Application of Non-Dimensional Models in Dynamic Structural Analysis of Cranes under Moving Concentrated Load

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Abstract. This paper gives an analysis of dynamic behaviour of the waterside boom, identified as the most important structural part, of the large ship-to-shore (STS) container cranes, under moving concentrated load. Non-dimensional mathematical model used in this paper presents a conceptual substitution of the real system of the mega container crane boom, and provides understanding and prediction of its dynamic behaviour under the action of moving trolley. The paper discusses the procedure for set up the nondimensional mathematical model of the crane boom as a necessary condition for qualitative estimation of structural parameters, such as the effects of the stiffness, mass and spatial position of structural segments on the dynamic structural response, i.e. on the values of deflection under the moving mass. Differential equations of the idealized mathematical model of the boom are solved numerically. The results obtained by the simulation of the trolley motion alongside the STS container boom during container transfer from shore-to-ship are implemented in parameter sensitivity analysis, in order to obtain the mutual influences of structural dynamic parameters. The variation of the values of the structural parameters (stiffness, geometry and mass) in the real diapason has shown their significant impact on the dynamic values of deflections. The procedure used in this paper can be applied in solving structural dynamics problems for some other related constructions such as e.g. slewing tower cranes, gantry cranes, unloading bridges, etc.

1 Introduction

The moving load problem is one of the fundamental problems in structural dynamics. In contrast to other dynamic loads this load varies not only in magnitude but also in position. The importance of this problem is manifested in numerous applications in the field of transportation. With respect to the methods one can look for the investigation of beams under moving loads in the fields of e.g. rail-wheel dynamics and magnetically levitated vehicles. The detailed review of previous researches, including comprehensive references list, can be found in a dedicated excellent monograph written by Fryba [1] on the subject of moving load problems, where most of the analytical methods previously used are described, as well as in the following review papers [2], [3], [4] and [5]. In the past 30 years applications of moving load problem have been presented in mechanical engineering studies. Typical structure under moving mass in mechanical engineering are overhead cranes, gantry cranes, unloading bridges, slewing tower cranes, cableways, guideways, shipunloaders or quayside container crane. Beam structures are commonly considered as three-dimensional bodies with one dimension significantly larger than the other two. In general, a length to width (height) ratio of 5-10 of the structural members is sufficient. This means that the mechanical structures subjected to moving loads, such as e.g. large span cranes, are often well qualified for being modelled as beam structures. It is generally applied fundamental beam theory according to Euler-Bernoulli and Saint Venant, which is not including warping (Vlasov) torsion, because cross sections of large cranes relevant for moving load analysis generally have closed thin-walled profiles, such as for bridge cranes, gantry cranes, container cranes, etc. Some of the structural assumptions in analysis of moving load problem are: initially straight beam, undeformable beam cross-section, linear elastic material and constant material properties over the beam cross section, small structural deformations, shear strains according to Saint Venant and negligible damping effects [4]. Bernoulli-Euler beam theory is valid for lower modes analysis of small vibration of slender beams at relatively low velocities that is just the case for supporting structures of cranes, where merely the lower modes of natural vibrations are of practical relevance for the analysis of dynamic behaviour and where the maximum currently existing velocities, for e.g. mega container cranes, are not exceeding 6 m/s that is small comparing with velocities of trains. For low velocities the bending effect is dominant factor on structural vibrations. The following two features distinguish the moving load problem in crane industry from that in civil engineering. The first is that the structure on which the moving mass moves always has travelling or rotating motion. The second is that the payload of a crane is attached via cables to a trolley moving along the structure. Thus the dynamics of a crane includes both the vibration of the structure and the dynamics of the payload pendulum.

The basic approaches in trolley modelling are: moving force model; moving mass model; trolley suspension model, existing in some special structures of unloading bridges. The simplest dynamic trolley models are the moving force models. The consequences of neglecting the structure-vehicle interaction in these models may sometimes be minor. In most moving force models the magnitudes of the contact forces are constant in time. A constant force magnitude implies that the inertia forces of the trolley are much smaller than the dead weight of the structure. Thus the structure is affected dynamically through the moving character of the trolley only. All common features of all moving force models are that the forces are known in advance. Thus structure-trolley interaction cannot be considered. On the other hand the moving force models are very simple to use and yield reasonable structural results in some cases. Moving mass as suspension model is an interactive model. Moving mass model, as well as moving force model, is the simplification of suspension model, but it includes transverse inertia effects between the beam and the mass. Interaction force between the moving mass and the structure during the time the mass travels along the structure considers contribution from the inertia of the mass, the centrifugal force, the Coriolis force and the time-varying velocity-dependent forces. These inertia effects are mainly caused by structural deformations (structure-trolley interaction) and structural irregularities. Factors that contribute in creating trolley inertia effects include: high trolley speed, flexible structure, large vehicle mass, small structural mass, stiff trolley suspension system and large structural irregularities. Finally, the trolley speed is assumed to be known in advance and thus not depend on structural deformations. For moving mass models the entire trolley mass is in direct contact with the structure. In general, the dynamic structure-trolley interaction predicted by such models is very strong. The trolley suspension model is representing physical reality of the system more closely (moving oscillator problem), but it is of little interest in cranes dynamics because, as a rule, the frame of the crane trolley is rigid.

