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RESEARCH OF DYNAMIC CHARACTERISTICS OF DOUBLE-BOOM CRANE

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Abstract

Double Boom Level-Luffing mechanism (with joint connected parts) is typical cranes structure at shipyard in the world. Cranes using these systems are characterized by heights over 80 meters and their weight often exceeding 500 tons. The importance of such large cranes imposes a need for detailed study of their dynamic behavior. This paper provides an overview of experimental and theoretical research of the reach changing system. At the same time, the main tasks for the engineers, analysts, researchers and manufacturers are indicated. The paper shows the results of existing cranes experimental tests (its real dynamic properties). The paper also shows the most suitable way of theoretical modeling of supporting structure and calculations of internal dynamic forces. The obtained results in the paper indicate a high level of internal forces and energy and thus the importance of a reliable system design for the reach change.

Key words: *Double-boom crane, Shipyard crane, Transient dynamic analysis, Luffing mechanism, FEM.*

1 INTRODUCTION

The portal rotating-cranes at ports and shipyards have had their specific forms characterized by large structures, long booms and high rotary columns. Their systems of the reach change are made with two booms and in the theory they are known as four-bar linkage mechanism[1]. Overall dimensions of such systems for reach changing allow manipulation of bulky loads and serve large areas of ports and shipyards. The increasing size of the ships built in the last thirty years has caused the demand for even larger cranes. About extensive experimental research on cranes with two booms were writing Eiler P. [1], Pielorz J. [2], M. Jovanovic [3]. Picturesque evolution of technical systems for reach changing, by D.Weinreich, can be seen on the Internet, lit [10]. The essence of the design of the system for reach changing with two booms (four-bar linkage mechanism) is in a special design that lead cargo by almost horizontal path. In this way less energy is spent on changing the cargo position, so the principal technological operation when working with cargo (reach changes) is economical (optimal). Specific high form of the structure for reach changing is elastic and lightweight. Therefore, the dynamics (vibration) can be observed in their work. Also, elastic deflections of long members can be seen.

It is therefore essential not to treat these cranes like rigid mechanisms. Consequently, it is necessary to apply modern methods of nonlinear analysis. Nonlinearity is not only geometric nature of steel construction elasticity, it is also conditioned by other phenomena caused by rheological changes of soil or coast at the ports and the crane's paths that are incorrectly sinking. Further in the text, first are presented some of the experimental researches carried out.

2 EXPERIMENTAL RESEARCH

Experimental research of cranes with two booms were carried as part of research projects in Serbia. Also studies have been done for the development of the domestic manufacturers MIN FITIP Niš (Serbia). The aim of the experimental testing of shipyard cranes Pula-2 (Uljanik - Pula, Croatia 1987) is the verification of its dynamic model [3]. The starting (measured) parameters allow a variety of further analysis, particularly the dynamic simulation of incidental events [12]. In the experiment, following technical tasks were observed which at the same time are objectives of scientific and technical development of cranes:

- Inspection of elastic characteristics of the portal, and of the structure for changing the reach is carried out in order to verify the starting properties of the model (the exact geometry and weight distribution).
- Checking the accuracy of rocker top trajectory which takes in cargo reach change.
- Checking uniformity distribution of crane weight on crane port path.
- Checking of achieved electric drive characteristics: power (electricity), engine torque, drive force and rotation speed.
- Checking the optimality of system for changing the reach balancing with loads and weights.
- Checking the eigenvalues (frequencies).

[1] The theoretical basis of cranes with horizontal guiding of cargo based on four-bar linkage mechanism was given by P.L.Čebišev 1889. The first crane with four-bar linkage mechanism was made by an English company Babcock & Wilcox, in 1913 for the purposes of the London's port. Operating efficiency of the system for the horizontal guidance of cargo influenced the general application of this technical concept as the standard of port and shipyard cranes. In Germany, a specific term was coined for this technical solution - WIPP system *(Niemann: "UBER WIPPKRANE" - Werft Rederei-Hafen No.l4, 22)*. The basis of the system for changing the reach is given by [7].

- Checking of the dynamic coefficients in work of a various drive mechanisms (lift, reach changes, movement).
- Checking the general dynamic stability of crane as high moving object (machines).
- Assessment of adaptation of the structure to the chosen kinematic characteristics.
- Ergonomic adaptation of crane purpose (shipbuilding): visibility and maneuverability.

