ESTIMATION OF THE DYNAMIC EFFECT IN THE LIFTING OPERATIONS OF A BOOM CRANE

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KEYWORDS

Lifting devices, overloading condition, vibration, motion command.

ABSTRACT

This paper describes a model for the study of the dynamic behavior of lifting equipments. The model here proposed allows to evaluate the fluctuations of the load arising from the elasticity of the rope and from the type of the motion command imposed by the winch. A calculation software was developed in order to determine the actual acceleration of the lifted mass and the dynamic overload inside the rope during the lifting phase. In the final part of the article an application example is presented, with the aim of showing the correlation between the magnitude of the stress and the type of the employed motion command.

INTRODUCTION

As is well known, the application of a load to a structure generates vibratory effects on the structure itself. This aspect is of a general nature and obviously it does not consider the type of structure and the rules for the load application, aspect, the latter, that influences the magnitude of the dynamic instead.

In a lifting equipment (cranes, bridge cranes, mobile cranes, etc..) this phenomenon appears to be particularly important, since it generates actions whose wrong assessment could compromise the structure, in particular as regards the phenomena of maximum stress, buckling, fatigue and equilibrium of the structure itself. Large bibliography of this subject has been elaborated in recent years (Bao, Zhang and Zhu 2011), as the problem, albeit overlooked in past years, now plays an important role during the design stage of a lifting device. To confirm what we said just think that the real force acting on the structure as a result of lifting can be equal to $1.5 \div 2$ times the nominal force generated from the hoist.

With the need to reduce the weight and increase the performances (see for example the mobile cranes where the frame is also more lightweight), this load increment cannot be absolutely neglected (Matteazzi and Solazzi 2000) (Matteazzi and Solazzi 2005).

It is important to observe that the value of the dynamic effect is about equal or even greater to the value of the safety factor (adopting a deterministic design methodology) and therefore its wrong estimation may lead to the design of a non-safe device. Another fundamental aspect resides in the fact that the dynamic effect increases not only the action, but also the maximum number of stress cycles to which the device is subjected; as is well known, such cycles, while being of limited amplitude, contribute significantly to the fatigue life of the machine.

The basic parameters of the stress cycle as the amplitude (related to the intensity of the dynamic effect), the numerosity (related to the damping of the structure) and the temporal sequence (related to the sequence operations of lifting and handling of cargo) are variables which cannot be ignored (Solazzi 2004). In the face of these problems, regulatory agencies have proposed proper coefficients that, multiplying the load to be moved, allow the designer to consider these phenomena in the calculation.

Starting from the studies reported in the literature and from the analysis of the Standards for the design of lifting equipments, we developed a mathematical model that allows to estimate the load fluctuations caused by the elasticity of the rope and by the elasticity of the structure, in order to evaluate or estimate the dynamic overload inside the rope during lifting.

A BRIEF OVERVIEW OF INTERNATIONAL STANDARDS ABOUT LIFTING DEVICES

From the regulatory point of view, there are several Standards for the design of a lifting device. All introduce a coefficient of dynamic overload that depends on the class of the structure. This class is defined in relation to the number of stress cycles to which the structure is subjected, to the spectrum of the load and to the lifting speed. Figure 1 shows an example of the dynamical effect, evaluated by experimental test on a working platform (Solazzi 2009; Solazzi and Scalmana 2012). In general is possible to observe that at the beginning or at the end of the movement, the structure is subjected to oscillations and thus to accelerations that increase the load acting on

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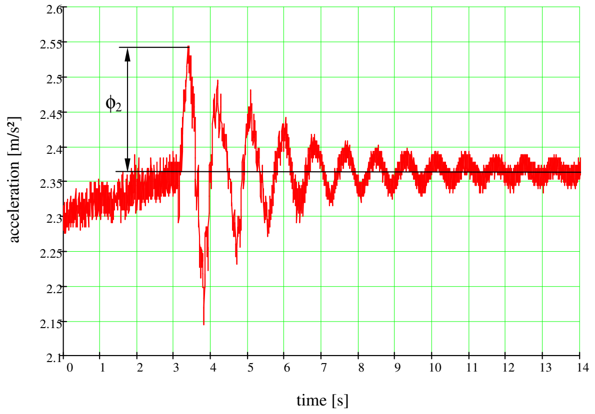


Figure 1: Dynamic overloads on a working platform in correspondence with a stoppage of the lifting motion.

the structure itself. Table 1 shows some values defined by these Standards, where β_2 is a parameter that depends on the hoisting class, V_0 is the velocity of lifting and ϕ_2 is the dynamical overloading.

