ESTIMATION OF STRUCTURAL DESIGN PARAMETERS OF HIGH PERFORMANCE CRANES BY USING SENSITIVITY FUNCTIONS

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Abstract: This paper presents an analysis of dynamic behavior of the high-performance ship-to-shore (STS) container crane (CC) waterside boom, identified as the most important structural part, under moving concentrated mass. Non-dimensional mathematical model applied in this paper is a conceptual substitution of the real system of the mega container crane boom (CCB), and enables understanding and prediction of its dynamic behavior under the action of moving trolley. The paper discusses the procedure for set up the nondimensional mathematical model of the CCB as a required condition for qualitative estimation of structural design parameters, such as the effects of the stiffness of structural segments on the dynamic structural response, i.e. on the values of deflection and bending moment under the moving mass. The results obtained by the simulation of the trolley motion alongside the STS CCB during container transfer from shore-to-ship are implemented in parameter sensitivity analysis, in order to obtain the corresponding sensitivity functions. The variation of the values of the structural design parameters (stiffnesses) in the real diapason shows their considerable impact on the dynamic values of deflections and bending moments.

Key words: crane, modeling, structural dynamic, simulation, sensitivity functions,

1. INTRODUCTION

Since its beginning 50 years ago (1959), the development of the container industry has been notable in many ways. Considering the enormous capital costs, one of the most remarkable changes has been in size of equipment and facilities. STS CCs are used in ports and terminals to transfer containerized cargo to and from ships. Over time, ships size and container weights have increased. High-performance mega container cranes with outreaches of 60 meters or more, lifts above rail of 46 meters, and capacities of 60 to 120 tons are already built or being built. While the size, mass and strength of the crane structure have also increased, the stiffness of the crane structure has not been increased proportionally. So, the crane response to trolley motion has changed, and can cause undesirable crane deflections in vertical plane. Increased trolley and hoist speeds are obvious targets for increased productivity [1, 2]. The crane is not only a part of the terminal system, but is also a system in its own right, and the optimum design requires balance. Currently there are two basic approaches regarding STS CC structural design. The first approach is to require a very stiff structure, with severe stiffness requirements and strict deflection limits, while the second approach requires a flexible supporting structure. The design problem in such a structure is in developing the optimum geometry and stiff forestay concept, where the sag in the forestay contributes to boom deflection. A detailed structural design process is required to minimize the weight and optimize the geometry and sections [2].

2. STRUCTURAL DYNAMICS AND MODELING OF STS CC UNDER MOVING LOAD: A LITERATURE REVIEW

The last 40 years has seen mounting interest in research on the modeling and control of cranes, but only few of them are treated container cranes dynamics. These models can be distinguished by different complexity of modeling and by the nature of the neglected parameters. The most common modeling approaches are the lumped-mass and distributed mass approach, as well as the combination of them are treated container cranes dynamics. These models can be distinguished by different complexity of modeling and by the nature of the neglected parameters. The most common modeling approaches are the lumped-mass and distributed mass approach, as well as the combination of

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some general conclusions, which can be applied to other, related configurations [6]. 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 [2, 6, 7, 8].

The moving load problem is one of the fundamental problems in structural dynamics. On the contrary to other dynamic loads this load varies not only in magnitude but also in position. The significance of this problem is manifested in many 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 basic approaches in trolley modeling 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 [8].

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 [9] is according to the authors’ best knowledge the first attempt to increase the understanding of the dynamics of cranes due to the moving load.

3. DIMENSIONLESS MODEL OF CC BOOM

It is necessary to consider the flexibility of the container crane upper structure. That can be done only after analyzing the dynamic behavior 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 [10]. 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 modeled by using FEM and by applying beam elements [2, 8].

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 behavior. 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 [2, 8].

In the next stage of modeling process, consisting of several intermediate stages and by making appropriate procedure for dynamic modeling 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 [2, 8]. 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 [2, 8]. Equivalent stiffnesses $c_1$ and $c_2$ represent 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 $L = L_6 = 65.8m$ 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 modeled as for a hinge, having in mind the real structural solution for the boom connection with other parts of the upper structure.

The problem of moving load is treated in the analysis presented in [8] 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 used 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). Non-dimensional mathematical model developed in [7, 8] and 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 [7, 8].

