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 enrols 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 fortele 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 nelineara. Daca modurile de vibratie sunt multiple, iar unele sunt apropiate ca frecventa, se procedeaza la o identificare parțială, pe grupe de moduri apropiate. Pe masura identificarii, modurile sunt stocate iar in etapa finala se procedeaza la identificarea prin regresie nelineara a tuturor modurilor din domeniul de frecventa de interes. In fig.5 este prezentat un panel de identificare modala parțiala si panelul final de identificare, pe toate cele 15 moduri de vibratie de interes. In ambele panele sunt prezentate suprapus trasele determinate teoretic (linie continua) si trasele determinate...
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, evidențiaza 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 fisierul contine 540 linii.

**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.
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 $F_q=5.59\text{Hz}$, 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 $F_q=16.93\text{Hz}$ the oscillation is achieved due to elasticity of the transverse frames, the longerons behaving like rigid;
- In the mode 3, at the frequency $F_q=36.58\text{Hz}$, 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 $F_q=50.18\text{Hz}$, 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 $F_q=51.16\text{Hz}$, 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 $84.23\text{Hz}$ si $89.35\text{Hz}$, are due to elasticity and asymmetry 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.

Periodic at every 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.

In the analysis the results of fatigue tests 6 and 7 were desprind urmatoarele rezultate:

- Modul 1, la frecventa $F_q=5.59\text{Hz}$, este datorat elasticității cablului si grinzii macarei si reprezinta o oscilatie pe verticala a ramei, oscilatie de corp rigid;
- In modul 2, la frecventa $F_q=16.93\text{Hz}$ oscilatia se realizeaza datorita elasticitatii cadrelor transversale, lonjeroanele comportandu-se ca rigide;
- In modul 3, la frecventa $F_q=36.58\text{Hz}$, este o oscillatie a datorita elasticitatii cadrelor transversale se suprapune oscilatia lonjeroanelor in modul propriu fundamental, cu evidentierea unor mici asimetrii ale elasticitatii materialelor din care au fost confectionate cele doua lonjeroane;
- In modul 4, la frecventa $F_q=50.18\text{Hz}$, lonjeronul marcat de punctele 1,2,3 (albastru) oscilaeaza in modul fundamental de vibrazie antrenandlonjeronul marcat de punctele 4,5,6 (rosu) care prezinta miscare de corp rigid. Este evidentatuternic nesimetria constructiva a celor doua lonjeroane, cel marcat de punctele 1,2,3 (albastru) prezintand o elasticitate mai mare decat lonjeronul marcat de punctele 4,5,6 (rosu);
- In modul 5, la frecventa $F_q=51.16\text{Hz}$, lonjeroanele oscilaeaza in faza, in modul fundamental de vibrazie;
- Modul 6 este identic ca forma cu modul 4 de oscilatie;
- Modurile 7 si 8, la frecventele de $84.23\text{Hz}$ si $89.35\text{Hz}$, se datoreaza elasticitatii si nesimetriei cablurilor de legatura ale ramei in carligul macarei, reprezentand oscilatii de corp rigid ale ramei de boghiu. Se face observatia ca, la testul de identificare modala efectuat la inceputa incercarilor statice si de oboseala, prinderea ramei in macara s-a realizat utilizand chinghi, in timp ce la testul efectuat dupa 6 milioane de cicli de oboseala, prinderea s-a realizat utilizand cabluri de otel, prinse in acelesi puncte ca si chingile;
- Modul 9 este identic cu modul 2, cu schimbarile liniei de simetrie a oscilatiei;
- Incepand cu modul 10 se regasesc formele de oscilatie anterioare cu schimbarile oscilatilor intre lonjeroane, datorita asimetriei elasticitatii tablilor 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 proprie ale structurii, intr-o zona in care exista frecvente excatatoro externe sau datorate conditiilor naturale de functionare, este periculoasa;
- validarea tehnologiei de fabricatie, deoarece pot fi puse in evidenta eventuale neomogininti in distributia caracteristicilor mecanice sau eventuale fisuri;
- identificarea zonelor slabe ale structurii, acestea fiind zonele ce prezinta inflexiuni mari la anumite moduri de vibrazie.