The application of moving load problem in cranes dynamics has obtained special attention on the engineering researchers in the last years, but unfortunately little literature on the subject is available. The paper [6] is according to the authors' best knowledge the first attempt to increase the understanding of the dynamics of cranes due to the moving load. Also, some other papers dealing with the same kind of problem, such as [7], [8], [9], [10], should be mentioned.

2 Non-dimensional mathematical model of the large container crane boom

The last 40 years has seen mounting interest in research on the modelling and control of cranes, but only few of them are treated container cranes dynamics. These models can be distinguished by different complexity of modelling and by the nature of the neglected parameters. The most common modelling approaches are the lumped-mass and distributed mass approach, as well as the combination of the first two approaches. A recent review on cranes dynamics, modelling and control is given in [11], but without considering problems of moving load influence on dynamic response of cranes.

Modern mega ship-to-shore (STS) container cranes have more than doubled in outreach and load capacity comparing to the older traditional constructions. This is not easily accomplished given the cantilevered nature of STS container cranes. A cantilever is structurally inefficient because almost all of the structural strength and weight is needed to support its own weight. The stiffness of the structure affects the deflection magnitude and the vibration frequency. By increasing the stiffness of the crane structure, the deflection will decrease and the vibration frequency will increase. In practice it is very difficult and expensive to do an experimental research on a real size mega STS container crane. For that reason the investigations on mathematical models are necessary, especially during the design stage. Simpler models of mega cranes enable easier mathematical analysis and give better insight in the design and the possibilities of different control algorithms. On the other hand, more complex models are necessary to approximate the reality closer, e.g. the flexibility of the crane structure will certainly affect the behaviour of the controller. But, it is impossible to include all effects of the real life in a mathematical model of large container crane [12]. However, numerical examination of a model that is not a prototype of some real system is of little interest unless some general conclusions, which can be applied to other, related configurations [10]. The choice of an adequate model of container crane should be determined by the particular problem under consideration and must take into account the eigenfrequencies of the container crane structure as a whole in order to enable suitable dynamic analysis of single structural parts, e.g. the crane boom.

It is necessary to consider the flexibility of the container crane upper structure. That can be done only after analyzing the dynamic behaviour of the whole structure of container crane, i.e. after defining natural modes of STS container crane structure either by experimental methods or by FEM. However, experimental methods, in some cases, can be very difficult. Because a FEM study uses numerical models that have a greater degree of resolution and refinement than experimental models, results obtained from a finite element analysis may be more accurate than those of any experimental model. Hence, FEM is a valuable tool for evaluating the structural dynamic characteristics of machines and structures, and can be used to estimate the natural frequencies and mode shapes for equipment and supporting structures [13]. The mentioned facts apply for all types of cranes. For dynamic analysis in this paper the mathematical model is set up for the real mega container crane, with monogirder boom (total boom length 69,2 m) and trapezoidal cross section. It is adopted the Machinery-on-Trolley (MOT) concept giving the heaviest possible value for total moving load of 175 t. The whole crane structure is modelled by using FEM and by applying beam elements, Figure 1 (at left). The first three modes, obtained by using FEM, are relevant for dynamic analysis: vibrations of boom in horizontal plane; vibrations of the structure in the vertical plane in direction of trolley motion; vibrations of the boom in the vertical plane and in vertical direction perpendicular to the direction of trolley motion. Excitation of structure in service due to the motion of load is most important from the aspect of dynamic analysis, and will be considered in this paper. The outreach (boom), due to its large dimension and flexibility, is the most representative structural part identified for analysis of dynamic behaviour. This fact confirms the cantilever nature of quayside container cranes, and imposes requirement for dynamic analysis of interaction problem between boom on the water side leg of the crane and trolley as a moving load, i.e. trolley impact on the change of maximum values of deflections. It is observed that the vibrations of the boom on the water side leg in vertical direction perpendicular to trolley direction are practically independent from other structural parts, and this vibration is recognized as one in the range of the first three vibrations with lowest frequencies most important for dynamic analysis, Figure 1 (at right) [12].

