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THE INFLUENCE OF STABILIZATION OF LOAD POSITIONING IN AN OFFSHORE CRANE ON TAUT-SLACK PHENOMENON IN A ROPE

The peculiarity of offshore cranes, i. e. cranes based on ships or drilling platforms, is not only a significant motion of their base, but also the taut-slack phenomenon. Under some circumstances a rope can temporarily go completely slack, while a moment later, the force in the rope can increase to nominal or even higher value. Periodic occurrence of such phenomena can be damaging to the supporting structure of the crane and its driver. In the paper, mathematical models of offshore cranes that allow for analysis of the taut-slack phenomenon are presented. Results of numerical calculations show that the method of load stabilization proposed by the authors in their earlier works can eliminate this problem.

1. Introduction

The authors of this paper have been interested in problems concerning load stabilisation of offshore cranes for a long period of time. In contrast to overland cranes, the base of offshore cranes is usually subject to considerable flotation motions due to the waving of the sea. Offshore equipment is mounted either on a ship deck or on a platform. Movements of offshore crane bases cause that the load sways significantly, even when the crane drivers do not perform any working movements. The load swaying makes reloading and construction work not only more difficult but also constitutes a major danger to the staff and the equipment itself. Namely, the load can hit the side of the ship or the platform structure causing considerable damage. A method of

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stabilization of the load positioning for offshore cranes with a moving base has been presented in [1].

Contacts with international researchers induced the authors to take a closer interest in the taut-slack phenomenon. This phenomenon concerns a situation when, under some conditions, a rope can temporarily go completely slack, while a moment later, force in the rope can increase to a nominal or higher value.

A major feature of strings, and also ropes used in offshore cranes, is the lack of ability to transfer compression load. The characteristic of string stiffness have discontinuous first derivative at zero point. An analysis of dynamics in this range leads to nonlinear relations. For overland cranes, during normal use, the phenomenon when a rope is completely slack is hardly ever observed. A different situation exists in the case of offshore cranes, especially when a load is under the water surface. Forces of resistance acting on the load are much greater when the load is moving in the water than when it is moving in the air. The forces depend on the shape of the load as well. They have a significant influence on the force in the hoisting rope, causing big oscillations of its value. Under specific conditions of periodical movement of the load, induced by the movement of the crane base (caused by the waving of the sea), the rope can be also periodically completely slack. The moment when the rope is slack is followed by the rope being taut, which can lead to considerable stress far above the nominal stress coming from the load weight. In practice, offshore rope-load systems should work under a stress lower than permissible, and simultaneously zero stress should not occur [2]. In Fig. 1, an example of the taut-slack phenomenon is presented. This is a graph of force in the hoisting rope of an offshore crane. Time durations denoted as ”T” are taut phases and denoted as ”S” are slack phases. Additionally, the phase when the value of the force is equal to zero is described as ”S_0”. Many scientific publications deal with maximal stress in the hoisting rope. Fewer papers treat zero stress problems or full taut-slack phenomenon [2-4].

In this paper, two models of offshore cranes presented in [5] have been used. A simplified one, with three degrees of freedom, serves to determine the drive function of the hoisting drum that enables stabilization of the load position. A more complex one, called the basic model, is intended for more accurate dynamic analysis. In both models, non-zero tension stiffness and zero compression stiffness of the hoisting rope have been assumed. As a result, the models enable us to analyse the taut-slack phenomenon. The experience of many authors shows that the most important ship movement for dynamic analysis is vertical motion in the plane containing a symmetrical axis [2, 6-8]. That is why in the paper numerical analysis is limited to the heave (vertical) motion of the crane base.