Table 1: Dynamic effect as a function of the Standard.

UNI 9309 (1988)	$\phi_2 = \phi_{2\,min} + \beta_2(V_0 - 0.2)$ $V_0 > 0.2$ m/s	
Hoisting Class (β_2)	$\phi_{2\,min}$	$\phi_{2\,max}$
0.2	1.00	1.30
0.4	1.05	1.60
0.6	1.10	1.90
0.8	1.15	2.20
DIN 15017 (1975)	$\phi_2 = \phi_{2\,min} + \beta_2 V_0$	
Hoisting Class (β_2)	$\phi_{2\,min}$	$\phi_{2\,max}$
0.132	1.00	1.30
0.264	1.20	1.60
0.396	1.30	1.90
0.528	1.40	2.20
UNI EN 13001 (2005)	$\phi_2 = \phi_{2\,min} + \beta_2 V_0$	
Hoisting Class (β_2)	$\phi_{2\,min}$	
0.17	1.00	
0.34	1.05	
0.51	1.10	
0.68	1.15	

For the design and testing of lifting equipments the coefficients associated to the dynamic effect are in some cases higher than the normal safety factors defined by the Standards. Moreover these values of ϕ_2 do not depend neither on the stiffness of the structure nor on the type of motion law, aspects that can strongly affect the amount of overload.

MATHEMATICAL MODEL

In most cases of practical interest, elasto-dynamic models with one or two degrees of freedom (DOF) allow to simulate with good approximation the dynamic behavior of lifting devices. The use of a 1-DOF model can be justified in the case where the structure is very rigid. If the structure presents a stiffness comparable to the lifting system, as in the example presented in this work, a 2-DOF modelling is recommended (Matteazzi and Minoia 2007).

Figure 2 shows in schematic form a model with two degrees of freedom that can be used to analyze the dynamics of a lifting device.

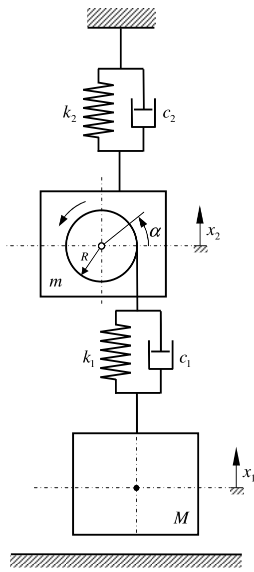


Figure 2: Two degrees of freedom model of a lifting device.

The vertical displacement of the load is represented by the coordinate x_1, while the vertical displacement of the lifting structure is indicated by the coordinate x_2. The symbols M and m respectively indicate the mass of the load and the mass of the lifting structure. The stiffness and the damping constant of the lifting system (ropes) are indicated by the symbols k_1 and c_1; in a similar way, the symbols k_2 and c_2 indicate the values of stiffness and damping of the structure. The rope is wound on a drum of radius R, driven by a geared motor unit; the model proposed here assumed to know the drive law of the drum i.e. the function $\alpha = \alpha(t)$. Figure 3 are represented the free body diagrams of the two masses, in order to highlight the forces that are generated during the lifting phase. In the picture the forces due to gravity were not represented because such forces are already balanced by the elastic reactions due to static deformations of the system. With the sign conventions shown Figure 3 we can write the following system of equations:

$$\begin{cases} M\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = R(k_1\alpha + c_1\dot{\alpha}) \\ M\ddot{x}_1 + m\ddot{x}_2 + c_2\dot{x}_2 + k_2 x_2 = 0 \end{cases}$$