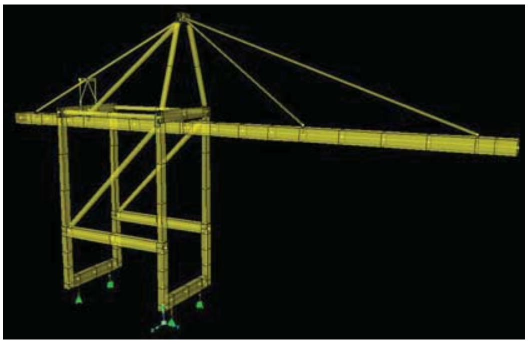
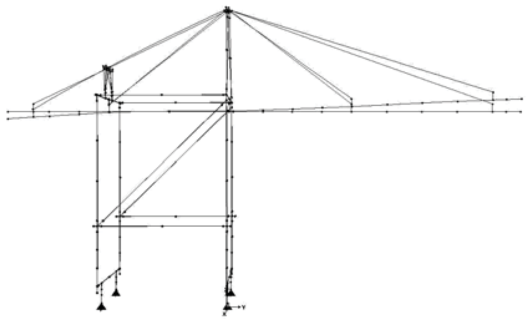

Figure 1. FEM model of the large STS container crane (at left) and vibration mode shape for vibrations of the crane in the vertical plane with frequency of f = 1.568 Hz (at right).

In the next stage of modelling process, consisting of several intermediate stages and by making appropriate procedure for dynamic modelling of structure, the idealized equivalent reduced model of the boom competent for writing differential equations of the moving load problem is obtained, as described in [12]. Relative deviation of natural lowest frequency of vibrations in vertical direction for idealized dynamic model is 1,36% in comparison with the FEM model. So, it is shown that the FEM model is quite acceptable for validation of reduced idealized dynamic model, and the obtained deviation is very small from the view-point of an engineer. Equivalent mathematical model relevant for setting up differential equations of system motion is shown in Figure 2 (at left) [12]. Equivalent stiffnesses c_1 and c_2 represents respectively the reduced stiffnesses of the inner stay and the forestay including the stiffness of the upper structure with mast, while the lumped masses M_1 and M_2 comprise the masses from the stays weight. Length $L_r = L = 65,8m$ presents the real trolley path between the point A and forestay connection with boom in point C. The boundary conditions in point A have to be modelled as for a hinge, having in mind the real structural solution for the boom connection with other parts of the upper structure, Figure 2 (at right) [12].

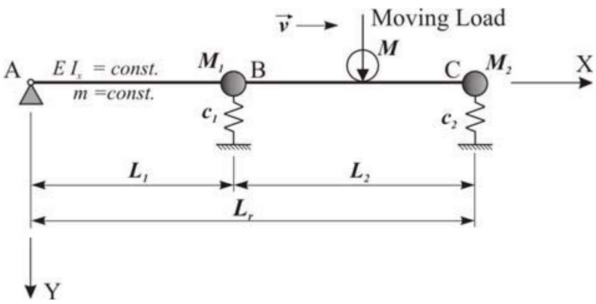

Figure 2. Mathematical model of the container crane boom (at left) and the structural solution for hinge in point A (at right).

The problem of moving load is treated in the analysis presented in [12] as a moving mass problem, i.e. the inertia of the trolley mass has not been neglected. Differential equations of motion are obtained by Lagrange's equations by using Assumed Modes Method as an alternative to Rayleigh-Ritz method and by neglecting dissipation function (damping). This method is employed to approximate the structural response in terms of finite number of admissible functions that satisfy the geometric boundary conditions of the mathematical model shown in Figure 2. Selection and estimation of the admissible functions is done by using variational approach. Mathematical model of moving mass includes in itself influence of the moving mass inertia, influence of the Coriolis centripetal force, and influence of the moving mass deceleration (braking).

The main goal of this paper is to present the non-dimensional mathematical model for the prediction of the boom dynamics of operating large STS container crane. Non-dimensional mathematical model used in this paper presents a conceptual substitution of the real system of mega STS container crane and provides general understanding of the dynamic behaviour of container crane boom under the action of moving trolley. So, the obtained results can be applied for analyzing dynamic behaviour of a series of similar constructions of STS container cranes. The mathematical model for set up non-dimensional equations of motion is shown in Figure 3.

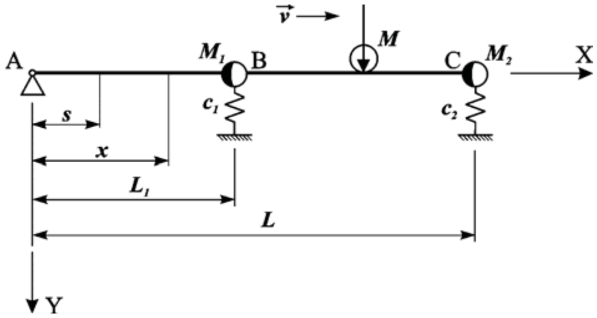

Figure 3. Model of the crane boom acted upon by the concentrated moving mass M.

The co-ordinate system shown in Figure 3 is assumed to be fixed in the inertial frame with the $\vec{i}$ unit vector parallel to the undeformed beam and the $\vec{j}$ unit vector pointing downward in the direction of the gravitational field g. The position of the mass at any instant is induced by $x = s$ in the i direction. The parameter s is a known and prescribed function of time. The quantity y is defined as the transverse deflection of an arbitrary point located at x along the beam. The position vector of a general point p on the deformed beam is given by:

$$\vec{p} = x\vec{i} + y\vec{j} \ . \tag{1}$$

The velocity at the point is:

$$\vec{v}_p = \dot{y}\vec{j} \ , \tag{2}$$

where $\dot{y} = \dfrac{dy}{dt}$.