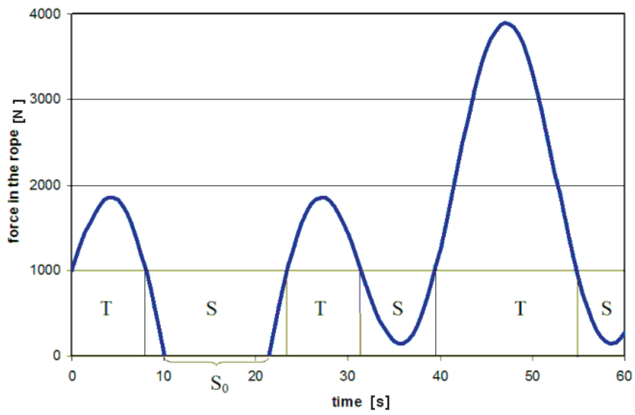

Fig. 1. Taut-slack phenomenon

2. Model of an offshore crane

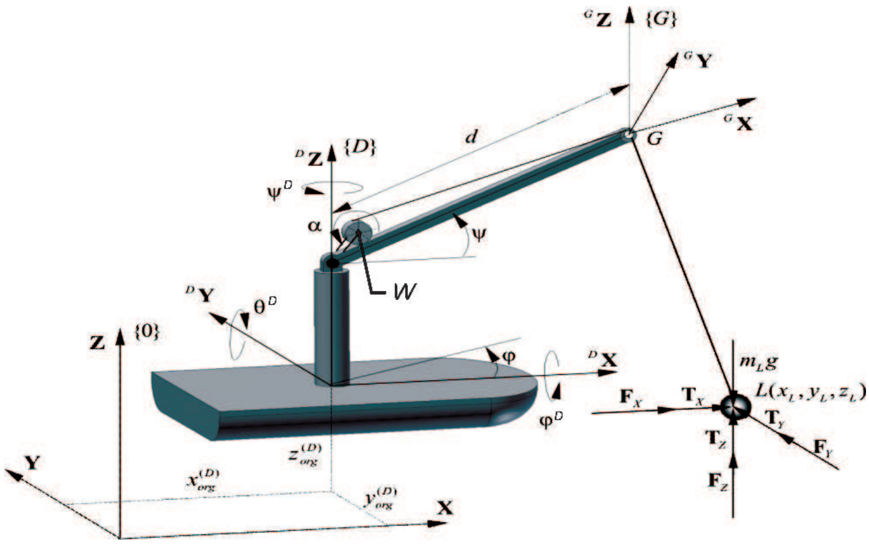

Fig. 2. Scheme of the model of an offshore crane

Because offshore cranes may work with considerable length of the hoisting rope, flexibility and damping in the rope have been taken into account in the modelling process. It has been assumed that the stiffness coefficient depends on the rope length and it is defined by the formula:

$$c = \frac{EF}{L_0 - \alpha r_D}, \tag{1}$$

where

$L_0 = L_{G0} + i_w L_{L0}$ – initial length of the rope,

L_{G0} – initial length of the segment WG (before loading and starting working motion) – Fig. 2,

L_{L0} – initial length of the segment GL (before loading and starting working motion) – Fig. 2,

i_w – the ratio of the pulley block,

α – the angle of rotation of the hoisting drum (time function),

r_D – the radius of the hoisting drum,

E – Young's modulus of rope material,

F – the rope cross-section.

The value of the force in the rope has been calculated as:

$$S = \delta\left(c\Delta + b\dot{\Delta}\right), \tag{2}$$

where

$$\delta = \begin{cases} 0 \text{ when } \Delta \le 0 \\ 1 \text{ when } \Delta > 0 \end{cases},$$

$\Delta, \dot{\Delta}$ – deformation and deformation velocity of the rope, respectively,

b – the damping coefficient of the rope.

In Fig. 2, $\mathbf{F}_X, \mathbf{F}_Y, \mathbf{F}_Z$ signifies external forces acting on the load, for example buoyancy forces or pressure forces induced by sea currents, $\mathbf{T}_X, \mathbf{T}_Y, \mathbf{T}_Z$ signifies damping forces. Table 1 compares the basic properties of both models used in the paper.

Table 1.

Comparison of models: simplified and basic

	Simplified model	Basic model
Form of description for points in the system	absolute coordinates	relative coordinates
Method of obtaining equations of motion	Newton's second law	Lagrange's equations of second order
Number of degrees of freedom	3	$12 + m$ where m is the total number of modes considered in the model of the jib
Drives considered	1. hoisting winch drum	1. hoisting winch drum 2. of slewing of the crane's upper structure 3. of reach changing (reach changing actuator)
Drive modelling method	kinematic driving	kinematic driving by a parallel system of a spring and a damper

cd. Table 1.