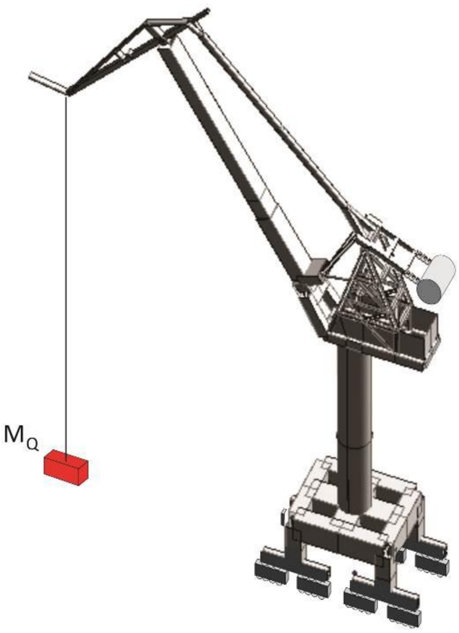


Fig 1. *Portal-rotating crane MIN Pula 2 (1987)*

The quality of crane modeling is verified through several criteria:

2.1 Checking the accuracy of paths

Trajectory of rocker top can be experimentally controlled by geodetic leveling method (optical method). The result of such measurement is shown in Figure 2. The upper (red) trajectory curve shows the theoretical model of the rigid four-bar linkage mechanism (boom), and a lower curve (blue) shows a trajectory determined by experiment [3]. The biggest determined difference of elastic displacement (deflection) of numerical and experimental research are max. 10.2% in the whole range of the reach changes, [3-1990.].

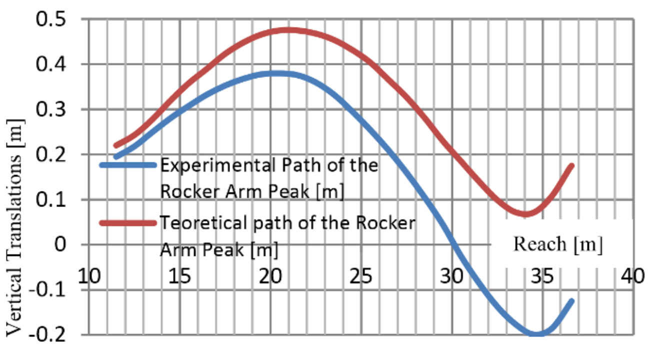


Fig 2. *Trajectory of cargo: Theoretical solution (Rigid Mechanical model) and experimental solution*

2.2 Checking elasticity of the model

The basis of the development of flexible models require prior knowledge of the elastic line. Individual elastic deformation is analyticaly determined by using the Castigliani theorem or the finite element method. By applying the Castigliano theorem, elastic deformation is determined as the sum of integrals of the deformation at the m-characteristic section of support structure including supports. Determination of the elastic deformation by this method as a sum integrals of the bending I_{XF}, I_{YF}, integral of the axial forces I_{XA}, I_{YA} and the integral shear force I_{XS}, I_{YS}, are defined by the relations 1. Summing individual deformation from bending (denoted F index), axial forces (indicated index A) and shearing (indicated index S) in the x and y direction, deformation of the tip of the boom is obtained (which define the elastic line).

$$
\begin{aligned}
\Delta x_F &= \sum_{i=1}^{m} \frac{1}{B_i} \int_0^L M_i \frac{\partial M_i}{\partial F} dz = \sum_{i=1}^{m} I_{XF}, \\
\Delta y_F &= \sum_{i=1}^{m} \frac{1}{B_i} \int_0^L M_i \frac{\partial M_i}{\partial Q} dz = \sum_{i=1}^{m} I_{YF}, \\
\Delta x_A &= \sum_{i=1}^{m} \frac{\partial}{\partial F} \int_0^L \frac{F_{A_i}^2}{2EA_i} dz = \sum_{i=1}^{m} I_{XA}, \\
\Delta y_A &= \sum_{i=1}^{m} \frac{\partial}{\partial Q} \int_0^L \frac{F_{A_i}^2}{2EA_i} dz = \sum_{i=1}^{m} I_{YA}, \\
\Delta x_S &= \sum_{i=1}^{m} \frac{\partial}{\partial F} \cdot \frac{\chi}{2GA_i} \int_0^L F_{Ri}^2 dz = \sum_{i=1}^{m} I_{XS}, \\
\Delta y_S &= \sum_{i=1}^{m} \frac{\partial}{\partial Q} \cdot \frac{\chi}{2GA_i} \int_0^L F_{A_i}^2 dz = \sum_{i=1}^{m} I_{YS},
\end{aligned}
\tag{1}
$$

In these relations F and Q are fictional force, M_i are the bending moments. **Bi** are bending stiffness. χ is Von Karman's coefficient, F_{Ri} - radial forces, F_{Ai} - axial force in the respective sections of the structure. **G** is the shear modulus, A_i-area of members section.