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 sau se au manifestat fenomene de cretare a tensiunilor mecanice pe structura ramei de boghiu. La sfarsitul celor 6 milioane de cicli la oboseala au fost efectuate sau se au manifestat fenomene de cretare a tensiunilor mecanice pe structura ramei de boghiu.
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 inceperea 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 inceputul incercarilor statice pentru excitarea structurii in punctul P1 si masurarii raspunsului in acelasi punct, P1.

In Tabelul 1 sunt prezentate frecventele propiilor 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](image)

Identificarea parametrilor modali pentru cazul excitarii in punctul P1 si masurarii raspunsului in punctul P1 la începerea incercarilor

<table>
<thead>
<tr>
<th>Exc.Point / Punct Excitare</th>
<th>Meas.Point / Punct Măsură</th>
<th>Mode No. / Număr Mod</th>
<th>Frequency / Frecvența (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>0 million cycles 6 million cycles</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>1</td>
<td>5.154874 / 5.154874</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>2</td>
<td>19.1921 / 19.1921</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>3</td>
<td>36.86271 / 36.86271</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>4</td>
<td>50.49089 / 50.49089</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>5</td>
<td>51.32551 / 51.32551</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>6</td>
<td>54.68747 / 54.68747</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>7</td>
<td>-</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>8</td>
<td>89.86969 / 89.86969</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>9</td>
<td>99.51985 / 99.51985</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>10</td>
<td>108.0955 / 108.0955</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>11</td>
<td>137.9573 / 137.9573</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>12</td>
<td>141.6464 / 141.6464</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>13</td>
<td>154.2832 / 154.2832</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>14</td>
<td>182.4826 / 182.4826</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>15</td>
<td>200.2565 / 200.2565</td>
</tr>
</tbody>
</table>

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

Din analiza frecventelor propiilor din incercarea la oboseala la inceputul incercarilor si dupa 6 milioane de ciclo 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 loads at fatigue.

REFERENCES
[1]. Larry N. (1993) - Fourier analysis for beginners - Indiana University;
[6]. EN 13749 - Railway applications - Wheelsets and bogies - Method of specifying the structural requirements of bogie frames.

La începerea incercărilor nu apare frecvența proprie de 83.76Hz. Explicația consta în faptul că la cele două incercări prinderea ramei în carligul macarului a fost diferită, prin chingi la început și cabluri de otel, la final. Modurile de la frecventele de 84,23Hz și 89,35Hz reprezintă oscilații de corp rigid ale ramei de bogiu;

- În general nu se constata o diferență notabilă între frecventele proprii determinate la începerea incercărilor și după cele 6 milioane de cicli.
- Se impune necesitatea continuării aplicării incercărilor de identificare și după efectuarea celor 2 milioane de cicli cu sarcini amplificate cu factorii de 1,2, respectiv 2 milioane de cicli cu sarcini amplificate cu factorii de 1,4, comparativ cu sarcinile nominale la oboseala.

CONCLUZII
1. Analiza modala experimentală poate fi utilizată cu succes pentru validarea conceptiei constructive și a integrității structurale a structurilor mecanice complexe, costisitoare sau care pot prezenta risc mare în funcționare.
2. Analiza modala experimentală poate fi utilizată pentru depistarea din timp a fenomenului de oboseala a materialului.
3. În vederea obținerii unor rezultate concluzionante privind utilizarea analizei modale experimentale, pentru depistarea din timp a fenomenului de oboseala a materialului, este necesară continuarea aplicării incercărilor de identificare modală și după efectuarea celor 2 milioane de cicli cu sarcini amplificate cu factorii de 1,2, respectiv 2 milioane de cicli cu sarcini amplificate cu factorii de 1,4, comparativ cu sarcinile nominale la oboseala, conform standardului european EN 13749.

Precizare: Lucrarea este realizată în cadrul contractului PN II-PCCA, nr. 192/2012.