For simplicity in the subsequent computations, the following dimensionless quantities, similar as in [14], are introduced:

$$\begin{gathered}\tau = t\sqrt{\frac{EI}{mL^4}}, \bar{L}_1 = \frac{L_1}{L}, \bar{L} = \frac{L}{L}, \bar{y} = \frac{y}{L}, \xi = \frac{x}{L}, \bar{s} = \frac{s}{L}, \bar{c}_1 = \frac{c_1 L^3}{EI}, \bar{c}_2 = \frac{c_2 L^3}{EI} \\ \bar{M} = \frac{M}{mL}, \bar{M}_1 = \frac{M_1}{mL}, \bar{M}_2 = \frac{M_2}{mL}, \bar{v} = v\sqrt{\frac{mL^2}{EI}}, \bar{g} = \frac{gmL^3}{EI}, \bar{a} = \frac{amL^3}{EI}\end{gathered} \tag{3}$$

By using the assumed mode method, the dimensionless deflection $\bar{y}$ can be expressed as

$$\bar{y} = \frac{y}{L} = \sum_{i=1}^{5} \bar{q}_i(t)\phi_i(\xi), \tag{4}$$

where ϕ_i are spatial functions that satisfy the prescribed geometric boundary conditions at the two ends of the beam. Those functions are adopted from the set of admissible functions. Admissible functions are employed because eigenfunctions of the system shown in Figure 2 cannot be determined in practice due to the non-standard boundary conditions and added lumped masses. Admissible functions are any set of functions that satisfy the geometric boundary conditions of the eigenvalue problem and are differentiable "n" times. Finally, after several iterations the following five ($n = 1,\bar{5}$) admissible functions (these functions should not be "blindly" selected) are assumed as [12]:

$$\phi_1(\xi) = \xi, \phi_2(\xi) = \sin \pi\xi, \phi_3(\xi) = \sin 2\pi\xi, \phi_4(\xi) = \sin 3\pi\xi, \phi_5(\xi) = \sin 4\pi\xi \ . \tag{5}$$

The equation of motion can be formulated using the Lagrangian approach either by considering the presence of the moving mass in terms of the contact force [14], or by including the mass to be part of the system, with the external force acting on the system given by gravitational force alone. For the second approach, including the mass to be part of the system shown in Figure 2 (at left), and by adopting $x = s$ and $s = s(t)$, and having in mind that $\dot{s} = v, \ddot{s} = a$, the kinetic energy of the moving mass T_M is:

$$T_M = \frac{1}{2} M \left[v^2 + \left(\frac{dy(s,t)}{dt} \right)^2 \right], \tag{6}$$

where

$$\frac{dy(s,t)}{dt} = \frac{\partial y(s,t)}{\partial s} v + \frac{\partial y(s,t)}{\partial t}, \tag{7}$$

and by substituting the expression (7) into the expression $y(s,t) = \sum_{i=1}^{5} \phi_i(s) \cdot q_i(t)$, we obtain $\frac{dy(s,t)}{dt} = \sum_{i=1}^{5} \left[\phi_i'(s) \cdot v \cdot q_i(t) + \phi_i(s) \cdot \dot{q}_i(t) \right]$, and finally the expression for the kinetic energy of the moving mass is

$$T_M = \frac{1}{2} M v^2 + \frac{1}{2} \sum_{i=1}^{5} \sum_{j=1}^{5} \left(\phi_i'(s) v q_i + \phi_i(s) \dot{q}_i \right) \left(\phi_j'(s) v q_j + \phi_j(s) \dot{q}_j \right). \tag{8}$$

The kinetic energy of the beam and concentrated masses M_1 and M_2, $T_{B,M}$ is:

$$\begin{aligned} T_{B,M} &= \frac{1}{2} \int_0^L m \left[\frac{\partial y(x,t)}{\partial t} \right]^2 dy + \frac{1}{2} M_1 \left[\frac{\partial y(L_1,t)}{\partial t} \right]^2 + \frac{1}{2} M_2 \left[\frac{\partial y(L,t)}{\partial t} \right]^2 = \\ &= \frac{1}{2} \int_0^L m \left[\sum_{i=1}^{5} \phi_i(x) \dot{q}_i(t) \right] \left[\sum_{j=1}^{5} \phi_j(x) \dot{q}_j(t) \right] dx + \frac{1}{2} M_1 \left[\sum_{i=1}^{5} \phi_i(L_1) \dot{q}_i(t) \right] \left[\sum_{j=1}^{5} \phi_j(L_1) \dot{q}_j(t) \right] + \\ &+ \frac{1}{2} M_2 \left[\sum_{i=1}^{5} \phi_i(L) \dot{q}_i(t) \right] \left[\sum_{j=1}^{5} \phi_j(L) \dot{q}_j(t) \right] \end{aligned} \tag{9}$$