Verification of the elastic properties: Analytical and numerical determined elastic displacements of the model (1) are checked by comparing them with the experimental one. First, they determine the elastic properties of the crane supports on the coast. Then, the elastic properties of the crane supports on the coast are determined. Crane relies over the system (group) of driving trolley on track. The track is built on piles (inserted into the coast) and the reinforced concrete structure of the base. Stiffness of supports of the portal crane is determined from the track stiffness and the stiffness of the driving trolley of portal. The average stiffness of the paths is obtained by measurement of the track deflection by applying known force. From the force-deflection ratio the rigidity of the track is determined (c_{sr}^2=1620000. kN/m). Figure 8 shows the measured deflection of rails under the legs of the portal (wheels) MIN Pula-2 crane in the regime of reach changing.

The stiffness of the driving trolley can be determined by FEM modeling of assembly of pair driving trolleys (balancers). Calculated deflection of all balancers (observed in object^2) is f_{BAL} = 0.00028 m, during the effect of unit force F_1 = 1000 kN. Then the rigidity of the balancers (driving system of the portal) is the ratio of the the unit force and the calculated deflection: c_{BAL} = 3,571,000 (kN/m).

Now it is possible to determine the stiffness of the support of the portal crane, by summing the stiffness of the track and the stiffness of balancer trolleys (series elastic spring connection). Thus (theoretical and experimentally) determined overall stiffness (c* = 1,116,000 kN/m), is further used in the FEM modeling for the determination of elastic displacement in the static and dynamic analyses.

2.3 Control of power machines for crane luffing

The research specifically search the area of high dynamic forces of drive for crane reach changing. Drives for reach change are mainly carried out as threaded spindle or rack and they are exposed to extreme pressure forces. Pressure forces are the consequences of dynamic phenomena when working with cargo. Knowing the greatest axial forces in drives for the reach changing enables control of stability of the main cinematic elements (spindle).

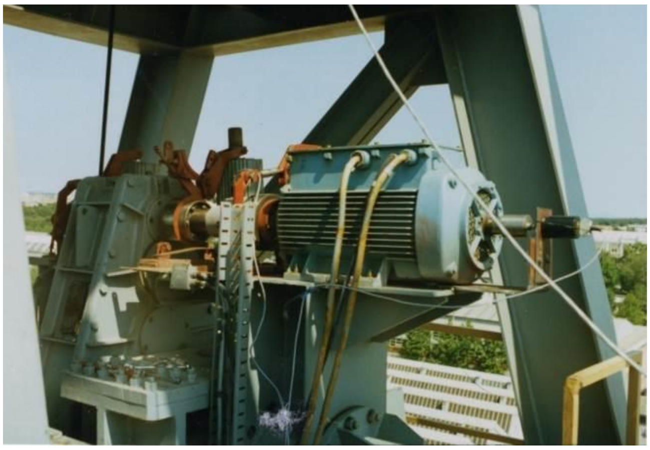


Fig. 3 *Measuring installation for angular speed, power and torque measuring of an asynchronous motors (spindle)*

Mode: Q =0 t, The decrease of the reach (36.75 - 17.40 m) in 1988. PULA-2, File: NVPOP3 (Measurement No.4)

