**Andrzej URBAŚ*, Stanisław WOJCIECH**
University of Bielsko-Biała
Willowa 2, 43-309 Bielsko-Biała, Poland
**Corresponding author*. E-mails: aurbas@ath.bielsko.pl

# DYNAMIC ANALYSIS OF THE GANTRY CRANE USED FOR TRANSPORTING BOP

**Summary**. In the paper the dynamic analysis of a gantry crane used for transporting of BOP (BlowOut Preventer) is presented. The crane is placed on a drilling platform. Sea waves cause motion of the platform and the load. Description of such systems can be used in the design process of control systems which allows us to compensate waving. Homogenous transformations and joint coordinates are used to describe behavior of the system. Equations of motion are derived using the Lagrange equations of the second order. In the paper the results of numerical calculations are presented as well.

# ANALIZA DYNAMICZNA SUWNICY BRAMOWEJ PRZEZNACZONEJ DO TRANSPORTU BOP

**Streszczenie.** W pracy przedstawiono model dynamiczny suwnicy bramowej przeznaczonej do transportu ładunku zwanego BOP (BlowOut Preventer). Suwnice tego typu montowane są na statkach lub platformach wiertniczych. Falowanie morza powoduje niepożądane ruchy ładunku, wpływające na pracę suwnicy. Podjęto zatem próbę opisu dynamiki układu, która umożliwiła zbadanie wpływu falowania na zachowanie układu. Do opisu zastosowano formalizm transformacji jednorodnych i współrzędne złączowe. Równania ruchu układu wyprowadzono stosując równania Lagrange'a II rodzaju. Przedstawiono wyniki symulacji numerycznych.

## 1. INTRODUCTION

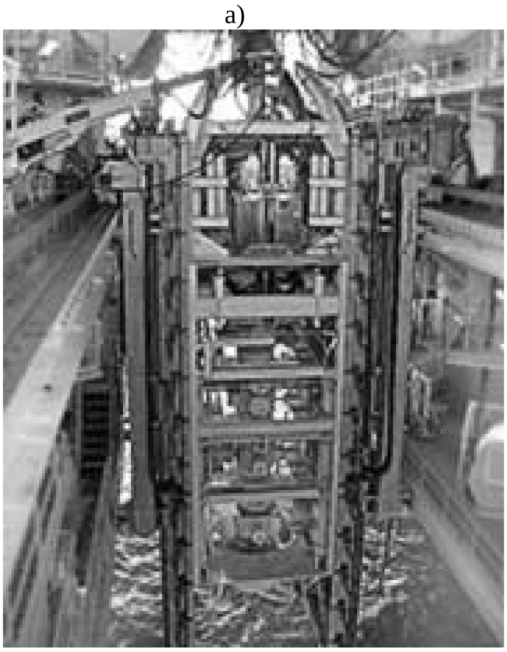
Dynamic analysis of offshore systems mounted on platforms or vessels is especially difficult because it is necessary to consider phenomena connected with sea waves [1,3,5]. The waving causes additional motion of the base (platform) and thus there are impulse forces in the system which should be taken into account in the design process not only in stress analysis but also in design of drive and control systems.

In the paper homogenous transformations and joint coordinates are used to describe dynamic analysis of the gantry crane [2,6]. Equations of motion are formulated using the Lagrange equations of the second order.

## 2. MATHEMATICAL MODEL OF THE SYSTEM

The gantry crane (Fig. 1) is considered as a system with 12DoF (Fig. 2). The frame $\{F\}$ is treated as a rigid body (6DoF) connected with platform $\{D\}$ by spring-damping elements (SDE). The load is also treated as a rigid body with 6DoF with respect to platform $\{D\}$. The load is connected with the frame by means of two flexible ropes and additionally its motion in $\hat{\mathbf{X}}^{(F)}$ and $\hat{\mathbf{Y}}^{(F)}$ directions is limited by guides.

a)



b)

Fig. 1. a) The gantry crane used for transporting BOP – PROTEA Gdańsk-Olesno, b) The BOP (Blowout Preventer) system

Rys. 1. a) Suwnica bramowa do transportu BOP – PROTEA Gdańsk-Olesno, b) Zestaw zaworów BOP