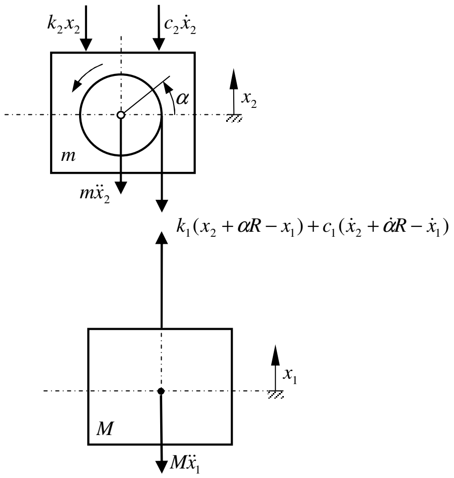


Figure 3: Free-body diagrams.

The above equations have been implemented in a software (Mathcad) and were solved numerically by going to determine variables x_1 and x_2 for example described in the following paragraph. As stated, the model here described requires to know the function $\alpha = \alpha(t)$ that defines the motion of the drum during the lifting phase. From a practical point of view, this motion law may be experimentally measured by mounting an angular displacement or a velocity transducer (encoder or tacho) on the axis of the drum.

Bearing in mind that, in most cases, the three-phase induction motor that drives the winch is controlled by a frequency converter, a device that allows you to adjust with a good approximation the acceleration and deceleration time intervals, it seems reasonable to adopt a law of motion with acceleration and deceleration that is easily definable in analytical form.

Therefore, for the purposes of the simulation of the dynamic behaviour of the system, we can assume valid the diagram represented in Figure 4, which can be immediately determined based on the knowledge of four parameters: the maximum speed $\dot{\alpha}_{max}$ and the three time intervals t_1, t_2 and t_3, which respectively represent the time duration of the phases of acceleration, constant speed and deceleration.

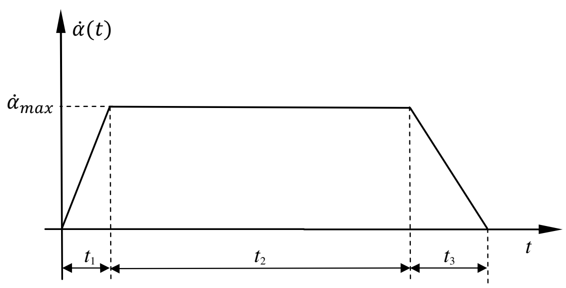


Figure 4: Angular velocity of the drum versus time.

APPLICATION OF THE MODEL: A BOOM CRANE

This section shows an example of an application of that previously reported. We have considered a boom crane with a load capacity of 50 tons and a reach of 80 m, designed and engineered according to the different dedicated Standards. It is the typical crane used in ports for loading and unloading of containers from ships. We made solid model and then the FEM model of the structure with the purpose, on one hand, to verify the crane and secondly to determine the displacement and its stiffness in different load configurations. Figures 5 and seq. report the design of the structure and its maximum displacement in the different load configurations typical for this kind of cranes. In particular, Figure 5 shows the displacements in static configuration (displacements induced only by own weight), Figure 6 shows the total and vertical displacements in the case where the load is positioned on the seaside on the ship and Figure 7 shows displacements in the case in which the load is positioned on the quay side.

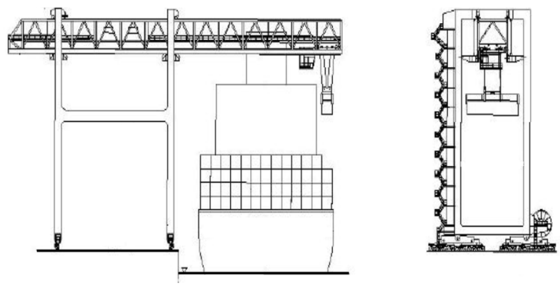


Figure 5: The retractable boom crane studied: boom length: 80 m, seaward side: 38 m, gantry width: 27 m, height (under hook): 45 m.