By using the well-known notation for the kinetic energy $T = \frac{1}{2} \sum_{i=1}^{n} \sum_{j=1}^{n} m_{ij} \dot{q}_i(t) \dot{q}_j(t)$,

the expression (9) for five admissible functions $n = 5$ can be written as:

$$T_{B,M} = \frac{1}{2} \sum_{i=1}^{5} \sum_{j=1}^{5} \dot{q}_i(t) \dot{q}_j(t) \left[\int_0^L m \phi_i(x) \phi_j(x) dx + M_1 \phi_i(L_1) \phi_j(L_1) + M_2 \phi_i(L) \phi_j(L) \right]. \tag{10}$$

The bending elastic strain of the beam including potential energy of the spring elements c_1 and c_2 is:

$$\begin{aligned} V_{B.M} &= \frac{1}{2} \int_0^L EI \left[\sum_{i=1}^{5} \frac{\partial^2 \phi_i(x)}{\partial x^2} q_i(t) \right] \left[\sum_{j=1}^{5} \frac{\partial^2 \phi_j(x)}{\partial x^2} q_j(t) \right] dx + \frac{1}{2} c_1 \left[\sum_{i=1}^{5} \phi_i(L_1) q_i(t) \right] \left[\sum_{j=1}^{5} \phi_j(L_1) q_j(t) \right] + \\ &+ \frac{1}{2} c_2 \left[\sum_{i=1}^{5} \phi_i(L) q_i(t) \right] \left[\sum_{j=1}^{5} \phi_j(L) q_j(t) \right] \end{aligned} \tag{11}$$

By using the well-known notation for the potential energy $V = \frac{1}{2} \sum_{i=1}^{n} \sum_{j=1}^{n} c_{ij} q_i(t) q_j(t)$,

the expression (11) for five admissible functions $n = 5$ can be written as:

$$V_{B,M} = \frac{1}{2} \sum_{i=1}^{5} \sum_{j=1}^{5} q_i(t) q_j(t) \left[\int_0^L EI \frac{\partial^2 \phi_i(x)}{\partial x^2} \frac{\partial^2 \phi_j(x)}{\partial x^2} dx + c_1 \phi_i(L_1) \phi_j(L_1) + c_2 \phi_i(L) \phi_j(L) \right]. \tag{11}$$

Potential energy of the moving mass is:

$$V_M = -M \cdot g \cdot y(s,t) = -Mg \sum_{i=1}^{5} \phi_i(s) q_i . \tag{12}$$

The total kinetic energy of the system is $T = T_{B,M} + T_M$, while the total potential energy of the system is $V = V_m + V_M$. By substituting the expressions for kinetic and potential energy in the Lagrangian of the system defined as $L = T - V$ and by using the well-known Lagrange's equations for the considered system $\frac{d}{dt}(\frac{dL}{\partial \dot{q}_j}) - \frac{dL}{\partial q_j} = 0$ [15], and dimensionless quantities given in (3), we obtain finally the non-dimensional equation of motion in the matrix form:

$$\left[\bar{M}\right]\left\{\ddot{\bar{q}}\right\}+\left[\bar{B}\right]\left\{\dot{\bar{q}}\right\}+\left[\bar{C}\right]\left\{\bar{q}\right\}=\bar{M}\bar{g}\bar{\Phi}\,, \tag{13}$$

where $\left[\bar{M}\right]=\left[\bar{m}\right]_{ij}+\bar{M}\left[\bar{H}\right]_{ij}$, $\left[\bar{B}\right]=2\bar{M}\bar{v}\left[\bar{A}\right]_{ij}$, $\left[\bar{C}\right]=\left[\bar{c}\right]_{ij}+\bar{M}\bar{v}^2\left[\bar{K}\right]_{ij}+\bar{M}\bar{a}\left[\bar{A}\right]_{ij}$, $\left[\bar{\Phi}\right]=\left[\phi(\bar{s})\right]_i$, while

$$\left[\bar{m}\right]_{ij}=\int_0^L\phi_i(\xi)\phi_j(\xi)d\xi+\bar{M}_1\phi_i(\bar{L}_1)\phi_j(\bar{L}_1)+\bar{M}_2\phi_i(\bar{L})\phi_j(\bar{L})\,,$$

$$\left[\bar{c}\right]_{ij}=\int_0^L\left(\frac{\partial^2\phi_i(x)}{\partial x^2}\right)\left(\frac{\partial^2\phi_j(x)}{\partial x^2}\right)d\xi+\bar{c}_1\phi_i(\bar{L}_1)\phi_j(\bar{L}_1)+\bar{c}_2\phi_i(\bar{L})\phi_j(\bar{L})\,,\quad \left[\bar{H}\right]_{ij}=\phi_i(\bar{s})\phi_j(\bar{s})\,,\quad \left[\bar{A}\right]_{ij}=\phi_i(\bar{s})\phi_j'(s) \quad \text{and}$$

$$\left[\bar{K}\right]_{ij}=\phi_i(\bar{s})\phi_j''(\bar{s})\,.$$

It is obvious that the non-dimensional matrix equation of motion (13) can be solved only numerically. This system of differential equations is solved numerically by using Runge-Kutta method of the V order (Method Runge-Kutta-Fehlberg, RK45), and by using program written in C++. System of differential equations is non-stiff, so the implementation of the fifth order Runge-Kutta method is strictly sustainable.