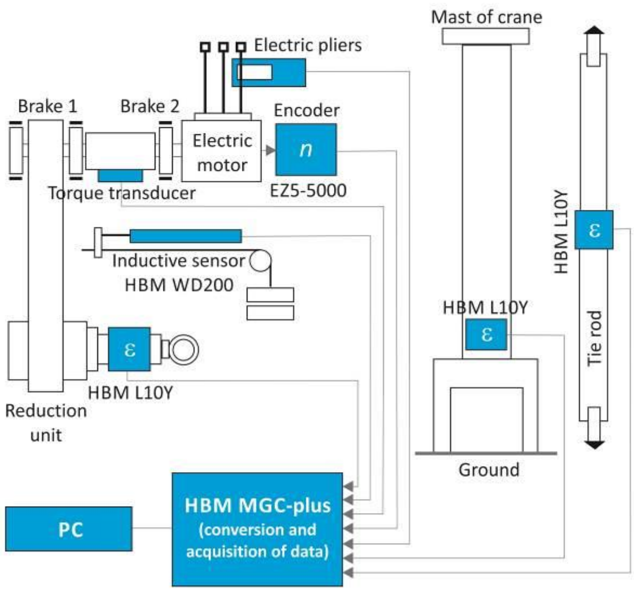


Fig. 4 *Plan of measuring experimental equipment*

Figure 3 shows a photograph of a part of the of drive for reach changing with installed sensors of current, torque, angular velocity of the engine and in particular the force on the drive member (not visible). Figure 4 shows the scheme of measuring equipment layout for experiment carrying out with the details of the sensor, encoder and cabling measuring devices. Figure 5 shows measurement results of mechanical and electrical parameters of the drives for reach changing in the unloaded operating mode (3). Figure 6 shows the results of driving machine measurement while working with a load of 12 t. Operating modes of up to 160 s were observed. The red curve is engine torque, the yellow curve is a curve of engine speed, green curve is the current of motor rotor, black curve is the curve of crane reach, lilac curve is the curve of crane tower deformation.

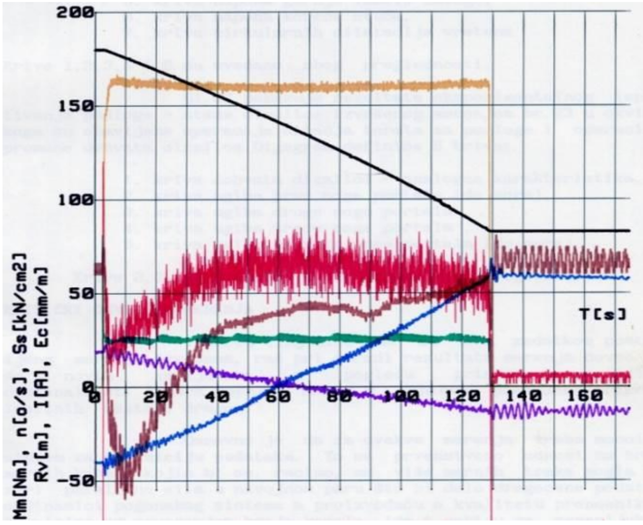
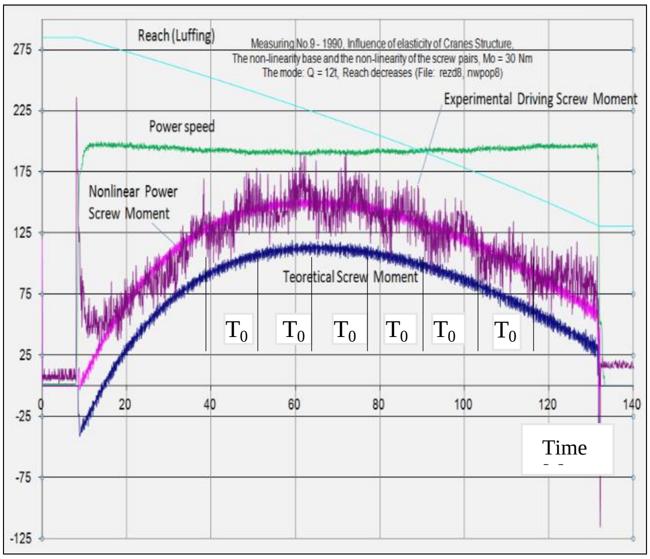


Fig. 5 *The results of mechanical and electrical characteristics measurements*



Legend:
- Motor moment of Elastic Structure (analitical) x3 [Nm]
- Motor moment of rigid Structure (analitical) x3 [Nm]
- Number of Motor revolution per second x 12 [o/s]
- Luffing of Crane Four-Bar Mechanism x 8 [m]
- Experimental Motor Moment x 3 [Nm]

Fig. 6 *Measuring No.9, [3] Influence of elasticity of Cranes Structure, The non-linearity base and the non-linearity of the screw pairs Mo= 30 Nm, The mode: Q = 12t, Reach decreases (File: rezd8, nwpop8)*

