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Dynamics and modelling of mega quayside container cranes

This paper discusses the impact of moving load to dynamic behavior of flexible construction of mega quayside container cranes, and focuses on the dynamic interaction between trolley and supporting structure caused by the moving load. The scope of the presented research is to give tha basic principles of dynamics and mathematical modelling of mega quayside container cranes, as well as how to obtain dynamic response of the crane boom structure due to moving mass, as are deflections, bending moments, dynamic magnification factor and acceleration of moving mass in vertical direction.

Keywords: *quayside container cranes, dynamics, modelling, moving load, deflections, dynamic amplification factor.*

1. INTRODUCTION

Since its inception about 50 years ago (first quayside container crane was built in January 1959 [16]), the progress of the container industry has been remarkable in many ways. Considering the enormous capital costs, one of the most remarkable changes has been in size of equipment and facilities. Quayside container cranes are used in ports and terminals to transfer containerized cargo to and from ships. Over time, ships size and container weights have increased. Mega container cranes with outreaches of 60-65 meters or more (expected 70 m), lifts above rail of 46 meters (maximum lift height 60 m), trolley velocity 250-300 m/min, hoisting velocity 100 m/min, and lifting capacities of 60-65 (expected 80 t or 120 t) are already built or being built. Present criterion for mega container cranes is to service ships that capacity is 9,000 TEU and at least 20 containers across deck, but expectations for future will require servicing Malacca-Max ships with capacity of even 17,000 TEU and 24 containers across [18]. Comparison between first container crane built in 1959 and mega container crane built in 2004 is presented in figure 1. 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.

There are currently two basic concepts regarding crane structure. The first one is extremely rigid (stiff) structure, with severe stiffness requirements, and the design problems of developing the optimum geometry and stiff forestay concept, where the sag in the forestay contributes to boom deflection. For instance, "Mitsubishi Heavy Industries" cranes have extremely rigid structures with strict deflection requirements in all three directions (perpendicular to gantry rails ~ 4 mm, vertical ~ 128 mm, parallel to gantry rails ~ 49 mm). But American President Lines (APL) specified strength requirements for crane structure, but did not specify deflection limits [14]. APL strategy plans to control the load and accommodate the crane deflections using electronics. This requires more complex software, but will result in a lighter (flexible) crane structure. A detailed structural design process is required to minimize the weight and optimize the geometry and sections. A properly designed monogirder (rectangular or trapezoidal cross section) boom is always lighter than the twin boom. Box beams are simple with only few details controlled by structural fatigue. Truss booms are lighter than box beams but more expensive due to welding costs and with many details controlled by fatigue. To meet strength and particularly fatigue requirements the heavier moving load results in a heavier structure. It is of significant importance for the efficiency of the crane to know in the design process the following dynamic features of the structure [5, 6, and 15]: resonance frequencies and vibration mode shapes, stiffness of whole structure and most important structural parts, behavior of structure during service operation, etc. An effective crane must be designed to suit the present and future needs of the end user.

Figure 1. Growth of quayside container cranes (1959-2004)

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2. DYNAMICS OF MEGA CONTAINER CRANES

The last 40 years has seen mounting interest in research on the modeling and control of cranes [1]. Gantry cranes are usually modeled in two dimensions. The most common modeling approaches are the lumped-mass and distributed mass approach. These models can be distinguished by different complexity of modeling and by the nature of the neglected parameters. Simple models enable easier and more mathematical analysis and give better insight in the design and the possibilities of different control algorithms and leads to better insight in robustness and stability. On the other hand more complex models are necessary to approximate the reality closer. However, it is impossible to include all effects of real life in a mathematical model, for example the flexibility of crane structure will have its influence on the behavior of the controller. In practice it is impossible to do experimental research on a real size crane (only on scale model). To simplify the mechanical model, researchers in the field of crane control are making the assumption that the elastic deformability of the crane can be neglected and it is assumed that all elements of the structure are of the infinite stiffness [2, 8, and 13]. That is incorrect for mega cranes with flexible structure.

Dynamic behavior of a large crane as a movable flexible structure is different than of a smaller crane. If the dynamic properties of the crane structure are unintentionally tuned to dynamic properties of the moving loads (masses), unfortunate responses can occur and have occurred. Vibration is a serious problem in mechanical systems that are required to perform precise motion in the presence of structural flexibility. Examples of such systems are flexible manipulators and container cranes [12]. Productivity has been seriously limited due to excessive vibrations, and millions of dollars have been spent to remedy resonance problems. Residual vibration at the end of a move is the most detrimental and the extent of the residual vibration limits the performance of the system.