During the transhipment of containers from shore-to-ship (ship loading) it happens in reality that the moving mass (trolley with pay load - container) reaches its maximum velocity in the vicinity of the vertical direction of the hinge. So, it is assumed that the moving load starts to move from the left end (hinge) of the beam. Also, it is assumed the maximum path of the trolley, i.e. loading of the endmost container ship cell. For that reason the simulation of motion of the moving load is done by assuming that the total time of motion consists of two parts: uniform motion during the time t_u and transient motion (constant deceleration - braking) during the time t_b, i.e. $t=t_u+t_b$. Hence, the displacement $f(t)$ from the left end of the beam becomes $f(t)=vt_u+\frac{1}{2}at_b^2$. The dimensionless deflection under the moving load ($\bar{y}=y_n(\xi,\tau)$), including convergence study for the admissible functions n = 2, 3, 4, 5, is shown in Figure 4. It can be seen that the fast convergence of the solution for adopted 5 admissible functions is obtained, and it is found the excellent agreement between n = 4 and n = 5. This fast convergence reveals on the adequate number of the adopted admissible functions.

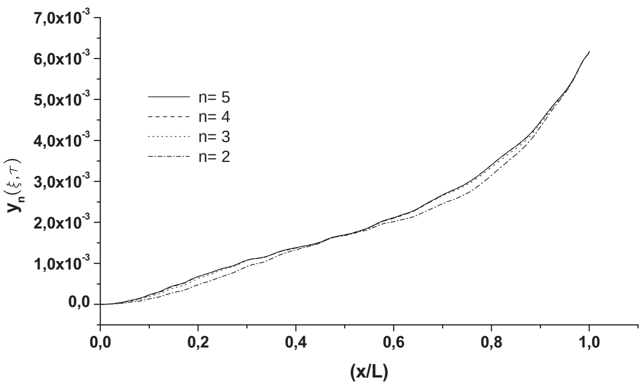

Figure 4. The convergence study for the dimensionless deflection y_n under the moving mass M.

3 Parameter sensitivity analysis of the dynamic system of the structure

For obtaining the qualitative estimation of the structural parameters and the dependencies of the non-dimensional boom deflection on the dimensionless structural parameters, the parameter sensitivity analysis method is used. Such a technique, although currently somewhat neglected by many in the system modelling and simulation fields, still have considerable relevance, particularly in structural dynamic problems. The obtained sensitivity functions can be used to establish the dependence of each part of a response time-history to each of the model parameters. Parameter sensitivity analysis techniques also provide useful methods for some types of validation problem [16]. This paper discusses the effects of changing structural parameters, such as stays stiffness, ratio between the moving mass and the mass of the beam (boom) and geometry of inner stay connection with the boom on the values of maximum deflections of the container crane boom. In this case parameter sensitivity analysis is used to validate the simulation model in relation to the model obtained by FEM, as the substitution of the real system of large STS container crane. It is worthwhile to mention that the variation of some dynamic parameters is done in the real diapason (not in the theoretical one) for modern constructions of mega STS cranes. That means that this investigation is particularly important from the aspect of an engineer's viewpoint. Only for some boundary states the certain level of imagination may be introduced, by expanding the scope of the real values of dynamic parameters in order to obtain the fitted sensitivity functions. That was necessary to validate the dynamic model.

The dependence of the dimensionless boom deflection $\overline{y} = y_n(\xi,\tau)$ under the moving load on the dimensionless inner stay stiffness $\overline{c}_1 = c_{1,n}$ is depicted in Figure 5. Dependence of the maximum values of the dimensionless boom deflections $y_{n,max}$ on the dimensionless values of the inner stay stiffness $c_{1,n}$ (a part of the sensitivity function in the real diapason) is shown in Figure 6. The dependence of the dimensionless boom deflection $\overline{y} = y_n(\xi,\tau)$ under the moving load on the dimensionless forestay stiffness $\overline{c}_2 = c_{2,n}$ is depicted in Figure 7. Dependence of the maximum values of the dimensionless boom deflections $y_n(\xi,\tau)_{max}$ on the dimensionless values of the forestay stiffness $c_{2,n}$ (a part of the sensitivity function in the real diapason) is shown in Figure 8.