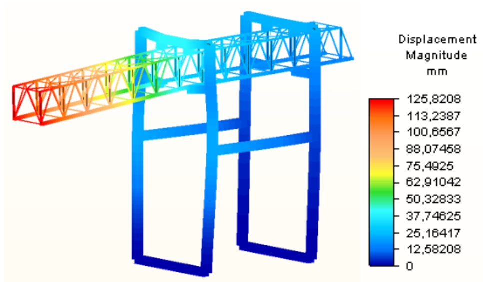


Figure 6: Structure displacement due to gravity effect.

The values of displacement and stiffness found are summarized in Table 2.

Table 2: Displacements and stiffness of the structure.

Load position	Displacement [mm]	Stiffness k_2 [N/mm]
Sea side	143.2	4190
Quay side	34.4	17450

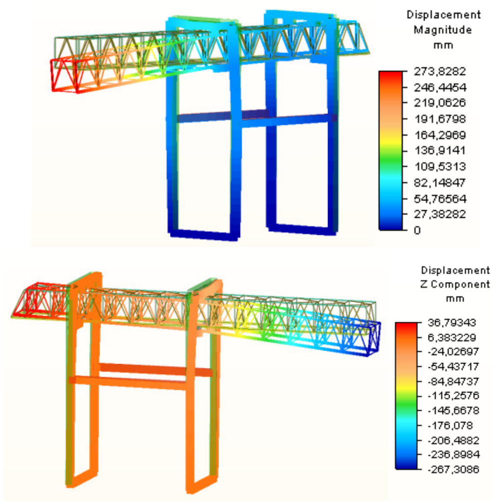


Figure 7: Structure displacement: load lifting from the seaward side: a) total displacement; b) vertical displacement.

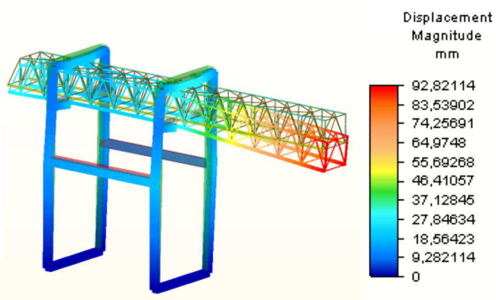


Figure 8: Structure displacement: load lifting from dockside.

These values correspond also to a minimum and a maximum stiffness configuration for the structure.

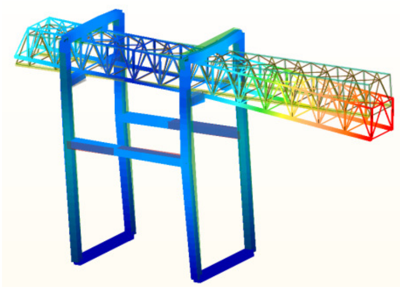


Figure 9: Trend of displacement of the crane relative to the first mode of vibration in vertical direction.

In parallel to the structural design a lifting system composed of a block and some pulleys has been employed.

In particular, we chose a system made by four pulleys (8 branches of rope 24 mm in diameter) and one made by five pulleys (10 branches rope 22 mm in diameter). On the basis of cables commercially available (in particular Warrington Seale type), we calculated both the stiffness of each section that composes the lifting system and the package consisting of all the branches.

During development, we considered two different manufacturers of ropes, but we found very similar results in terms of stiffness. Because in practice, in the specific case of the container of 50 tons, the operation of lifting the load takes place at a variable height in relation to the position of the container that has to be moved on the ship, we performed a procedure in order to calculate the equivalent stiffness of the structure in relation both to the system lifting (8 or 10 branches) and to the height of the load (see Table 3).

Table 3: Displacements and stiffness of lifting system related to system configuration.