Pedestal	modelled as a rigid body, stiffly fixed to the base (ship's deck)	
Load	modelled as a concentrated mass	
Rope	flexible with damping	
Jib	rigid	capable of flexing – the jib has been discretized using the modal method
Description of base's motion	pseudo-harmonic: $\beta_i^{(D)} = \sum_{j=1}^{n_i^{(D)}} A_{i,j}^{(D)} \sin(\omega_{i,j}^{(D)} t + \varphi_{i,j}^{(D)})$ for $i = 1,\ldots,6$,, where: $\begin{bmatrix} \beta_1^{(D)} & \ldots & \beta_6^{(D)} \end{bmatrix}^T = \begin{bmatrix} x_{org}^{(D)} & y_{org}^{(D)} & z_{org}^{(D)} & \varphi^{(D)} & \theta^{(D)} & \psi^{(D)} \end{bmatrix}^T = \boldsymbol{\beta}^{(D)}$, $A_{i,j}^{(D)}$ – j^{th} amplitude in the i^{th} direction, $\omega_{i,j}^{(D)}$ – j^{th} angular frequency in the i^{th} direction, $\varphi_{i,j}^{(D)}$ – j^{th} phase angle in the i^{th} direction.	
Equations of motion	$\begin{cases} m_L \ddot{x}_L = S\dfrac{x_D - x_L}{L_{DL}} + F_X + T_X \\ m_L \ddot{y}_L = S\dfrac{y_D - y_L}{L_{DL}} + F_Y + T_Y \\ m_L \ddot{z}_L = S\dfrac{z_D - z_L}{L_{DL}} + F_Z + T_Z - m_L g \end{cases}$ where: $S = c\,(L + L_{L0} + \alpha \cdot r_B) + b_l \left(\dot{L} + \dot{\alpha} \cdot r_B\right)$ $L^2 = (x_G - x_L)^2 + (y_G - y_L)^2 + (z_G - z_L)^2$	Described later
Integration method for the equations of motion	Runge-Kutta method of fourth order	

The equations of motion in the basic model have been derived along the lines of [6] and [9]. Lagrange's equations of the second order have been used:

$$\frac{d}{dt}\frac{\partial E}{\partial \dot{q}_k} - \frac{\partial E}{\partial q_k} + \frac{\partial V}{\partial q_k} + \frac{\partial D}{\partial \dot{q}_k} = Q_k \qquad \text{for} \qquad k = 1,\ldots,n, \tag{3}$$

where:

$\mathbf{q} = \begin{bmatrix} q_1 & \ldots & q_k & \ldots & q_n \end{bmatrix}^T$ – generalised coordinates,

$\dot{\mathbf{q}} = \begin{bmatrix} \dot{q}_1 & \ldots & \dot{q}_k & \ldots & \dot{q}_n \end{bmatrix}^T$ – generalised velocity,

E – kinetic energy,

V – potential energy,

D – energy dissipation function,

Q_k – non-potential generalised force corresponding to generalised coordinate q_k.

The vector $\mathbf{q}$ of generalised coordinates can be written as:

$$\mathbf{q} = \left[\ \mathbf{q}^{(D)^T} \quad \varphi \quad \psi \quad \mathbf{q}^{(J)^T} \quad \alpha \quad \mathbf{q}^{(L)^T}\ \right]^T, \tag{4}$$

where:

$\mathbf{q}^{(D)} = \left[\ x_{org}^{(D)} \quad y_{org}^{(D)} \quad z_{org}^{(D)} \quad \varphi^{(D)} \quad \theta^{(D)} \quad \psi^{(D)}\ \right]^T$ – vector of the generalised coordinates of the base (deck),

$\mathbf{q}^{(J)} = \left[\ q_1^{(J)} \quad \ldots \quad q_k^{(J)} \quad \ldots \quad q_m^{(J)}\ \right]^T$ – vector of the generalised coordinates of the jib,

$\mathbf{q}^{(L)} = \left[\ x_L \quad y_L \quad z_L\ \right]^T$ – vector of the generalised coordinates of the load,

φ – rotation angle of the crane's pedestal (upper structure) – slewing angle,

ψ – inclination angle of the undeformed jib,

α – rotation angle of the hoisting winch's drum.

Relationships which determine individual terms of the Lagrange's equations are obtained in analogy to the case of a mobile crane considered in [10]. Of particular interest is modelling the jib with the modal method described there. To ensure that the crane's base moves according to the assumptions of Table 1, the following condition must hold:

$$q_i^{(D)} = \beta_i^{(D)}(t) \qquad \text{for} \qquad i = 1, \ldots, 6. \tag{5}$$

Forces and torques acting on the crane's base to make it move according to relationships (5) must be therefore introduced into the system. They are assumed to form the following vector:

$$\mathbf{R}^{(D)} = \left[F_x^{(D)}, F_y^{(D)}, F_z^{(D)}, M_x^{(D)}, M_y^{(D)}, M_z^{(D)}\right]^T. \tag{6}$$

Forces and torques of force are depicted in Fig. 3.