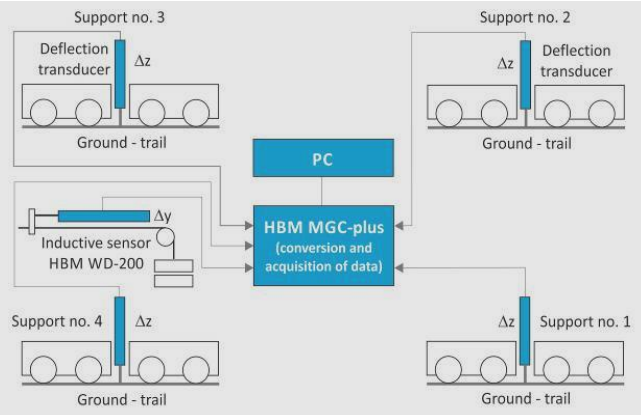


Fig 7. *Layout of measuring equipment for the deflection of crane tracks measurement*

Figure 7 shows a diagram of equipment for the deflection paths measurement. Transducers of tracks deflection (Deflection transducers, number 1,2,3,4) are inductive and they are mounted on the beams 6 m long. Beams - transducer carriers are attached on the ends (perpendicular on the track direction) to the ground. Thus was measured local rail deflection and ignored the deflection of paths (coast). In Figure 8 are shown the results obtained by the deflection caused by the first lifting of test cargo and than with reduction of the reach to a minimum (Rmax ⟶ Rmin). Experimental measurements also discover disadvantages, so it was noted that one group of wheels is not in contact with the rails (in the same position on the track) and does not cause deflection of paths. Therefore, its pair (second group of portal supports in the same direction of the track) has a much higher (summary) deflection. The lack of contact was determined by manually turning the wheels.

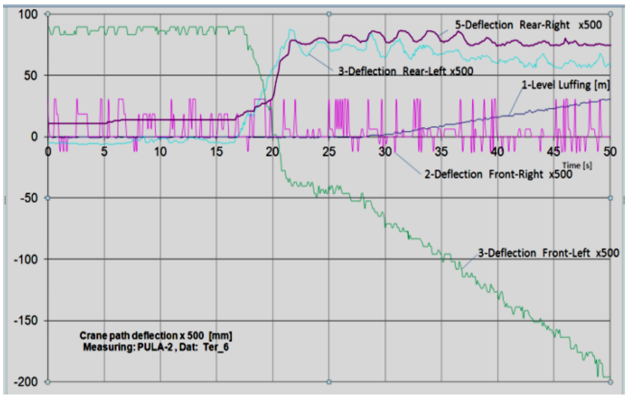


Fig 8. *Crane path (trail) deflections x 500 [mm] Regime: Lifting Q=0 t - 12 t, Luffing:* $R_{max} \rightarrow R_{min}$

1-Luffing level Lx6 [m] Navy Blue
2-Deflection of front-right leg Portal h1x500 Lilac [mm]
3-Deflection of front-left leg Portal h2x500 [mm] Green
4-Deflection of rear-left leg Portal h3x500 [mm] Blue
5-Deflection of rear-right leg Portal h4x500 [mm] Purple

2.4 CONTROL OF EIGENVALUE

Dynamic properties of the cranes can also be checked by measuring the period of free oscillations of the structure. Figure 6 shows an experimental record of cargo swinging (weight of 12 t) caused by operation of reach changing by the mechanism for reach changing (after t=10 s). From the diagram was read an average period of cargo oscillation Texp (2) = 13.85 sec. The free oscillation period of cargo is controled by a model based on the mathematical pendulum with the rope length of 43 m, T_{MAT} = 13.75 sec. Comparative control value was obtained numerically with the FEM modal analysis. Thus, the periods of oscillation of the hanged cargo (weight of the M15) for the observed reaches of jib is $T_{FEM(2)}$ = 13.454 ÷ 14.680 sec. Comparing the analytical (T_{MAT} = 13.75 sec) and numerical ($T_{FEM\ (2)}$ = 13.454 ÷ 14.680 sec) period of model oscillation with the experimentally obtained period of oscillation of the cargo on the rope (T exp (2) = 13.85 sec), the agreement of results was determined. It was concluded that this simple model can be used as the basis for dynamic analysis.