Ropes Nr.	Rope diam. [mm]	Lifting height [mm]	Displacement [mm]	Single rope Stiffness [N/mm]	Total Stiffness k_1 [N/mm]
8	24	65000	159.6	470	3760
8	24	45000	110.5	680	5430
8	24	10000	24.6	3050	24430
10	22	65000	152.0	400	3950
10	22	45000	105.2	570	5700
10	22	10000	23.4	2570	25660

To determine the mass of the lifting system involved in vibration, we performed a modal analysis considering the lifting of the load on both the sea and the port side and we calculated the natural frequencies of the system together with their mass participation. Figure 8 shows the displacement of the crane relative to the first mode of vibration. The first frequency found was about 1.5 Hz and. Being this value very low, we can already observe that the dynamic actions induced for example by an earthquake have a limited importance in structural design and verification of the crane (Solazzi 2011; Solazzi 2012). The value of the modal mass found for that configuration is equal to 64 tons, approximately 21% of the entire mass of the structure. By adding this value to that of the mass of the lifting device we determined the mass m necessary for the mathematical model. This mass resulted equal to 80 tons. On the base of the FEM results obtained, we performed a dynamic analysis using the model defined above. We chose a configuration such that the lifting speed was set on about 0.25 m/s and the time needed to get the speed regime condition was set on 4 sec.

The masses M and m, defined and calculated, were equal to $M = 50\,\text{tons}$ and $m = 80\,\text{tons}$. The constants c_1 and c_2 have been fixed on $5 \times 10^4\ \text{Ns/m}$.

Note that, in general, the damping values of c_1 and c_2 are strongly variable in relation to the type of structure (welded or bolted) and also to the type of rope (internal damping). In this work they were derived from the results of experimental tests by using the logarithmic decrement methodology (see Figure 1). The following graphs show the results in terms of speed and acceleration of the load and the structure (crane + lifting system).