Container cranes have more than doubled in outreach and load capacity. This is not easily accomplished given the cantilevered nature of quayside container cranes [4]. 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. Not only is the vibration of the crane unacceptable operationally, it may be unacceptable structurally because of additional fatigue damage.

There are only few scientific papers concerning dynamics and structural behavior of quayside container cranes. One of the main reasons is that these machines are not so widely used as standard overhead traveling cranes. Other reasons can be found in the fact that concept of mega container cranes is still in progress and all dynamic problems could not be considered to the full. Here will be given the very short survey of some important papers concerning dynamics and structural vibrations of large quay container cranes.

In the researches presented in [5, 6] experimental and numerical dynamic analysis of container crane structure with optimal dynamic characteristics has been done. It is noticed that dynamic features as a result of optimization should be as high as possible natural frequencies, particularly ones most important for service operation (operation of hoisting mechanism and trolley motion) and minimal weight of structure. The most important conclusions are that experimentally measured values of crane natural frequency are in very close agreement with numerical results of the Finite Element Method mechanical model. The first numerical results, prior to the experiment, were notably different only for the fourth vibration mode (not important for analysis of crane behavior during service operation). Their conclusions are: Reliable mechanical dynamic model of a container crane and related structures which are in general steel-box-beam welded structures can be successfully made on the basis of FEM; Beams and girders of such structure can be appropriately modeled with line beam elements, while the Timoshenko beam model can produce slightly better results than the Euler-Bernoulli model; Geometric nonlinearity gives better results, because it is more realistic, but the error of the first order theory is relatively small, so that taking geometric nonlinearity into account is preferable, but not absolutely necessary;

3. DYNAMIC INTERACTION BETWEEN TROLLEY AND STRUCTURE

The need for fast and safe containers loading and unloading from ships in order to minimize service time requires a control of the crane motion that optimises the crane's dynamic performance. The two-dimensional cycle is divided in three motions: load hoisting, transfer and load lowering. The problems are the reduction of the total time of load transport (time-optimal trajectory control) and reduction of the swing of the load at its end position, including accurate positioning of the load. The sway is caused by dynamic interaction between the trolley drive and the structure. The continuing trolley operation causes the girders and gantry frame deflections. A validated math model is necessary for conducting detailed study of the dynamic behaviour of the container crane and control schemes, because structural control problems arise from the various dynamic problems. The basic outline of dynamics of quay container cranes is shown in figure 2 [7, 17].

It is important to know that two basic structural phenomena in quayside container crane dynamics, i.e. vertical vibrations of girder due to the motion of trolley with load – moving load problem, and excessive sway of moment resistant gantry frame in the trolley travel direction, are both consequence of dynamic interaction between trolley, hanging load and crane's supporting structure. These two problems are the most significant low frequency–large displacement dynamic problems with the entire quayside crane structure, and could be solved in practice by applying structural control to suppress vertical vibrations of girder, and excessive sway of gantry frame.