The co-ordinate system shown in Figure 3 is assumed to be fixed in the inertial frame with the $i$ unit vector parallel to the undeformed beam and the $j$ 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. By using the assumed mode method, the dimensionless deflection $\bar{y}$ can be expressed as

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

(1)

where $\phi_i$ 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 $(n = 1, 5)$ admissible functions (these functions should not be “blindly” selected) are assumed as [2, 8]:

$$\phi_1(\xi) = \xi, \phi_2(\xi) = \sin \pi \xi, \phi_3(\xi) = \sin 2\pi \xi,$$

$$\phi_4(\xi) = \sin 3\pi \xi, \phi_5(\xi) = \sin 4\pi \xi$$

(2)

The equation of motion is formulated using the Lagrangian approach by including the mass to be part of the system, with the external force acting on the system given by gravitational force alone as described in [8]. For straightforwardness in the subsequent computations, the following dimensionless quantities, as in [8], are introduced:
Finally the dimensionless equation of motion is obtained in the matrix form [8]:

\[
[M] \{\ddot{q}\} + [B] \{\dot{q}\} + [C] \{q\} = \tilde{M} g \bar{\Phi}
\]  

(4)

where:

\[
[M] = \begin{bmatrix} \tilde{M} \\ \tilde{M}_1 \\ \tilde{M}_2 \\ \vdots \\ \tilde{M}_n \end{bmatrix}, \quad [B] = 2 \tilde{M} \begin{bmatrix} \tilde{A} \end{bmatrix},
\]

\[
[C] = \begin{bmatrix} \tilde{C} \\ \tilde{M} \tilde{C} \\ \tilde{M} \tilde{C} \end{bmatrix} + \tilde{M} \begin{bmatrix} \tilde{R} \\ \tilde{R} \end{bmatrix} + \tilde{M} \begin{bmatrix} \tilde{A} \end{bmatrix},
\]

\[
[\bar{\Phi}] = \begin{bmatrix} \phi(\bar{x}) \end{bmatrix}.
\]

It is obvious that the non-dimensional matrix equation of motion (4) 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 transshipment 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 \( f(t) = v t + \frac{1}{2} a t^2 \). The dimensionless deflection under the moving load \( \bar{y} = y(x, t) \), 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. The same analysis for the dimensionless bending moment \( (M_{\bar{y}} = -E h y(x, t)) \) under the moving mass is shown in Figure 5. The convergence is not so fast as for the deflection and reveals on the necessity of assuming 5 admissible functions.

4. SENSITIVITY FUNCTIONS

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 dependency 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 [11]. 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. The variation of some structural parameters is done in the real diapason (not in the theoretical one) for modern constructions of mega STS cranes. 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 \( \bar{y} = y(x, t) \) under the moving load on the dimensionless inner stay stiffness \( c, c_{l_a} \) is depicted in Figure 6. Dependence of the maximum values of the dimensionless boom deflections on the dimensionless values of the inner stay stiffness is shown in Figure 7.

\[
\tau = t \sqrt{\frac{EI}{mL^2}}, \quad \bar{\nu} = \frac{v}{L}, \quad \bar{\xi} = \frac{x}{L}, \quad \bar{\zeta} = \frac{s}{L}, \quad \bar{\xi} = \frac{c_1 L^3}{EI}, \quad \bar{\zeta} = \frac{c_1 L^3}{EI},
\]

(3)
The dependence of the dimensionless bending moment \( M = M(x, t) \) under the moving load on the dimensionless inner stay stiffness \( c_{1,n} \) is depicted in Figure 8. Dependence of the maximum values of the dimensionless bending moment on the dimensionless values of the inner stay stiffness \( c_{1,n} \) (sensitivity function) is shown in Figure 8.

The dependence of the dimensionless boom deflection under the moving load on the dimensionless outer stay stiffness \( c_{2,n} \) is depicted in Figure 10. Dependence of the maximum values of the dimensionless boom deflections on the dimensionless values of the outer stay stiffness \( c_{2,n} \) (sensitivity function) is shown in Figure 11.

The dependence of the dimensionless bending moment under the moving load on the dimensionless outer stay stiffness is depicted in Figure 12. Dependence of the maximum values of the dimensionless bending moment on the dimensionless values of the outer stay stiffness \( c_{2,n} \) (sensitivity function) is shown in Figure 13.
5. CONCLUSION

The deterministic computer simulation of the non-dimensional mathematical model of the STS CCB, 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 CCB 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 and bending moment on the dimensionless structural parameters such as stiffnesses. 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 [11] as a way of model validation.

The main conclusions obtained by parameter sensitivity analysis are:

- The variation of the values of structural parameters (stiffness) has the significant impact on the dynamic values of deflections and bending moments. This conclusion can be used for making a new and more rational approach in the design of mega STS CC.
- Before adopting the final design solution of the STS CC it is necessary to analyze in detail the values for stiffnesses of stays.

ACKNOWLEDGMENT

A part of this work is the contribution to the Ministry of Science and Technological Development of Serbia funded Project TR 14052.

REFERENCES


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