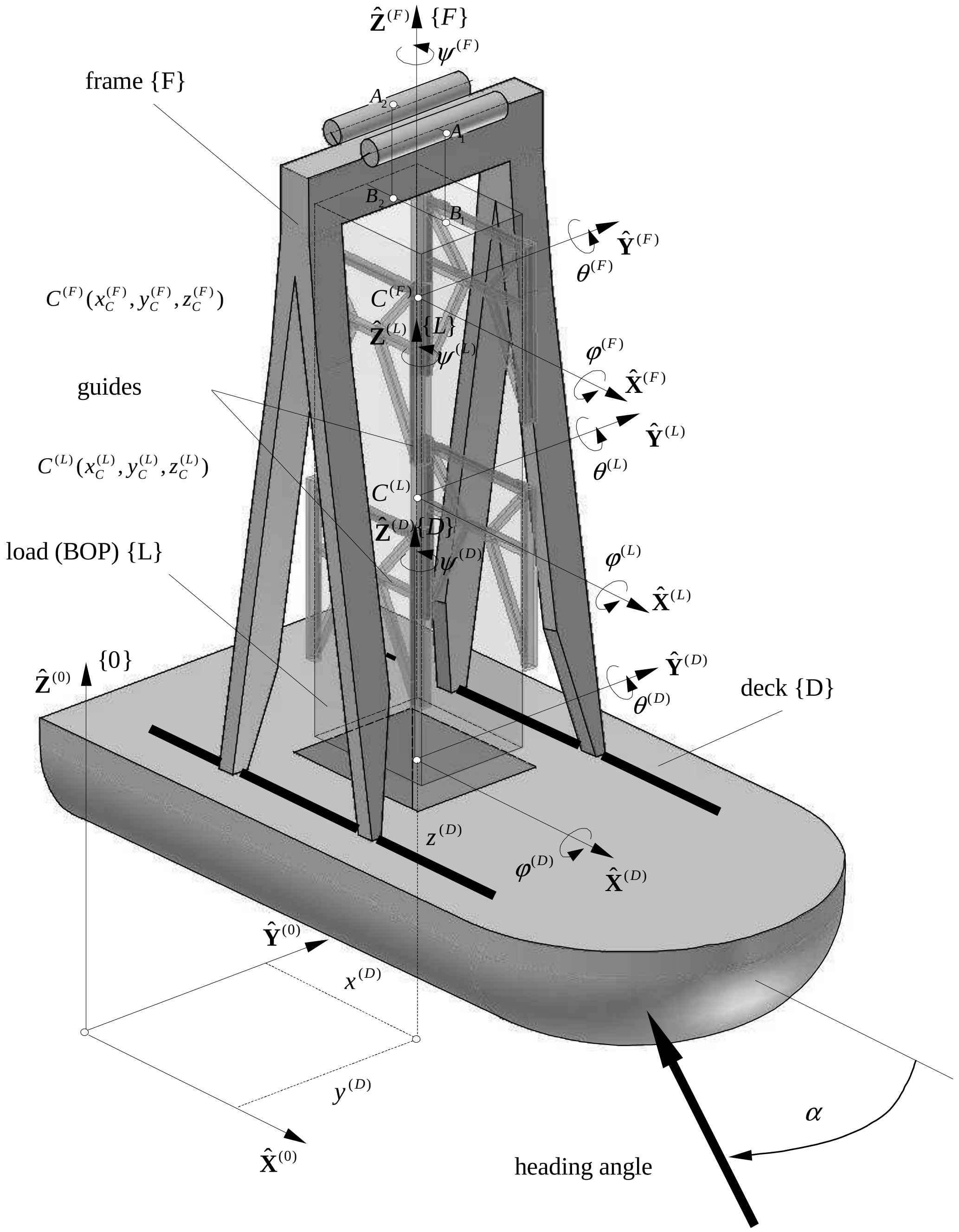



Fig. 2. Model of the gantry crane
Rys. 2. Model suwnicy bramowej

It is assumed that the motion of the platform is known. Consequently the position of coordinate system $\{D\}$ with respect to global coordinate systems $\{0\}$ is known.

The position of coordinate system $\{D\}$ with respect to $\{0\}$ is defined by:

$$\left.\begin{aligned} x^{(D)} &= x^{(D)}(t) \\ y^{(D)} &= y^{(D)}(t) \\ z^{(D)} &= z^{(D)}(t \end{aligned}\right\} \text{ the coordinates of the origin of coordinate system } \{D\} \text{ in } \{0\}, \qquad (1.1)$$

$$\left.\begin{aligned}\psi^{(D)} &= \psi^{(D)}(t)\\ \theta^{(D)} &= \theta^{(D)}(t)\\ \varphi^{(D)} &= \varphi^{(D)}(t\end{aligned}\right\}\text{ the Euler angles [2] which describe any possible orientation of }\{D\}\text{ in }\{0\}. \quad (1.2)$$

The motion of frame $\{F\}$ and load $\{L\}$ with respect to platform $\{D\}$ are described by independent parameters which are components of the vectors:

$$\mathbf{q}^{(F)} = \begin{bmatrix} x^{(F)} & y^{(F)} & z^{(F)} & \psi^{(F)} & \theta^{(F)} & \varphi^{(F)} \end{bmatrix}^T, \tag{2.1}$$

$$\mathbf{q}^{(L)} = \begin{bmatrix} x^{(L)} & y^{(L)} & z^{(L)} & \psi^{(L)} & \theta^{(L)} & \varphi^{(L)} \end{bmatrix}^T. \tag{2.2}$$

It is assumed that frame $\{F\}$ is connected with the platform by means of spring-damping elements with large values of stiffness and damping coefficients. Because the motion of load $\{L\}$ is limited by means of guides, angles $\varphi^{(F)}, \theta^{(F)}, \psi^{(F)}, \varphi^{(L)}, \theta^{(L)}, \psi^{(L)}$ are small and the transformation matrices from coordinate systems $\{F\}$ and $\{L\}$ to $\{D\}$ can be presented in the linearised form [5,1]:

$$\tilde{\mathbf{B}}^{(F)} = \begin{bmatrix} 1 & -\psi^{(F)} & \theta^{(F)} & x^{(F)} \\ \psi^{(F)} & 1 & -\varphi^{(F)} & y^{(F)} \\ -\theta^{(F)} & \varphi^{(F)} & 1 & z^{(F)} \\ 0 & 0 & 0 & 1 \end{bmatrix}, \tag{3.1}$$

$$\tilde{\mathbf{B}}^{(L)} = \begin{bmatrix} 1 & -\psi^{(L)} & \theta^{(L)} & x^{(L)} \\ \psi^{(L)} & 1 & -\varphi^{(L)} & y^{(L)} \\ -\theta^{(L)} & \varphi^{(L)} & 1 & z^{(L)} \\ 0 & 0 & 0 & 1 \end{bmatrix}. \tag{3.2}$$

The transformation matrices from local coordinate systems $\{F\}$ and $\{L\}$ to global coordinate system $\{0\}$ can be writen in the form:

$$\mathbf{B}^{(F)} = \mathbf{A}^{(D)} \tilde{\mathbf{B}}^{(F)}(\mathbf{q}^{(F)}), \tag{4.1}$$

$$\mathbf{B}^{(L)} = \mathbf{A}^{(D)} \tilde{\mathbf{B}}^{(L)}(\mathbf{q}^{(L)}), \tag{4.2}$$

where

$$\mathbf{A}^{(D)}(t) = \begin{bmatrix} c\psi^{(D)} c\theta^{(D)} & c\psi^{(D)}\, s\theta^{(D)}\, s\varphi^{(D)} - s\psi^{(D)}\, c\varphi^{(D)} & c\psi^{(D)}\, s\theta^{(D)}\, c\varphi^{(D)} + s\psi^{(D)}\, s\varphi^{(D)} & x^{(D)} \\ s\psi^{(D)}\, c\theta^{(D)} & s\psi^{(D)}\, s\theta^{(D)}\, s\varphi^{(D)} + c\psi^{(D)}\, c\varphi^{(D)} & s\psi^{(D)}\, s\theta^{(D)}\, c\varphi^{(D)} - c\psi^{(D)}\, s\varphi^{(D)} & y^{(D)} \\ -s\theta^{(D)} & c\theta^{(D)}\, s\varphi^{(D)} & c\theta^{(D)}\, c\varphi^{(D)} & z^{(D)} \\ 0 & 0 & 0 & 1 \end{bmatrix}.$$

The equations of motion are derived using the Lagrange equations of the second order:

$$\frac{d}{dt}\frac{\partial E}{\partial \dot{q}_k} - \frac{\partial E}{\partial q_k} + \frac{\partial V}{\partial q_k} + \frac{\partial D}{\partial \dot{q}_k} = Q_k, \qquad k = 1, \ldots, 12, \tag{5}$$

where: $E$ – is the kinetic energy of the system, $V$ - is the potential energy of gravity forces, $D$ - is the dissipation of energy of the system, $Q_k$ - are non-potential generalised forces, $q_k$, $\dot{q}_k$ - are generalised coordinates and velocities, respectively – the componets of the vectors (2).

The kinetic energy of the system, the potential energy of gravity forces and its derivatives can be calculated as it has been shown in [5].

Additionally we have to take into account:

- the energy of spring deformation of the ropes,
- the energy of spring deformation of the guides,
- reaction forces of rigid or flexible supports.

Fig. 3 presents connection of the load and ropes. The load is hoisted by means of two barrel systems. In mathematical model of the gantry crane, the hoist system is reduced to two flexible ropes.