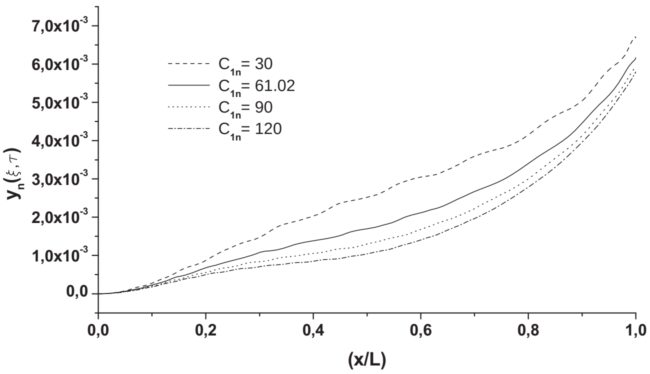

Figure 5. Dependence of the dimensionless deflection y_n under the moving mass on the inner stay stiffness $c_{1,n}$.

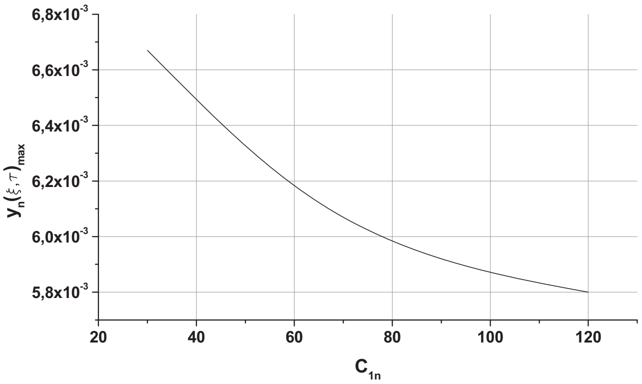

Figure 6. Dependence of the maximum value of dimensionless deflection $y_{n,\max}$ on the inner stay stiffness $c_{1,n}$.

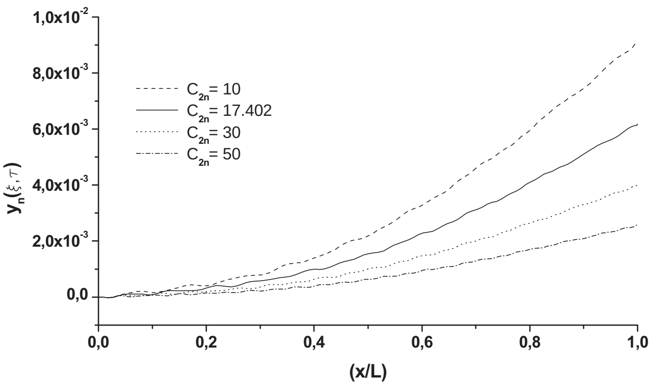

Figure 7. Dependence of the dimensionless deflection y_n under the moving mass on the forestay dimensionless stiffness $c_{2,n}$.

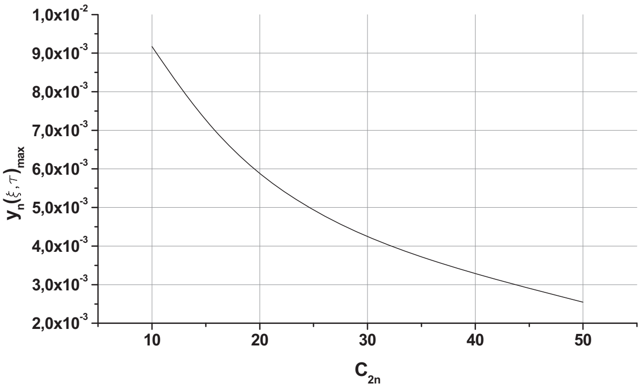

Figure 8. Dependence of the maximum value of dimensionless deflection $y_{n,\max}$ on the forestay stiffness $c_{2,n}$.

The dependence of the dimensionless boom deflection $\overline{y} = y_n(\xi,\tau)$ under the moving load on the normalized dimensionless ratio between moving mass M and mass of the beam (boom) mL (m is the distributed mass of the boom) $\overline{M} = M_{nor} = \dfrac{M}{mL}$ is shown in Figure 9. Dependence of the maximum values of the dimensionless boom deflections $y_n(\xi,\tau)_{max}$ on the normalized values of the ratio between masses M_{nor} (a part of the sensitivity function in the real diapason) is shown in Figure 10.

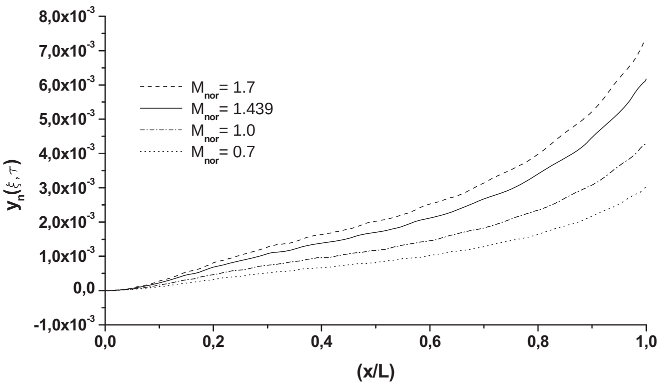

Figure 9. Dependence of the dimensionless deflection y_n under the moving mass on the normalized mass M_{nor} .