3 CLASS OF DYNAMIC RESEARCH

Cranes for shipbuilding are high, elastic, weight more than 300 t. Therefore, the following research tasks are set in their design:

- Dynamic stability of the crane support structure in terms of the lowest given eigenvalues. This refers to the reduced presence of low ranges vibration. It is achieved by increasing the rigidity of long members.
- Dynamic shock and dynamic coefficients when cargo swinging caused by rope deflection for the maximum prescribed (allowed) hangings angle. Checks the ability of the structure to receive the extreme dynamic forces.
- Dynamic shocks caused by an unexpected raising or releasing of the cargo. That is the control of supporting structure on the unexpected impact effect (ramp function) with the aim to find the weak parts of the structure.
- Dynamic strikes caused by extreme wind effects. Checks the dynamic and static stability of pulsed hurricane wind effect. Wherein the worst variants are from the resonant frequency of the wind.
- Dynamic phenomena caused by the chosen kinematics for the reach change or load lifting. It checks the speed adjustment with the general stability of the structure. Won't the drive cause jiggling of the structure? Is the change of the reach and lifting (kinematics) steady?
- Dynamic stability at ambiental seismic shock. Checks the effects of the expected seismic action on the structure. Checks are going up to the level of the floor faulting. Appearance of coast denivelation is the risk for the construction static stability.
- Checking the dynamic stability of the crane when it bumps into the end of the track due to the act of frontal wind.
- Checking the dynamic result caused by random events like the fall of cargo, or bumping into or collision with another crane.

The next research question is which methods for dynamic analysis to choose? Research of P. J Sadler, Sandor NG [4], which deal with the elasto dynamic analysis of four-bar linkage mechanism, show that a harmonic analysis gives complex solutions of differential equations, which are not suitable for analytical solution. Therefore, a lot of research in the field of applied mechanisms, introduce the finite element method [5,6,12,13,14], in order to replace "rigid" models with real-elastic properties. Therefore, for practical reasons,

the finite element method (FEM) for real - practical research is being used in the last twenty years.
The selection of the non-linear model for the analysis of crane structure is performed because of extreme geometric nonlinearity caused by a relatively large elastic deflection. For practical reasons, usually is applied the variations formulation of the FEM method [8]. For the solution of forced oscillations of discrete mechanical model of cranes with two booms and four-bar linkage mechanism are proposed differential equations of motion:

$$\left[M\right]^t\{\ddot{u}\}+{}^t\left[C\right]^t\{\dot{u}\}={}^t\{f_{ext}\}-{}^t\{f_{int}\} \tag{2}$$

where [M] and [C] is mass and damping matrix, $\{F_{EXT}\}$ and $\{f_{int}\}$ external (excitation) and the interior (elastic) forces of a set of finite elements, $\{\dot{u}\}$ and $\{\ddot{u}\}$ the first (velocity) and the second (acceleration) time derivative of structural displacement, t (left index) time at which the quantity observed (ie, acceleration, damping, speed and force). Procedure of nonlinear analysis is described in detail in the paper [11].
Geometric nonlinear analysis requires calculation of the stress $\{\sigma\}$ in the current structural configuration and then the integration of these stresses in the structural continuum ${}^tV$, in order to obtain internal structural forces. It follows the equation (2) in which [B] is the matrix deformation-shift:

$${}^t\{f_{int}\} = \int_{{}^tV} {}^t[B]^t\{\sigma\}d^tV \tag{3}$$

Geometric nonlinear analysis requires direct integration of equations of motion. Here is used an implicit method for solving differential equations (PLC SIEMENS - FEMAP 2017)(1). Newton-Raphson's iterative procedure was applied. Thus, the system of equations for the geometrically nonlinear structural dynamic analysis at time $t+\Delta t$ has the form (4):