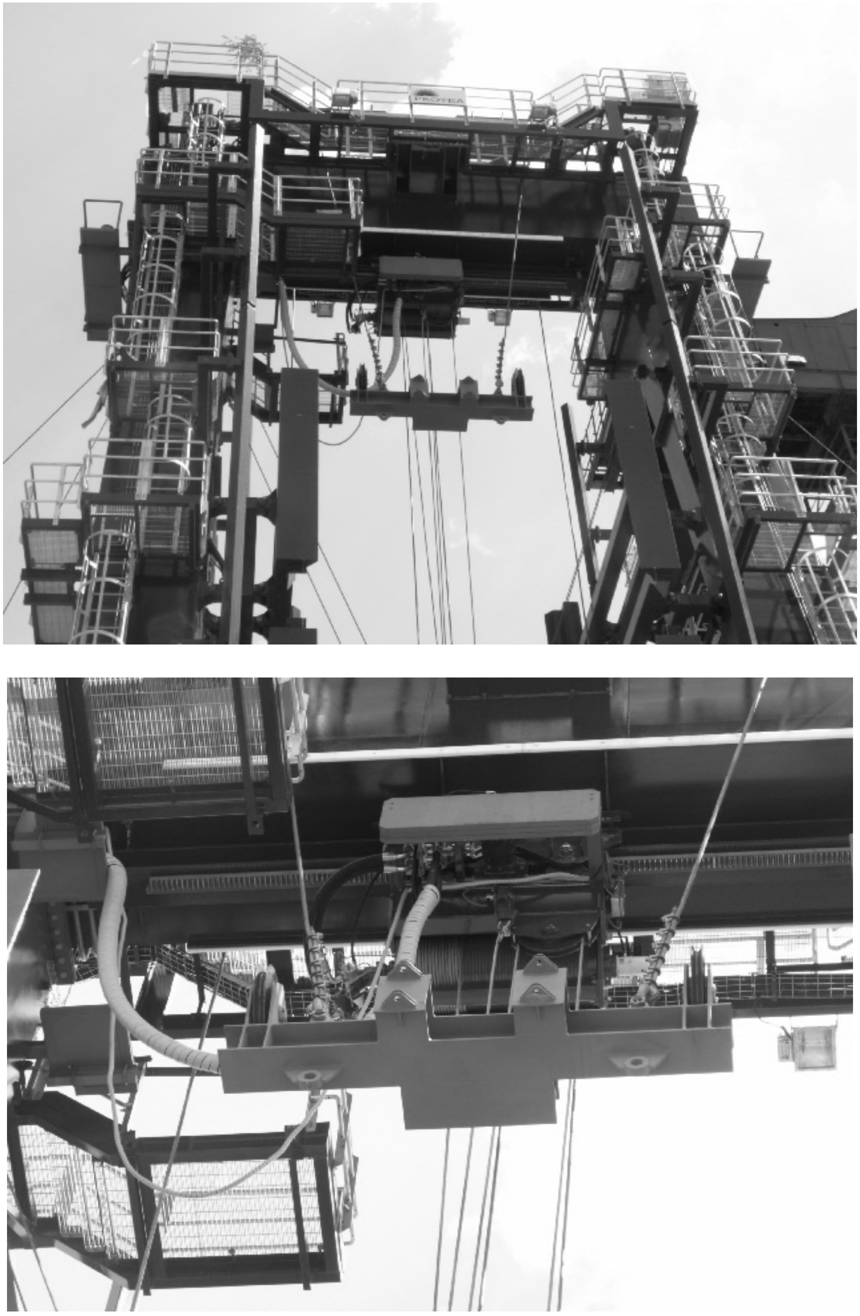


Fig. 3. Hoist system
Rys. 3. Układ zawieszenia ładunku

The motion of the load is limited by means of four guides which positions are set up by hydraulic cylinders (Fig. 4).

Fig. 4. Guide system
Rys. 4. Układ prowadnic

It is assumed that the load (rectangular prism) can be in contact with the guides only along lines $A^{(k)}$, $B^{(k)}$, where $k = 1, 2, 3, 4$ (Fig. 5a).

The guides are modelled as spring-damping elements with backlash $(\text{sde}\,E^{(k,p)})$, which limits the motion of the load in $\hat{\mathbf{X}}^{(D)}$ and $\hat{\mathbf{Y}}^{(D)}$ directions (Fig. 5b).

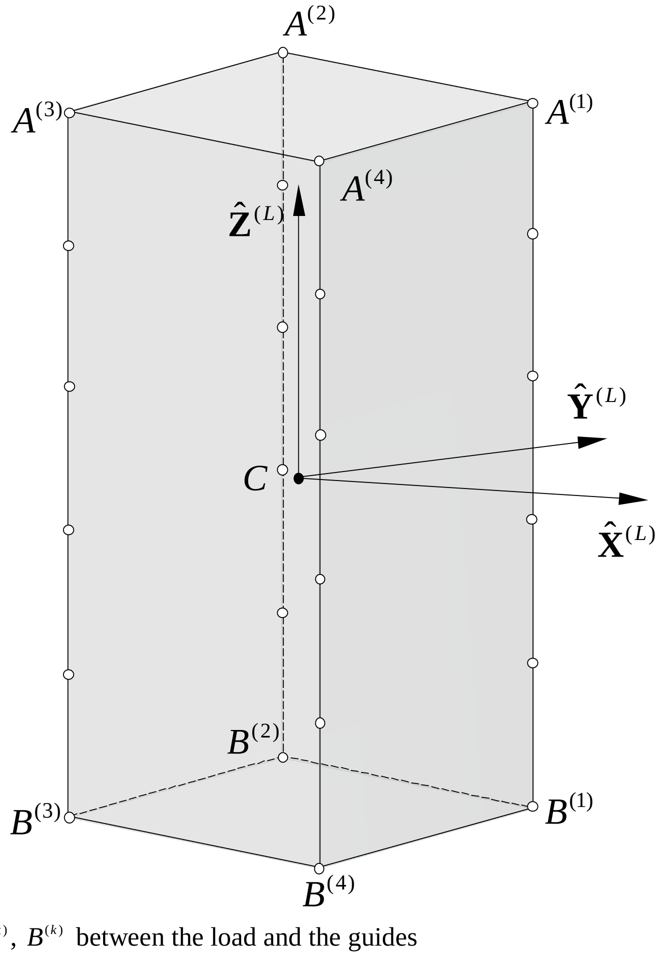
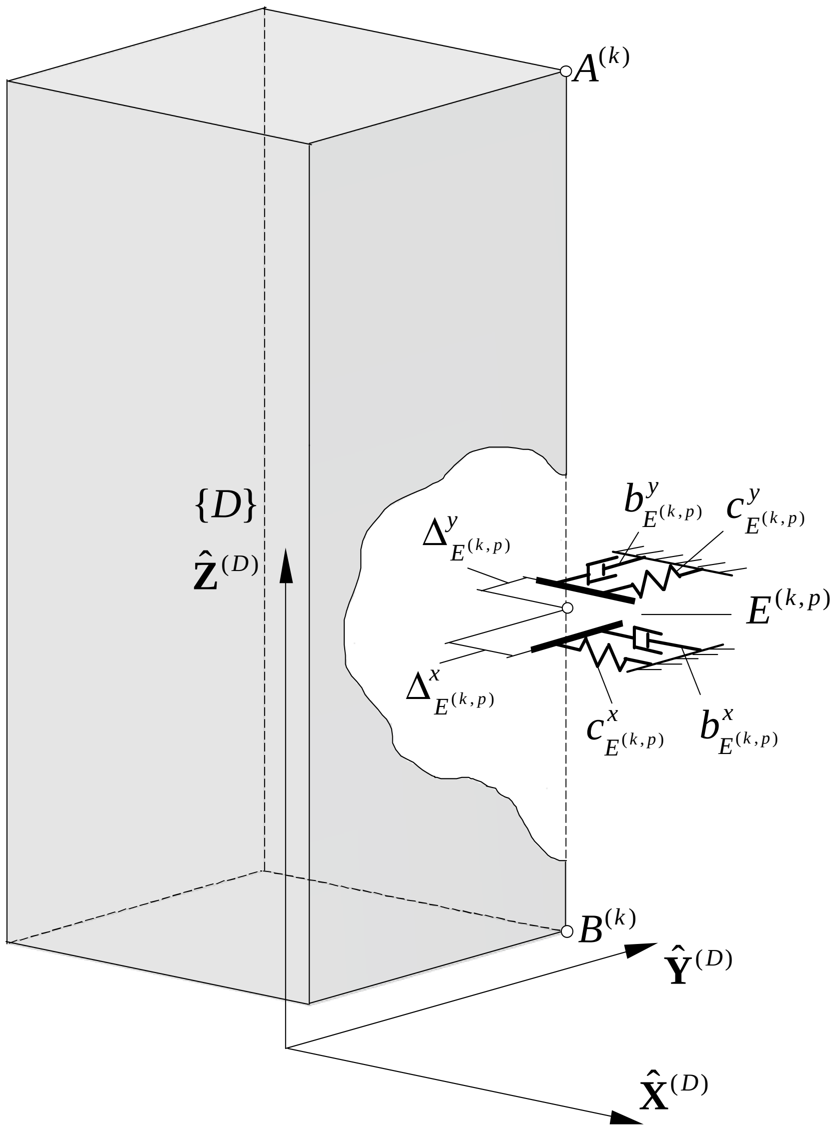