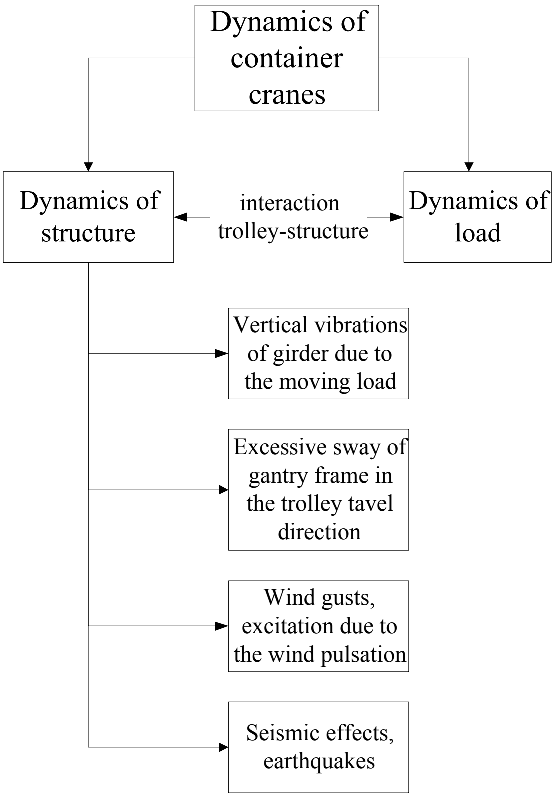

Figure 2. Basic outline of container crane dynamics

4. MOVING LOAD PROBLEM

Vertical vibrations of crane girders are the consequence of moving load, i.e. trolley with load. Vertical vibrations are important for electronic load control. To solve the problems of vertical vibrations it is necessary to apply theory of structural vibrations due to the moving load, and use analytical models, or use Finite Element Method. Today, most of the software for FEM analysis already have module for moving load analysis, but they are solving problem quasi-statically, with no possibility of obtaining dynamic factors. Analyses of structures subjected to moving loads have been carried out, at first, for the railways purposes, but in the years after became more general for various purposes in dynamics, and was used for several structures subjected to moving loads as guide-ways, bridges, cableways, and of course, cranes [9, 10]. The moving load problem focuses on structure – vehicle interaction. In figure 3 is presented an analysis of vehicle models examples [11]. The physical approach is the vehicle traveling at speed "v", and can be modeled as suspension model (b), moving mass model (c), or moving force model (d). As the trolley construction for cranes is rigid as a rule, the suspension model (b) can be momentary discarded. The dilemma is which of two models (c and d) is more appropriate for the case of crane trolley.

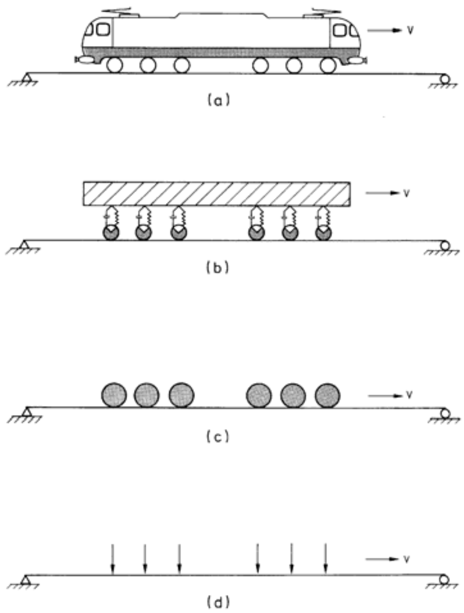

Figure 3. Basic outline of moving load modeling

5. MATHEMATICAL MODEL

This work discusses influence of trolley motion to dynamic behavior of quayside large (mega) container cranes for servicing container ships with 24 containers across beam (Malacca-max container ship). It is ascertained that the outreach (boom), due to its large dimension and mass, 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 water side leg of the crane and trolley as a moving load, i.e. trolley impact on the change of maximum values of deflections and bending moments.

According to the proposed and adopted concept of flexible structure of crane, an original mathematical model is set up. The first step in moving load analysis is to make a reliable FEM model, and to find corresponding natural frequencies for dynamic analysis. In figure 4 is shown FEM models for the adopted structure of quayside crane [15].

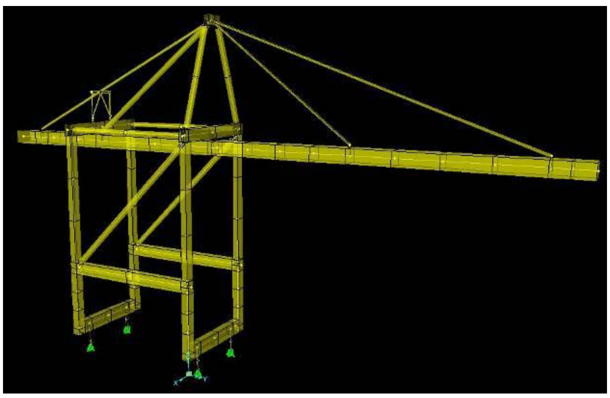

Figure 4. FEM model of mega quayside container crane

It is observed that vibration of the boom on water side leg in vertical direction perpendicular to trolley direction are practically independent from other structural parts, and this vibration is recognized as one between first three vibrations with lowest frequencies most important for dynamic analysis, figure 5 [15].