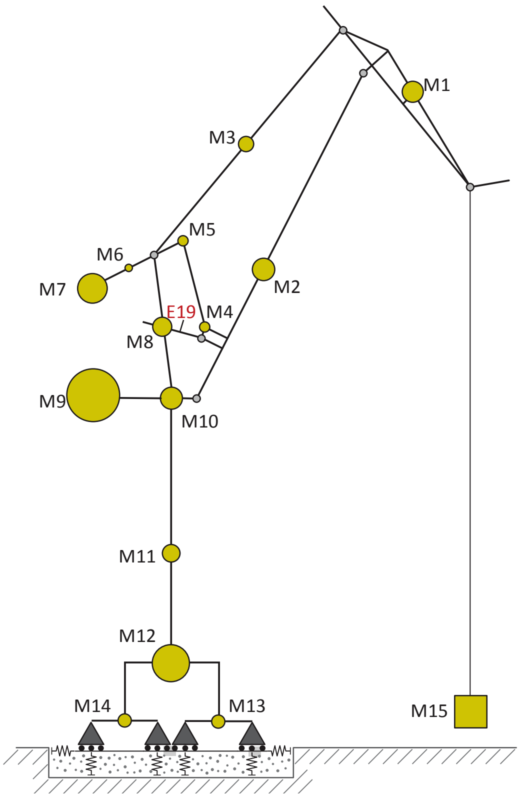


***Fig. 8.** Model for dynamic analysis*

$$\begin{aligned}&\left[M\right]^{t+\Delta t}\{\ddot{u}\}^{(k)}+{}^{t+\Delta t}\left[C\right]^{t+\Delta t}\{\dot{u}\}^{(k)}+\\&+{}^{t+\Delta t}\left[K_T\right]^{t+\Delta t}\{\Delta u\}^{(k)}={}^{t+\Delta t}\{f_{ext}\}-{}^{t+\Delta t}\{f_{int}\}^{(k-1)}\end{aligned} \tag{4}$$

where $[K_T]$ is the tangent stiffness matrix, Δ time increment and k the iteration. Tangent stiffness matrix with the increment of displacement allows estimation of internal forces at time $t+\Delta t$.
Figure 7 presents the simple physical model of crane for shipbuilding with 15 concentrated masses.

4 TRANSIENT FEM ANALYSIS CAUSED BY CARGO SWINGING

Searching for dynamic response on excitation caused by swinging of the largest cargo (on a long rope) declined for angle of ± 3º is only one of the above research tasks. Swinging large cargo has similarity with "throwing the hammer" and could undermine the stability of the whole structure.
Observed task is a task of forced dynamic action, with the damping in materials and joints and variable position of the oscillating load. The solution is sought in the application of nonlinear dynamic FEM analysis. As by analyses are observed transitory dynamic behaviors, transient analysis (Anglo-Saxon terminology) is therefore applied. Transient analysis (transient response analysis) is carried out by the numerical integration of the equations of motion. The non-linearity of elastic deformation of structure during the static and dynamic forces is treated for solving this problem by the Newton-Raphson method. The analysis includes the total structural dynamic damping force proportional to the displacement q and preferred structural dynamic coefficient G = 0.05 [8]. Two additional parameters are used in transient response analysis to convert structural damping to equivalent viscous damping, parameter $\boldsymbol{\omega_3}$ and parameter $\boldsymbol{\omega_4}$. The speed of the frequency damping is determined first by modal analysis for certain position of the four-bar linkage mechanism ω_2 = 0.42 ÷ 0:47 Hz, ω_3 = 0.93 ÷ 0.74 Hz, ω_4 = 1.40 ÷ 0.92 Hz. For execution of the nonlinear procedure is adopted N_{step} = 30000, with a time step increment Δt_{incr} = 0.01 sec. By analysis is obtained behavior of the model for a period of 300 sec. Newton-Raphson iteration is limited on up to 10 iterations and the maximum non-linearity convergence criterium is defined by displacement tolerance of ε = 0.001. In this study, a member for reach change control was considered (item E-19 between 14-21 knots). Here is observed dynamic change of force on a member for control of the reach change, taking into consideration its responsibility (on the model it is marked as element E19, Figure 8). On the crane is the cargo with mass of M = 12 t. One realization of calculated transient analysis (description of the inner axial force change in the member E19) is presented in Figure 9, at different values of damping. Cargo and the pivot point (on a see-saw) with verical direction formed angle of 3°. The research suggests differences in the linear and nonlinear analysis [12].
Experimental research carried out directly on the ropes discover the duration of the process separation of cargo from the ground and the oscillation of lifted cargo - Figure 10. The figure shows the process of two successive lifting

and lowering the cargo to the ground. Time of cargo separation from the ground is strictly an elasticity characteristic of crane structure and of the ground from which the cargo is raised. These measurements were performed with special transducer developed at the Faculty of Mechanical Engineering in Nis, which measures the force in the rope during the rope movement through the encoder. The encoder is stationary. Time duration determined by measurement of the transient process can be used to simulate dynamic impact on new types of of cranes with higher speeds. The experiment detects frequency spectrum characterising the structure and real damping of the whole system. Measured dynamic behavior enables to assess whether the stability of the structure is compromised with some the action on the structure. Also, the measurements show whether the kinematics is adapted to the stability of the crane. That makes possible to increase processing performance of these machines.