Fig. 5a. Contact lines $A^{(k)}$, $B^{(k)}$ between the load and the guides
Rys. 5a. Linie styku $A^{(k)}$, $B^{(k)}$ ładunku z prowadnicami




b)

Fig. 5b. Spring-damping element with backlash
Rys. 5b. Element sprężysto-tłumiący z luzem

Fig. 6 presents the way of connecting the gantry to the deck. The system is supported on the rigid rails. The motion in $\hat{\mathbf{X}}^{(D)}$ direction is realized by system of 33 rollers (15 in contact). The motion in $\hat{\mathbf{Y}}^{(D)}$ direction is limited by system of four pressure rollers on each leg.

Fig. 6. Connecting the gantry crane to the deck
Rys. 6. Posadowienie suwnicy BOP

The gantry crane is also protected by antylift system which limits the motion of the system in vertical direction ( $\hat{\mathbf{Z}}^{(D)}$ axis) (Fig. 7).

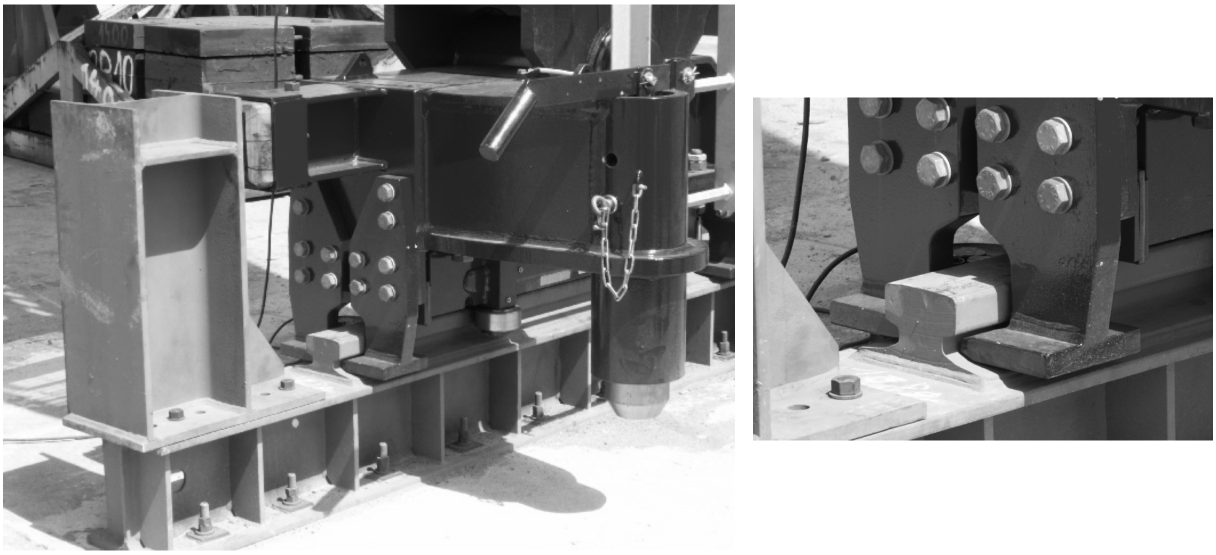


Fig. 7. Antylift system
Rys. 7. Układ zabezpieczający

Fig. 8 presents the model of flexible connection of the frame legs with the deck ( $P^{(k)}$ point).