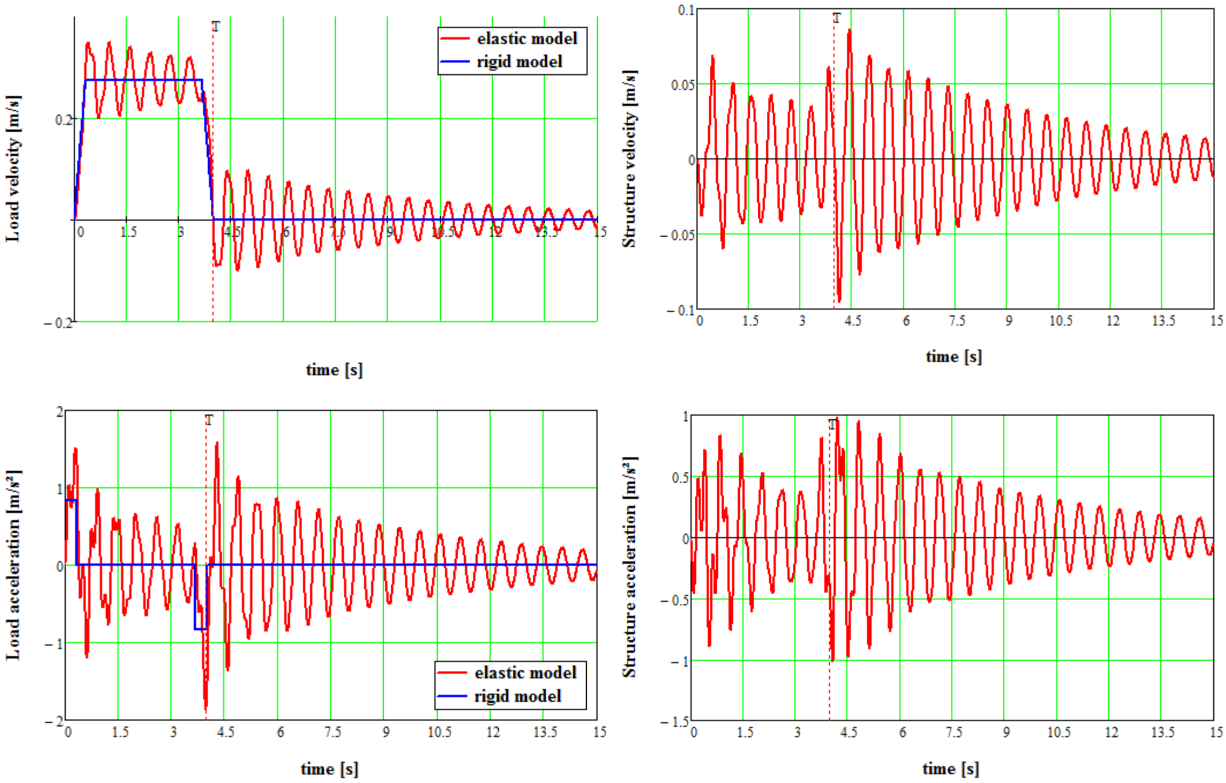


Figure 10: Load and structure velocity and acceleration in maximum stiffness condition.

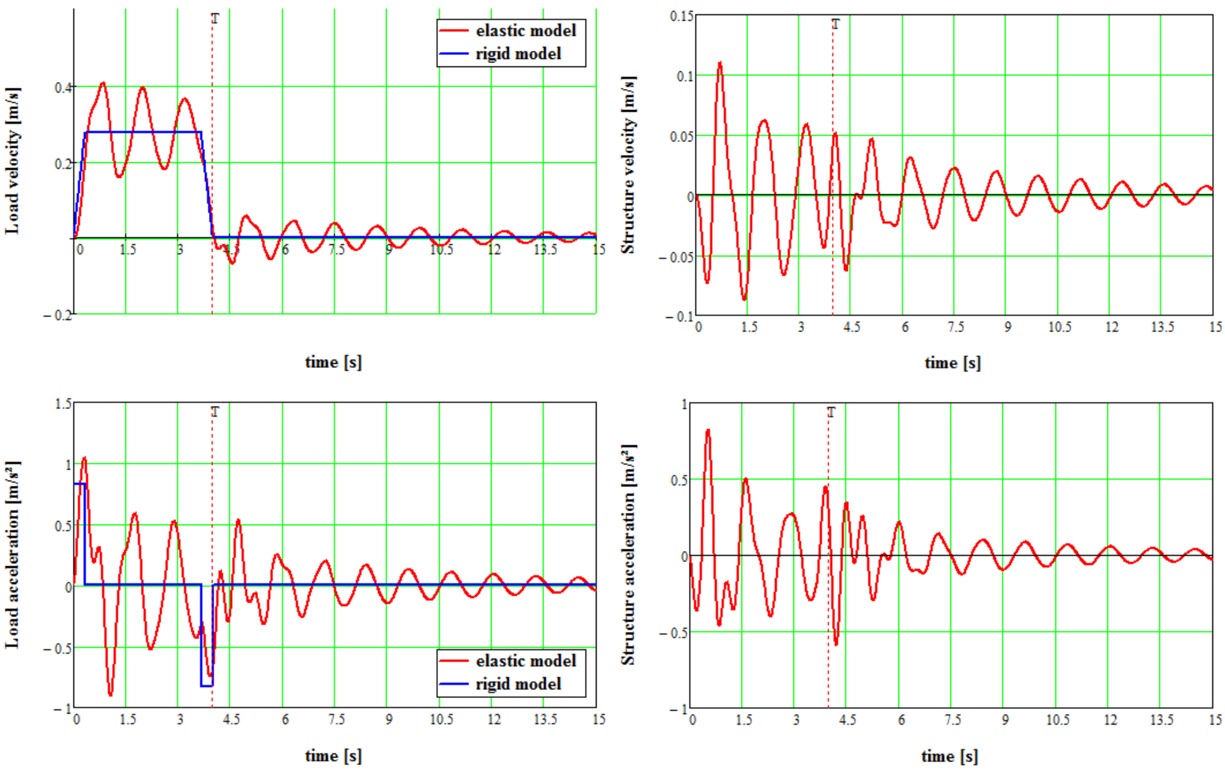


Figure 11: Load and structure velocity and acceleration in minimum stiffness condition.