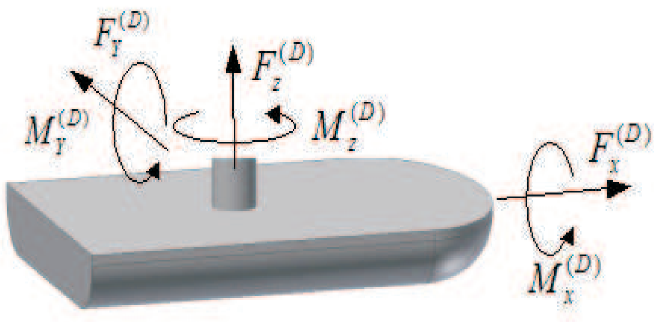

Fig. 3. External forces and their moments acting on the crane's base

The model of an offshore crane is ultimately described by the following equations of motion:

$$\mathbf{A}\ddot{\mathbf{q}} - \mathbf{D}\mathbf{R}^{(D)} = \mathbf{F}, \tag{7}$$

where:

$\mathbf{A}$ – mass matrix,

$\mathbf{R}^{(D)}$ – vector defined by Eq. (6),

$\mathbf{D} = \begin{bmatrix} \mathbf{I}_{6\times 6} \\ \mathbf{0}_{\overline{m}\times 6} \end{bmatrix}$,

$\bar{m} = 14 + m - 6$,

$\mathbf{I}_{6\times 6}$ – identity matrix of dimension 6×6,

$\mathbf{F}$ – vector on the right side of the equation of motion including among others the terms of Lagrange's equation related to velocities and generalised coordinates as well as non-potential forces not accounted for in vector $\mathbf{R}^{(D)}$, which must be supplemented by constraint equations:

$$\mathbf{q}^{(D)} = \boldsymbol{\beta}^{(D)}. \tag{8}$$

3. Stabilization of the load position

As has been mentioned in section 1, in this paper we consider the heavy motion as the most critical one. Thus, stabilization of the load position is reduced in these cases to compensation vertical oscillation of the load. A method of stabilization of the load position, based on calculation of drive functions in quasi-static conditions, was proposed in [1]. In order to ensure sufficient numerical effectiveness, the task has been solved by the simplified model. Before the desired drive functions can be determined, it is necessary to set the initial position of the mass m_L, i.e. the initial values of the coordinates x_L^0, y_L^0, z_L^0 for certain α^0, F_X, F_Y, F_Z. Next, points of time must be defined:

$$t_i = t_0 + i \cdot \Delta t \qquad \text{i} = 1, ..., \text{p}, \tag{9}$$

where

$\Delta t = \dfrac{t_k - t_0}{p}$,

t_0, t_k – start and end time of load stabilization, respectively,

p – number of intervals into which the time interval $\langle t_0, t_k \rangle$ has been divided.

After neglecting inertial terms in the dynamic equations and assuming that at points of time $t = t_1, ..., t_p$ the following conditions should be fulfilled:

$$z_L|_{t=t_i} = z_L^0 \qquad \text{i} = 1, ..., \text{p}, \tag{10}$$

equilibrium equations for $t = t_i$ can be written:

$$f\left(\alpha^{(i)}\right) = z_G - z_L^0 + \frac{L_{0L}}{S}(F_Z - m_L \cdot g) = 0 \qquad \text{i} = 1, ..., \text{p}, \tag{11}$$

while $z_G = z_G(t_i)$ depends on base motion of the crane's base.

The force S should be calculated according to the formula given in Table 1 for the simplified model.

Equations (11) form a system of p non-linear algebraic equations with p unknowns $\alpha^{(i)}$. These equations have been solved by applying Newton's iteration method, taking the starting point for subsequent iterations to be:

$$\alpha^{(i),0} = \alpha^{(i-1)} \tag{12}$$

The continuous functions $\alpha = \alpha(t)$ have been obtained by connecting points $\alpha^{(i)}$ with splines.

4. The results of numerical simulations

As it has been described above, the courses of hoisting winch angle are calculated using the simplified model and omitting dynamic phenomena. However, these courses compensating sea waving can be successfully applied in dynamic simulations using both models. The further calculations are carried out using the basic model of the crane.