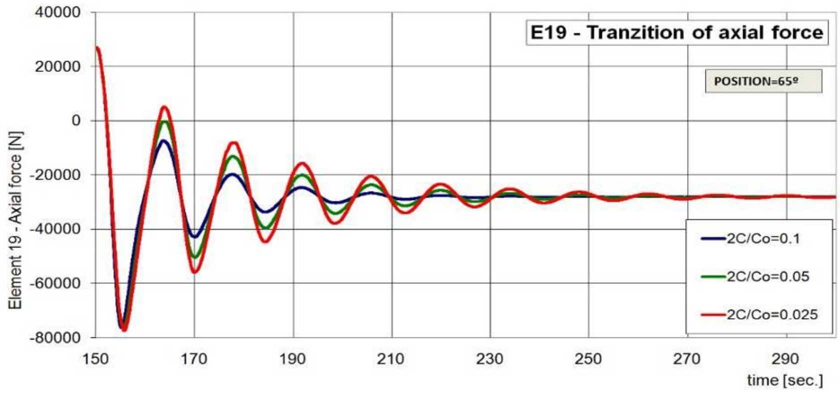


Fig 9. *The result of the dynamic modeling:*
Forces in the member for reach change with different damping (positions φ = 65 °)

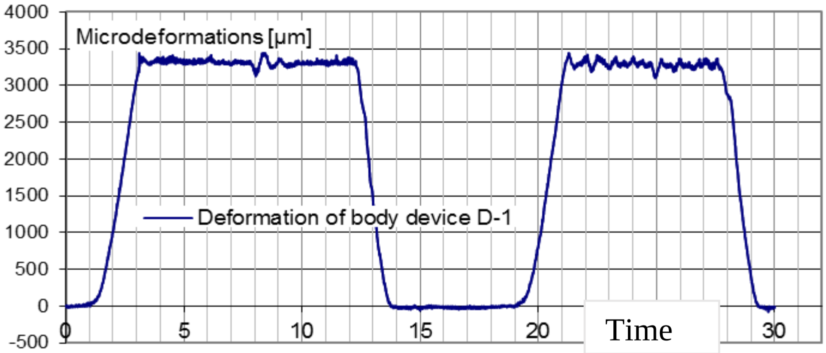


Fig 10. *Measurement of force in a rope while cargo lifting from the ground (two times)*

4 CONCLUSION

1 Identification of many non-linear phenomena in cranes required control (verification) of analytical and numerical results with the experimental studies.

2 It is recommended to use non-linear analysis (theory of second order) for the introduction of elastic deformation in the analysis. Elastic deformations have a significant impact on the internal forces in the joint of system for reach changing. With the introduction of elastic deformation, static and dynamic tasks can be modeled better.

3 The obtained data from the experiments allow the elimination of the assumption about the duration and nature of transient dynamic phenomena. This enables a more realistic simulation. In order to use other people's experimental research, it is recommended in the future to introduce public databases of built structure.

4 FEM technology simplifies the work process with a lot of structural data and successfully replaces the classic analytical approach. Numerical Analysis today is practical mathematical discipline in dynamic analysis for treatment of the real geometry of high cranes.

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Crane PULA MIN-2 is characterized by a structure of 67 (m) height, the maximum of reach of 40 (m), with a base of the portal 6.x8. (m), a total weight of 400 (t). The crane has a system for the reach change in the form of four-bar linkage mechanism and members and rocker connected with joints. Payload is 25/15/5 (t), the reach 27/37/40 (m), lifting height is 45 + 10 (m), the rate of reach change is 10 (m / min) and with working drive class 3. Crane is formed on the column with height of 30 (m) above which is compact on rotary platform mounted tower. On the tower were seated the main members for the reach change: tie road, jib and rocker. Drive device for the reach change is made with helical screw, which is acting on main jib. The drive mechanism for the reach change is driven by asynchronous motor. Balancing of the crane system of boom is carried by the structure in the form of four-bar linkage mechanism, lever and balancer with a weight mass 21 (t). Balancing of the entire crane was carried out by a fixed weight of 100 (t) on the rotating platform. The crane has a lift for the staff, well-controlled drives and still modern solution of the system for a reach change. The designer of the crane is MIN Niš FITIP ie the mechanical engineer Budimir Velimirović (1947) and Civil Engineer Ljuba Stanković (1947). For the successful implementation of this project diligently worked about 100 workers of MIN Niš. MIN Niš between the 1983 and 1988 produced 13 port and shipyard cranes for the former Yugoslavia. Photo on the right shows a crane-2 in the Pula shipyard "Uljanik" in Pula (now Croatia) recorded in 1988. The crane is still in use today.

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