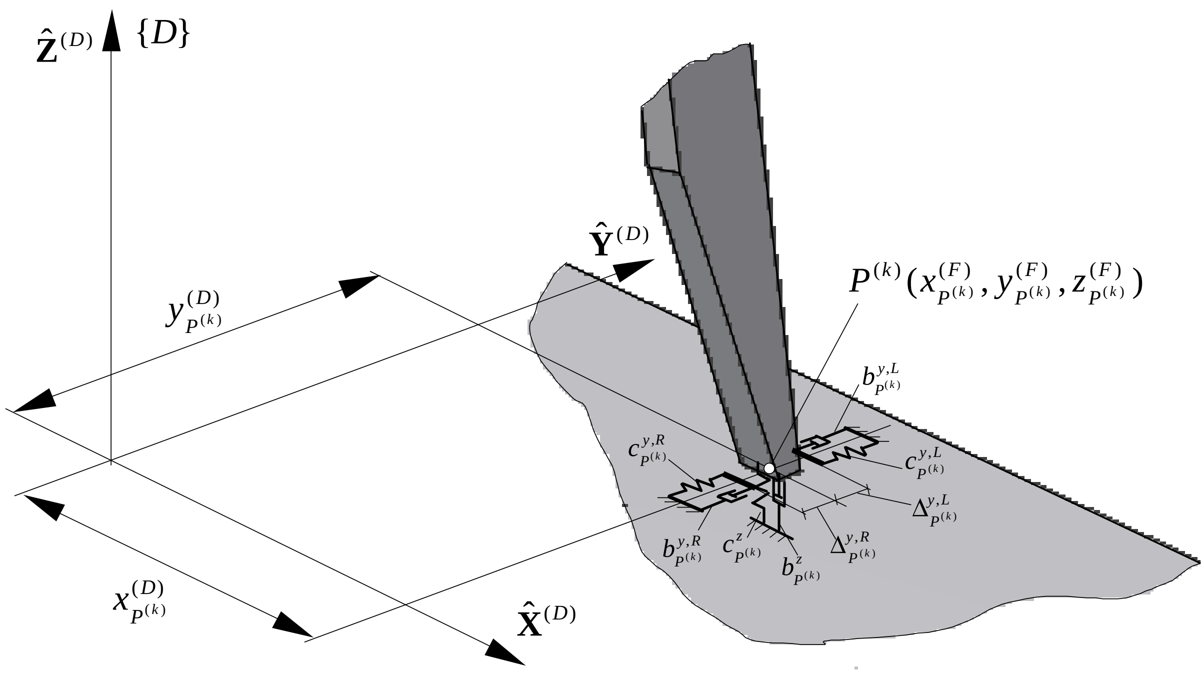



Fig. 8. Flexible connection of the frame to the platform
Rys. 8. Podatne mocowanie ramy do pokładu

The structure of the equations of motion is different and depends on selected drive functions of the motion of the frame in $\hat{\mathbf{X}}^{(D)}$ direction:

A. flexible drive system

$$\mathbf{A}\ddot{\mathbf{q}} = \mathbf{f}\ , \tag{6}$$

B. rigid drive system

$$\begin{aligned} \mathbf{A}\ddot{\mathbf{q}} - \mathbf{DF} &= \mathbf{f} \\ \mathbf{D}^T\ddot{\mathbf{q}} &= \ddot{\boldsymbol{\delta}} \end{aligned}\ , \tag{7}$$

where $\mathbf{q} = \begin{bmatrix} \mathbf{q}^{(F)} \\ \mathbf{q}^{(L)} \end{bmatrix}$,

$\mathbf{F} = \begin{bmatrix} F_x^{(1)} \\ F_x^{(4)} \end{bmatrix}$,

A is constant matrix,

$\mathbf{f} = \mathbf{f}(t, \mathbf{q}, \mathbf{q})$,

$\boldsymbol{\delta} = \begin{bmatrix} \delta_1(t) \\ \delta_2(t) \end{bmatrix}$ is the vector describing the displacements of $P^{(1)}$ and $P^{(4)}$ points in $\hat{\mathbf{X}}^{(D)}$ direction.

The model of dynamics of the gantry crane considered is described by 12 (eq. 6) or 14 (eq. 7) ordinary differential equations of the second order. In order to integrate the equations of motion the Runge-Kutta method with constant step-size is used.

Having assumed $\ddot{\mathbf{q}} \equiv 0;\ \dot{\mathbf{q}} \equiv 0$ in eq.6 and eq. 7 the vector **q**, reaction forces in ropes and in SDEs can be calculated. In this case the system is described by the set of nonlinear equations:

$$\mathbf{f}(t, \mathbf{q}) = 0\,, \tag{8}$$

The dependence **f** versus *t* is caused by expressions connected with the base motion of the deck.

Equations (8) are solved using the Newton iteration method.

## 3. NUMERICAL CALCULATIONS

For the mathematical model of the gantry crane the computer program has been developed. The following input data are taken from the technical documentation [4].

– the mass of the frame and the load: $m^{(F)} = 73\ 955$ kg , $m^{(L)} = 550\ 000$ kg ,

– dimensions of the load $4.8 \times 5.5 \times 20.3$ m .

In table 1 conditions of work for BOP are presented [4]. It is assumed that the motion of the deck is described by heave, pitch and roll motions which are written as follows: $x^{(D)} = y^{(D)} = \psi^{(D)} = 0$, $z^{(D)} = z_0^{(D)} + a_3 \sin(2\pi t / T)$, $\theta^{(D)} = a_5 \sin(2\pi t / T)$, $\varphi^{(D)} = a_6 \sin(2\pi t / T)$ where $z_0^{(D)} = 36\,\mathrm{m}$, $T = 10$ s, $a_3, a_5, a_6$ are amplitudes depending on heading and weather conditions.

Table 1

Conditions of work for BOP

| HEADING $\alpha$ | OPERATIONAL | | | SURVIVAL | | |
|---|---|---|---|---|---|---|
| | heave $z^{(D)}$ $a_3$ | pitch $\theta^{(D)}$ $a_5$ | roll $\varphi^{(D)}$ $a_6$ | heave $z^{(D)}$ $a_3$ | pitch $\theta^{(D)}$ $a_5$ | roll $\varphi^{(D)}$ $a_6$ |
| 0° | 0.1343 | 0.0023 | 0 | 0.4458 | 0.0061 | 0 |
| 45° | 0.1115 | 0.0008 | 0.0023 | 0.3521 | 0.0023 | 0.0077 |
| 90° | 0.1140 | 0 | 0.0045 | 0.3724 | 0 | 0.0138 |

Fig. 9a presents the comparison of *z* component of forces in $P^{(1)}$ point and Fig. 9b presents the elongation of rope $\Delta l_{A_1B_1}^{(1)}$. In both cases it is asumed that only heave motion of the platform and the heading angle equal to $0^\circ$.