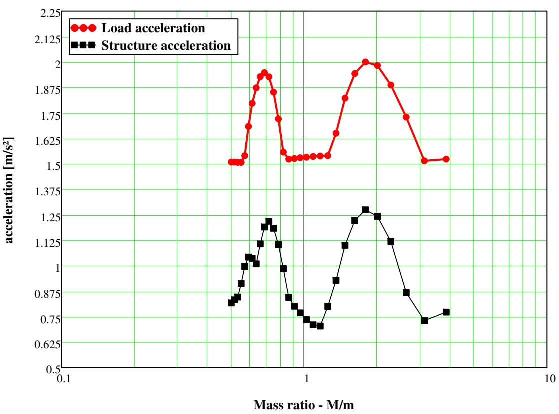
Among the configurations of stiffness available we chose the two most significant and useful to make consideration about the dynamic behavior of the crane:

- the condition of maximum stiffness for the structure and for the lifting device ($k_1 = 25660\,\mathrm{N/mm}$; $k_2 = 17450\,\mathrm{N/mm}$, Figure 10);
- the condition of minimum stiffness for the structure and lifting device ($k_1 = 3760\,\mathrm{N/mm}$; $k_2 = 4190\,\mathrm{N/mm}$, Figure 11).

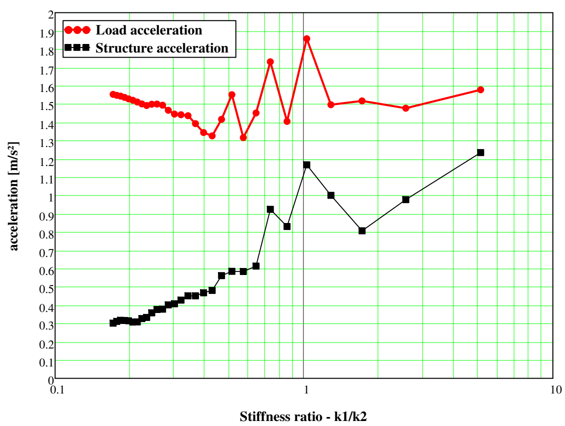
Comparing these graphs with experimental data it is possible not only to calibrate the model, but also to derive useful information for the crane design.

The natural frequencies of vibration of the structure are related to the stiffness and both the masses of the system with two degrees of freedom; this aspect also has important implications for the phenomenon of fatigue for the materials.

Since the purpose of this work is to evaluate the effect of overstressing induced by the operation of loading we also present in Figure 12) two plots that put in relation the acceleration of the load and the structure as a function of the ratio M/m and of the ratio k_1/k_2. In particular for the Figure 12a the configuration is: $M = 50\,\text{tons}$, m = variable, $k_1 = 25660\,\mathrm{N/mm}$, $k_2 = 17450\,\mathrm{N/mm}$ and for the Figure 12b the configuration is: $M = 50\,\text{tons}$, $m = 80\,\text{tons}$, $k_1 = 25660\,\mathrm{N/mm}$, $k_2 = variable$).



a)



b)

Figure 12: Load and structure acceleration vs M/m (a) and vs k_1/k_2 (b).

As can be seen in more than one configuration the values found are higher than those considered in the Standards. An incorrect account of the same therefore could mean an higher risk for the structure both in terms of local or global buckling and in term of overturning.

CONCLUSIONS

An analysis of the dynamic effects in lifting devices has been presented in the paper. After an initial overview of the Standards in this field, we developed a lumped-parameter model with two degrees of freedom and we have applied it to the design of a boom crane, for quantifying the dynamic response. Through FEM simulations, we determined the stiffness of the structure in different geometric configurations. By the design of the lifting system (composed by ropes and drum) we evaluated the relation between stiffness and type of rope and number of branches. These parameters have been used and implemented in the calculation model and we determined the dynamic effect of the lifting of the load on the entire structure. We found that this effect, absolutely not negligible as regards the strength of the material and the crane stability, is also strictly dependent from the combination of structure and lifting system and from the mode of operation of the load. This procedure allows to estimate and compare in a very simple way all possible configurations of stiffness, mass, load velocity, etc. and to choose the best for a precise application. However, given the importance of this phenomenon, this work is the first phase of a larger project that will see the application of the model to other lifting equipment and its optimization through other rules of load movement.

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