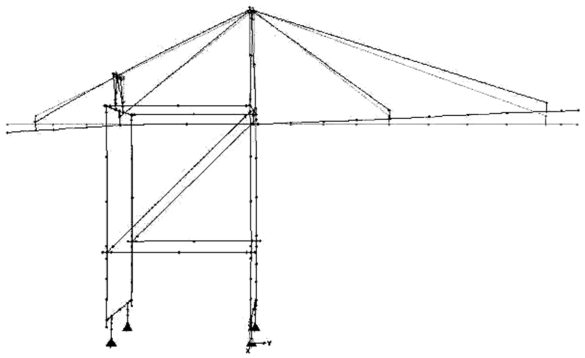

Figure 5. Vibrations of crane in vertical direction, *f* = 1,568 Hz

In the next stage of modeling process, consisting of several intermediate stages by making appropriate procedure for dynamic modeling of structure, we obtain the idealized equivalent reduced model of boom competent for writing differential equations for moving load problem. Relative deviation of natural lowest frequency of vibration for idealized dynamic model is less than 1.4% in comparison with FEM model of whole structure. Reduced model comprises components as linearized springs and lumped masses, but the mass of the boom is modeled as distributed one. The selected quay crane has two forestays, outer and inner, and the procedure of finding equivalent dynamic model is presented in figures 6 and 7 [15].

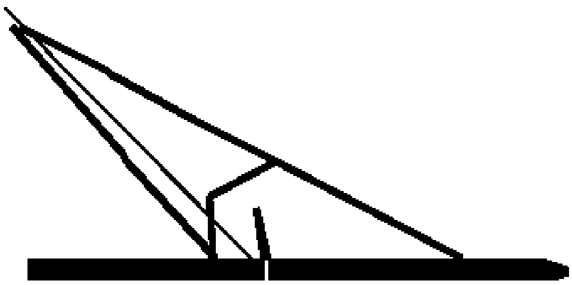

Figure 6. Model of boom with two stays

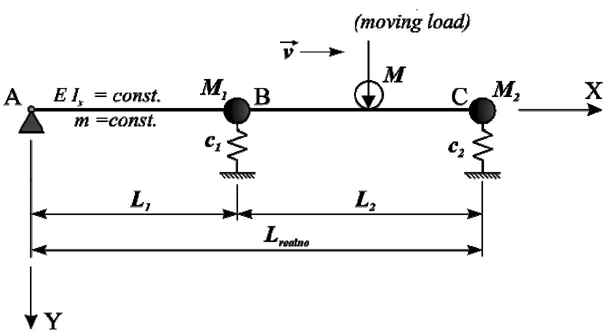

Figure 7. Equivalent model for moving load analysis

This model is competent for analysing loading/unloading of a ship cell, as is shown in Figure 8.

Figure 8. Loading a ship´ s cell

The problem of moving load is treated in the analysis as a moving mass problem, i.e. the inertia of the trolley mass has not been neglected. Differential equations of motion are obtained from Lagrange's equations by using Assumed Modes Method in such a way that continuum is discretized by finite number of admissible functions and with neglecting dissipation function (damping). Selection and estimation of admissible functions is done by using variational approach. Mathematical model of moving mass includes in itself influence of moving mass inertia, influence of Coriolis centripetal force, and influence of deceleration of moving mass (braking). Deflection of the slope is assumed in the shape [15]:

$$y(x,t)=\sum_{i=1}^{n}\phi_i(x)\cdot q_i(t)\,, \tag{1}$$

where the assumed and admissible functions are

$$\begin{aligned}&\phi_1(x)=\frac{x}{L},\quad \phi_2(x)=\sin\frac{\pi x}{L},\quad \phi_3(x)=\sin\frac{2\pi x}{L},\\&\phi_4(x)=\sin\frac{3\pi x}{L},\quad \phi_5(x)=\sin\frac{4\pi x}{L}\ .\end{aligned} \tag{2}$$

And $q_i(t)$ are generalized coordinates – displacements.

Differential equations for the system presented in Figure 7 are finally:

$$\begin{aligned}&\sum_{j=1}^{5}\Big[m_{ij}+M\phi_i(s)\phi_j(s)\Big]\ddot{q}_j(t)+\\&+\sum_{j=1}^{5}\Big[2Mv\phi_i(s)\phi_j^{'}(s)\Big]\dot{q}_j(t)+\\&+\sum_{j=1}^{5}\Big[c_{ij}+Mv^2\phi_i(s)\phi_j^{''}(s)+Ma\phi_i(s)\phi_j^{'}(s)\Big]q_j(t)=\\&=Mg\phi_i(s)\,,\ i=1,2,3,4,5\,.\end{aligned} \tag{3}$$

The system of differential equations was finally written in a matrix form and solved numerically by using Runge-Kutta method of V order (Method Runge-Kutta-Fehlberg), and by using program written in C++. For the crane with outreach of 65.8 m (between the hinge and end real position of trolley), the values of deflections under moving mass depending on the number of admissible functions ($n = 2,3,4,5$) are shown in figure 9. Fast and good convergence of solution for adopted 5 admissible functions is fulfilled.