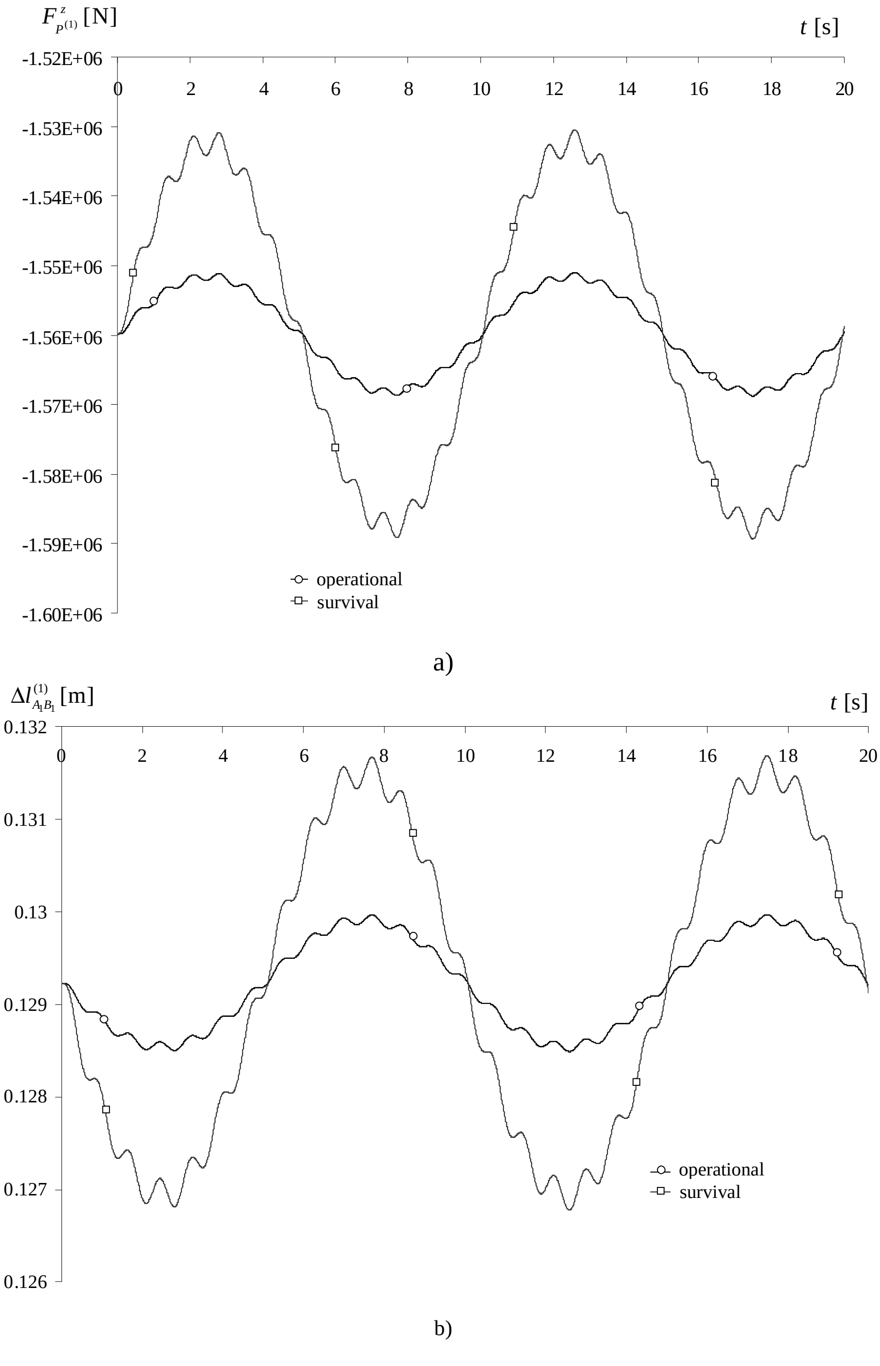



Fig. 9. The influence of working conditions on: a) forces in connection point $P^{(1)}$, b) the elongation of rope 1
Rys. 9. Wpływ warunków pracy na: a) wartość siły w podporze $P^{(1)}$, b) wydłużenie liny 1

Fig. 10 presents *z* component of forces at points $P^{(1)}$ and $P^{(4)}$. The results have been obtained for heading angle equal to $45^{\circ}$ in operational conditions of work.

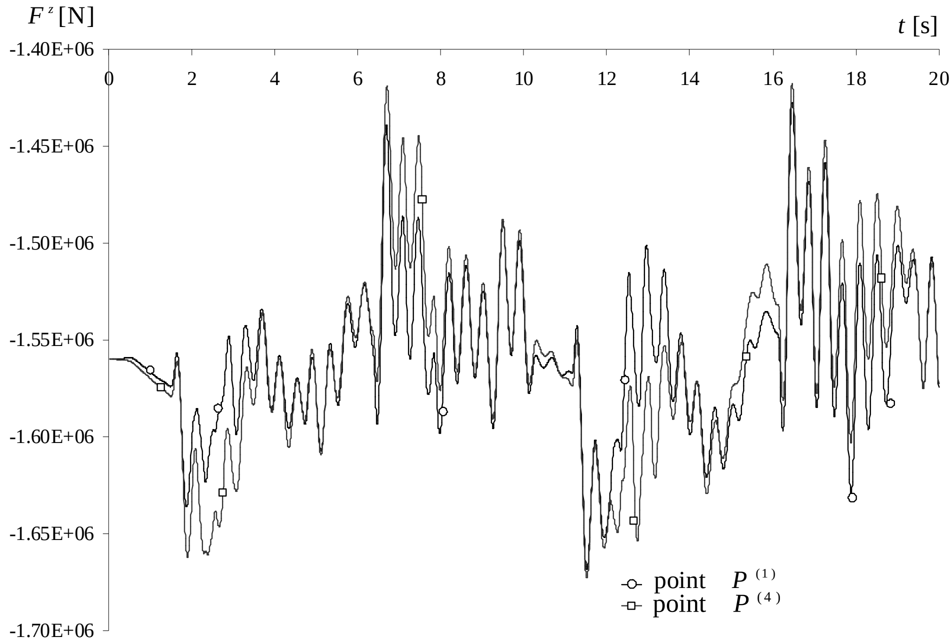



Fig. 10. Reaction forces in $P^{(1)}$ and $P^{(4)}$ points
Rys. 10. Siły reakcji w punktach $P^{(1)}$ i $P^{(4)}$

The influence of backlash $\Delta^x_{E^{(k,p)}}$ and $\Delta^y_{E^{(k,p)}}$ on *z* component of forces at point $P^{(1)}$ can be observed in Fig. 11. It is assumed that heading angle equals to $90^\circ$ in operational conditions of work and only pitch motion of the platform is taken into account.

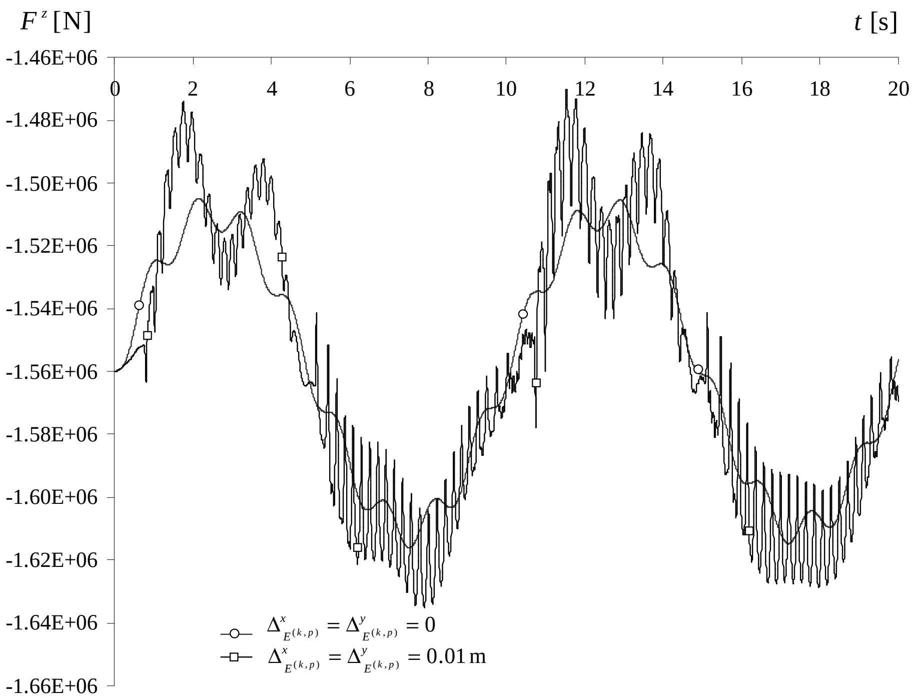



Fig. 11a. The influence of backlash in SDE on reaction force (component *z*) in point $P^{(1)}$
Rys. 11a. Wpływ luzu w EST na składową *z* siły reakcji w punkcie $P^{(1)}$