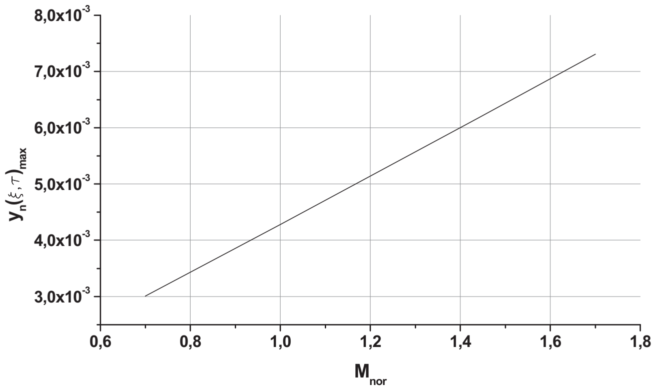

Figure 10. Dependence of the maximum value of dimensionless deflection $y_{n,\max}$ on the normalized mass M_{nor} .

The dependence of the dimensionless boom deflection $\overline{y} = y_n(\xi,\tau)$ under the moving load on the change of attaching point (geometric position alongside the boom) $\overline{L_1} = L_{1n}$ of the inner stay is shown in Figure 11. Dependence of the maximum values of the dimensionless boom deflections $y_n(\xi,\tau)_{max}$ on the dimensionless position of the inner stay along the boom (a part of the sensitivity function in the real diapason) is shown in Figure 12.

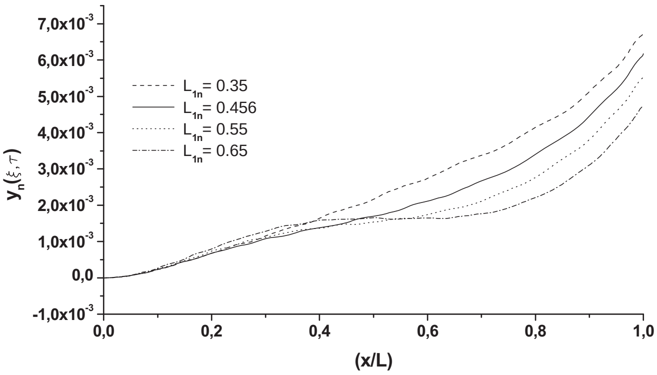

Figure 11. Dependence of the dimensionless deflection y_n under the moving mass on the inner stay geometry $L_{1,n}$.

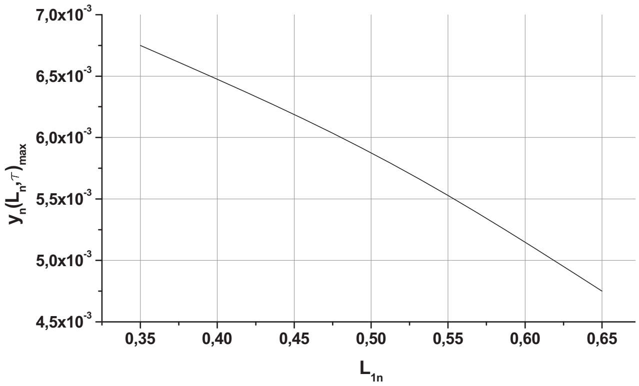

Figure 12. Dependence of the maximum value of dimensionless deflection $y_{n,\max}$ on the inner stay geometry $L_{1,n}$.

4 Conclusions

The crane is not only the part of the terminal system, but is also a system in its own right, and the optimum design requires balance. The deterministic computer simulation of the non-dimensional mathematical model of the STS container crane boom, as the most important structural part, is a kind of the numerical experiment instead of the extremely costly experiments on a real size crane or a scale-model. The results obtained by the simulation of the trolley motion alongside the STS container boom from shore-to-ship are used for parameter sensitivity analysis, in order to obtain the qualitative estimation of the structural parameters and the dependencies of the non-dimensional boom deflection on the dimensionless structural parameters such as are stiffnesses, masses and geometric configuration. At the same time, the parameter sensitivity analysis is used as a way of the model validation. External validation of model is done by the experts and experienced professional engineers (expert scrutiny) dealing with problems of large container cranes, as suggested in [17] as a way of model validation. The structure of mathematical model is consistent with the computer program, without problems in algorithm structure; thereby the internal verification of mathematical model is also done [16].

The main conclusions obtained by parameter sensitivity analysis are:

- The variation of the values of structural parameters (stiffness, geometry and mass) in the real diapason has the significant impact on the dynamic values of deflections. This conclusion can be used for making a new and more rational approach in the design of mega STS container cranes.
- Before adopting the final design solution of the STS container crane it is necessary to analyze in detail the values for stiffnesses and geometry of stays, as well as the location of the point of attachment of inner stay to the boom.
- The results obtained in this work may be used as a basic point for setting new and improved control algorithms by considering the structural flexibility of the boom, instead of the traditional and common approach where the crane structure is assumed to be rigid.

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