Numerical calculations have been performed for an offshore crane, whose main parameters are: length of the jib 15 m, angle of jib inclination 50°, mass of load 2500 kg, and nominal (assumed) z coordinate of load – 200 m (200 m under sea level). The vertical motion of the base has been described by a harmonic function in the form:

$$z_{org}^{(D)} = 3\sin(0,897t + 1,57) \tag{13}$$

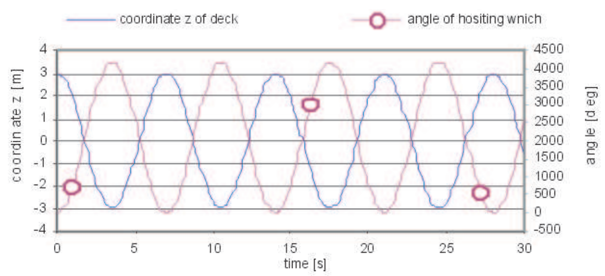

Fig. 4. Time courses of z coordinate of the deck and angle of hoisting winch

The time course of z coordinate of the deck and the course of drive function of the hoisting winch are shown at Fig. 4. One can see that both courses have the same frequency. In the graphs from Fig. 5, time courses of z coordinate of load and forces in the rope are presented. The courses denoted as "without stabilization" concern the cases when the crane has not realized any working motions. The courses denoted as "with stabilization" concern the cases when the drive of hoisting drum has been active.

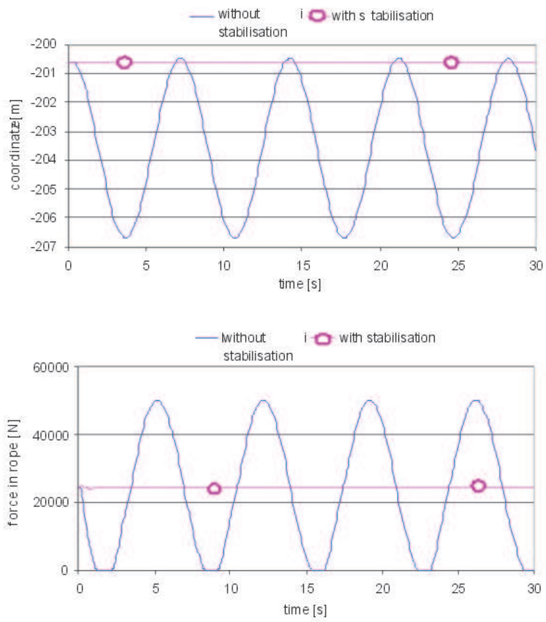

Fig. 5. Time courses of z coordinate of load and forces in the rope

The application of the hoisting winch drive function determined omitting dynamic phenomena is allowed in these special conditions when the mass of the load is big and frequencies are relatively low. This procedure permits to radically reduce time of calculations.

5. Conclusions

These examples of results of numerical calculations show that, under specific conditions, the taut-slack phenomenon can occur in the hoisting rope of an offshore crane. Periodical complete slacking of the rope is the essence of this phenomenon. The taut-slack phenomenon is caused by the significant waving of the sea and significant forces of resistance of the water acting on the load. These forces are correlated with the shape of the load. The method of stabilization of load position presented here eliminates this adverse effect. In future research, the authors intend to analyze the influence of additional forces (e.g. current forces) acting on the load on the taut-slack phenomenon.

Another interesting problem is the occurrence of the taut-slack phenomenon in ropes connecting ships (offshore cranes) with submarine vehicles.

It is important to note that the taut-slack phenomenon can significantly inhibit launching or lifting loads with large cross-section surface from the water. This problem is important especially in the case of submarine vehicles. Producers of launching cranes (jib and A-frame cranes) equip them with special devices which prevent slacking of the rope. This will also be a concern of ours in the future.

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Wpływ stabilizacji położenia ładunku żurawia typu offshore na zjawisko odciążania i dociążania liny nośnej

S t r e s z c z e n i e

Specyfika pracy żurawi typu offshore, a więc żurawi posadowionych na jednostkach pływających, nie polega tylko i wyłącznie na znacznych ruchach unoszenia ich bazy spowodowanych falowaniem morza. Innym niekorzystnym zjawiskiem jest odciążanie i dociążanie liny nośnej. W literaturze anglojęzycznej zjawisko to nazywane jest "taut-slack phenomenon". W pewnych warunkach może dojść do chwilowego, całkowitego odciążenia liny. Natomiast po niedługim czasie wartość siły w linie przewyższa zwykle znacznie wartość obciążenia nominalnego, wynikającego z ciężaru ładunku. Cykliczne powtarzanie się takiej sytuacji jest oczywiście bardzo niekorzystne dla całej konstrukcji nośnej żurawia i jego układów napędowych. W niniejszym artykule przedstawiono modele matematyczne żurawia typu offshore pozwalające na analizę zjawiska taut-slack. Zaprezentowano wyniki symulacji numerycznych wskazujące, że zastosowanie zaproponowanej we wcześniejszych pracach autorów stabilizacji położenia ładunku eliminuje ten problem.