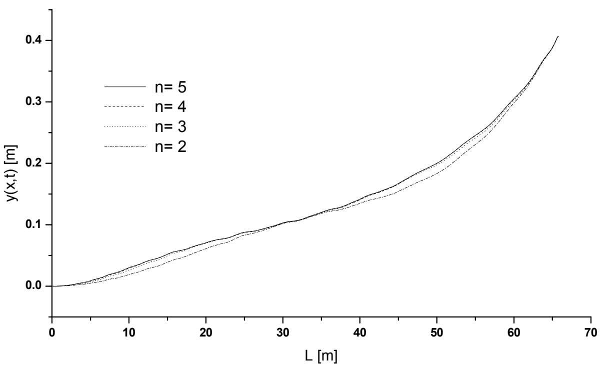

Figure 9. Deflection of boom under moving mass

The change of bending moment of boom under the moving mass depending on the number of adopted admissible functions ($n = 2,3,4,5$) is shown in Figure 10.

In figure 11 is presented change of dynamic amplification factor of deflection at the free end of outreach due to the motion of moving mass. Maximum obtained value of dynamic amplification factor of deflection is 1.137, i.e. absolute increase of 13.7% with respect of static deflection. This value could be compared with those recommended and used by various design codes and standards defining maximum wheel loads for calculation of crane runway. For instance code AC-317 proposes the value of 1.4 (trolley and spreader) and 1.7 (lifted load) for dynamic amplification factor. It is conclusive that mentioned values recommended by design codes are too high, and they should be revised in order to minimize their values.

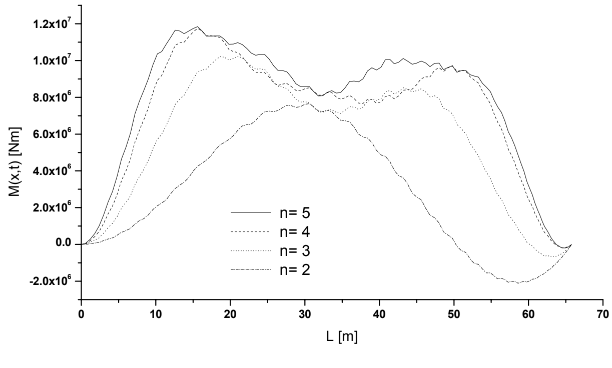

Figure 10. Values of bending moment under the moving mass

Values of acceleration of moving mass in vertical direction are presented in figure 12. Maximum value of acceleration is obtained to be equal 0.0165 g. This value belongs to the class of clearly appreciable accelerations in the frequency range between 1 and 10 Hz [3], and it is not as high to be disturbing or unpleasant for crane operator.

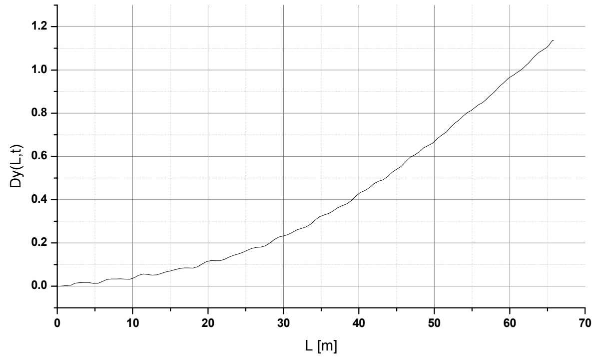

Figure 11. Dynamic amplification factor of deflection for free end

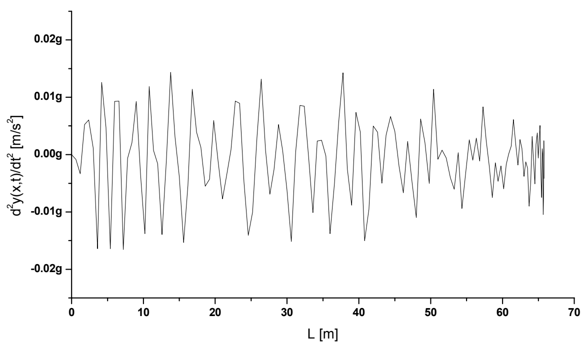

Figure 12. Acceleration of moving load in vertical direction

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