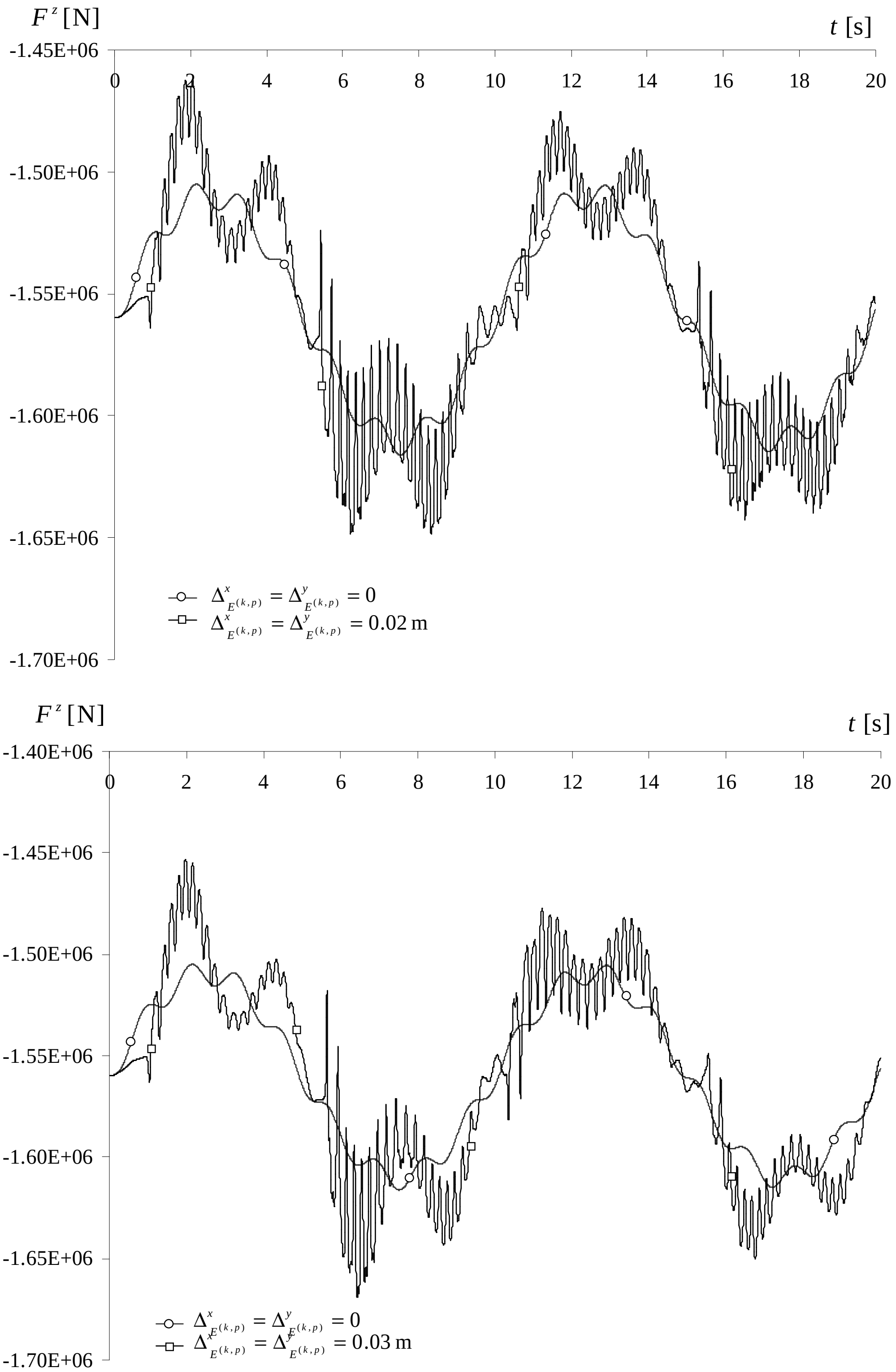
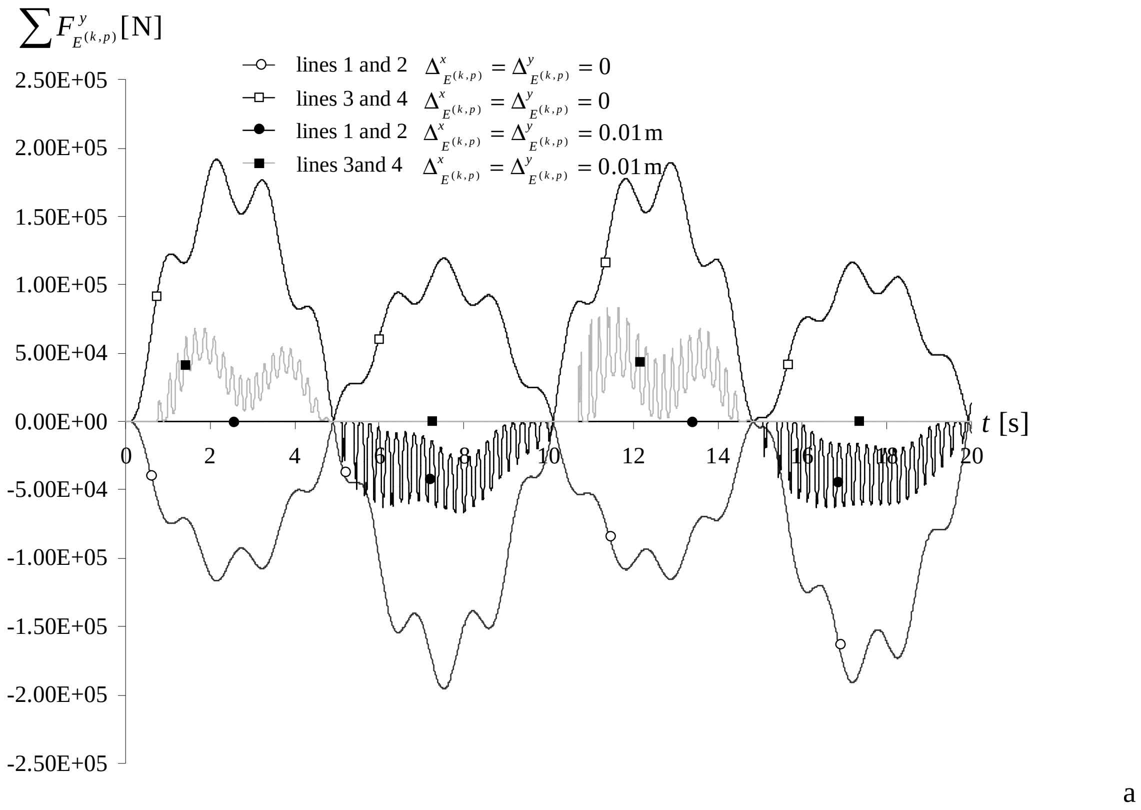



Fig. 11b,c. The influence of backlash in SDE on reaction force (component *z*) in point $P^{(1)}$

Rys. 11b,c. Wpływ luzu w EST na składową *z* siły reakcji w punkcie $P^{(1)}$

Fig. 12 presents the influence of backlash $\Delta^x_{E^{(k,p)}}$ and $\Delta^y_{E^{(k,p)}}$ on the sum of forces acting between the guides and the load. It is assumed that the heading angle equals to $90^\circ$ and pitch motion of the platform in operational conditions of work is taken into account.



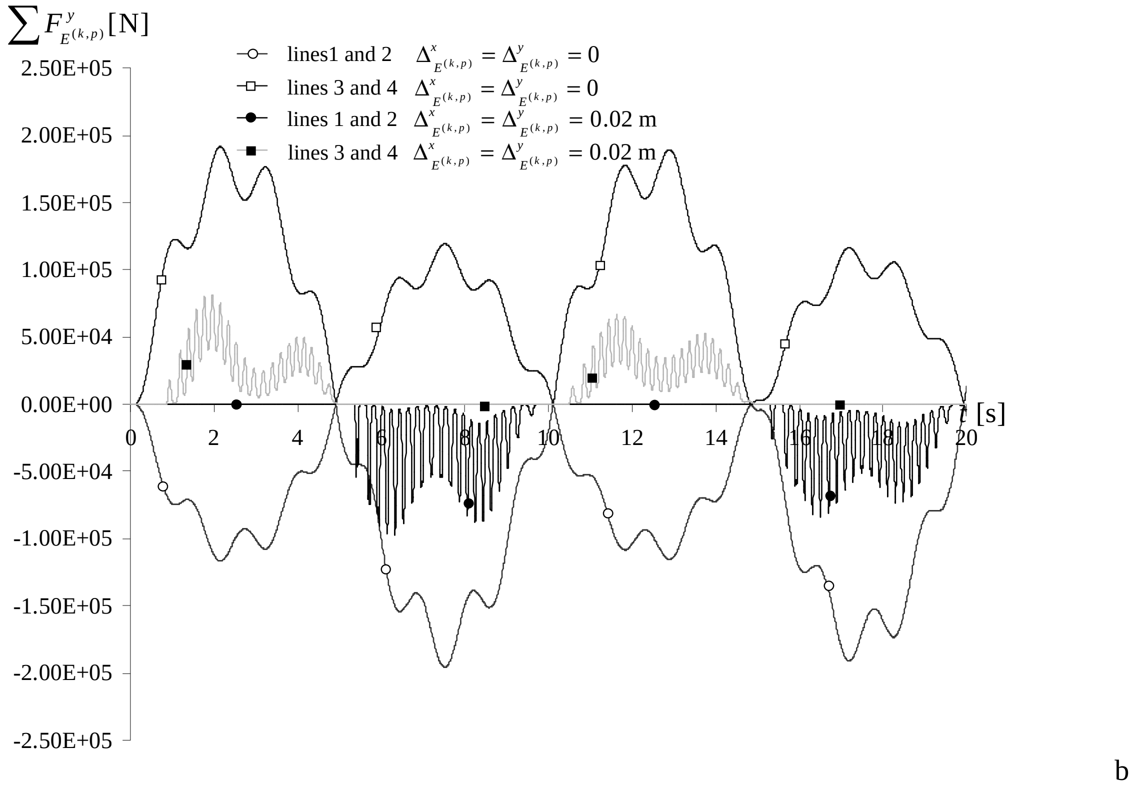




Fig. 12. The influence of backlash in SDE on the sum of forces $\sum F^{y}_{E^{(k,p)}}$

Rys. 12. Wpływ luzu w EST na sumaryczną wartość siły $\sum F^{y}_{E^{(k,p)}}$

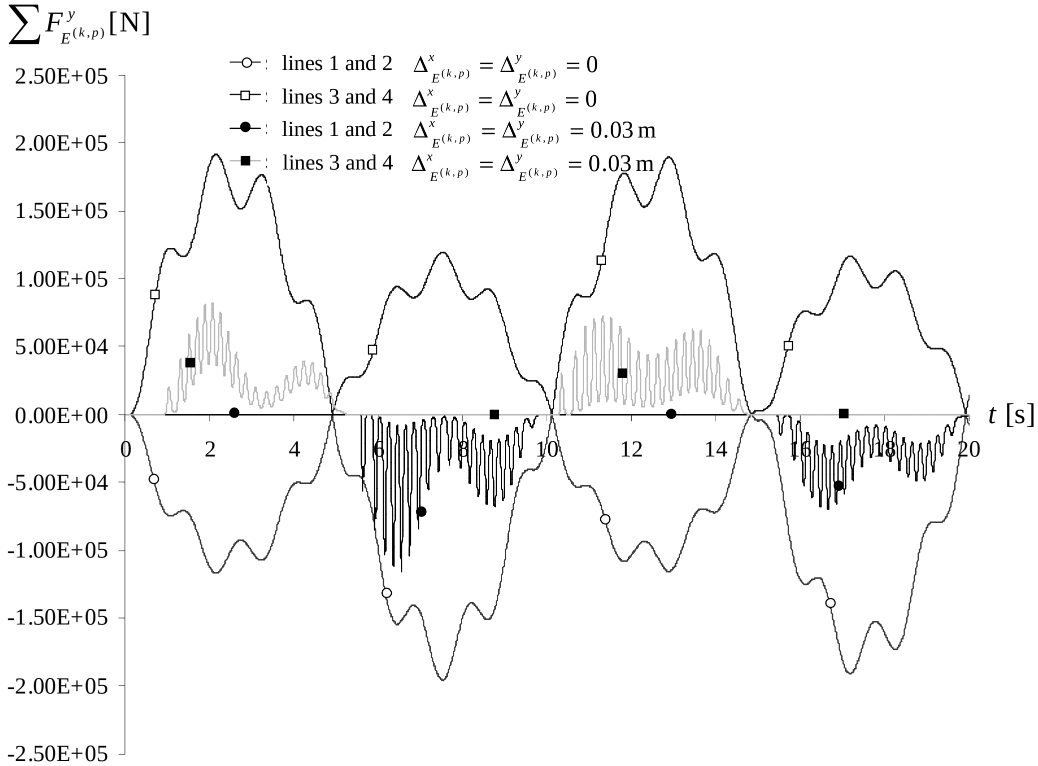



Fig. 12. The influence of backlash in SDE on the sum of forces $\sum F^{y}_{E(k,p)}$

Rys. 12. Wpływ luzu w EST na sumaryczną wartość siły $\sum F^{y}_{E(k,p)}$

## 4. CONCLUSIONS

Numerical simulations presented prove that the dynamic analysis of off-shore systems is very important in the design process. Impuls forces acting in the system should be included in stress analysis of the guides and also the frame. We are going to analyse drive systems in order to find out how to compensate the frame and the load motion caused by waving. The numerical calculation presented in Fig. 10 indicates that drive systems in certain conditions should work independently.

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