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(54) METHOD AND DEVICE TO CONTROL IN OPEN-LOOP THE SWAY OF PAYLOAD FOR SLEWING CRANES

(57) Method and systems for controlling the swing of a suspended load by a rotary crane moving along two axes, trolley and slewing, that are moving independently or together, and independently or together to the movement of the load along a third axis, the vertical (hoisting) axis.The device (10, 11, 12) calculates the speed profiles of slewing and trolley axes using as input variables the length of the cable in which is suspended the load, the distance between the axis of rotation and the trolley position, the set constant acceleration of slewing and trolley axes, the air resistance forces acting on the payload and on the hoisting system, the damping of the rotation movement due to the structure's elasticity of the hoisting system.

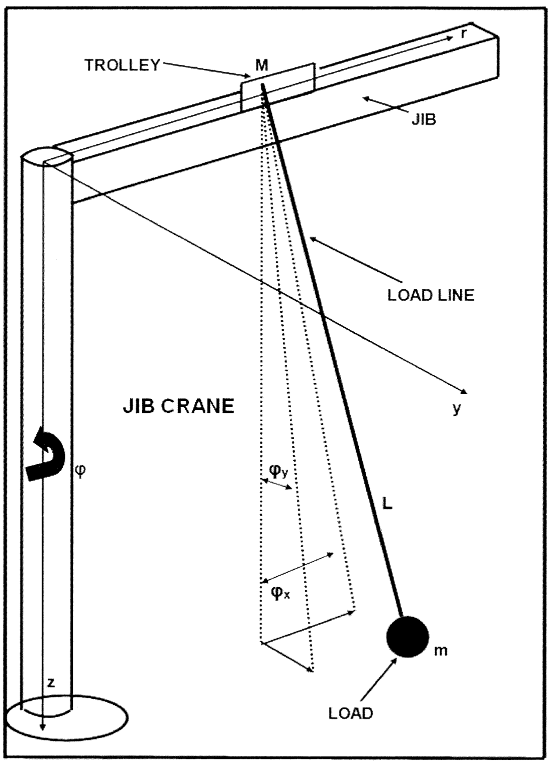

FIG. 1

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Description

1. BACKGROUND OF THE INVENTION.

[0001] In a general way, it is possible to divide cranes in two categories:

industrial cranes and building (rotary) cranes.

[0002] For the first category of cranes an anti-sway method is easier to develop as a consequence of the decoupling among the equations of the movement. In fact industrial cranes exhibit a one degree-of-freedom sway in the solution equations.

[0003] At the contrary, Slewing cranes, such as Tower cranes or self-erecting cranes, are subjected to the condition that the load-line attachment undergoes to rotation. In this case the equations of the movement are necessary coupled. Conventionally, point-to-point operations using slewing cranes are performed by operators with the goal to not excite the pendulum modes of their cable and payload. Typically these pendulum modes exhibit low frequencies.

[0004] The two fundamental motions of the payload in the horizontal plane, resulting from the operator commands produce a sway tangential to the arc of circle defined by the tip of the crane jib and a sway radial to the same rotation due to the centrifugal acceleration of the payload.

[0005] Currently, speaking about slewing cranes, the swing of the payload is not compensated in any automatic way. Only for pre-planned motions where the desired start and stop positions of the payload are well specified it is possible to define an automatic anti-sway method. But, that is not the typical situation of slewing crane for building sector. In this case, at today, only the high experience of the crane operator allows to bring the payload to a stop point without swing.

[0006] The study of slewing cranes involves the analysis of three movements: the slewing motion of the jib, the radial motion of the load suspension point (trolley) and the hoisting of the payload.

[0007] The movements of the crane cause sway of the load. There are extensive researches on anti-sway for industrial (overhead travelling) cranes. That is due, in this case, to the possibility to decouple the movements and, as a consequence of this, to use linear models to solve the motion equations. Instead of that, slewing crane movements are characterized by the appearance of Coriolis and Centrifugal forces. So the equations become non-linear and the application of linear control theory leads to several problems.

[0008] In this invention, in order to solve this not-linear problem, we make considerations concerning non-linear force interactions, using the Lagrange equations governing the physical system.

[0009] The fundamental movements of a Tower or jib (slewing) crane typically are:

1. A rotation (or slewing) of the jib. The jib is connected to the top of a vertical shaft. The jib has a suspension point for the payload that is suspended thorough cables. The jib performs a rotation or slewing movement about a vertical axis Z that corresponds with the vertical shift of the crane (Slewing movement).
2. A linear movement of the suspension point of the load along the jib. This second movement can be referred as a translation movement. We will refer, in the present invention, to cranes where the suspension point of the load is a trolley, the translation movement then being performed along the horizontal axis X of the jib (Trolley movement).
3. A device able to hoist the load. This device is defined typically using cables which pass through the crane. The length of the cables is variable in order to move the load on a vertical axis Z. This third possible movement along the axis Z is usually named as vertical movement (or Hoisting movement).

[0010] As a consequence of this industrial apparatus, typically, a building crane has three inputs as movement commands which command in an independent way the trolley movement, the slewing movement and the hoisting movement.

[0011] It is obviously desirable to reduce the sway of the payload in order to obtain two positive effects: increase the safety of the load transfer and reduce the operational time to the shorter possible time.

[0012] At difference of the sway generated from linear movements, the sway due to a rotation, as a consequence of the centrifugal force, remains present even if the tangential component of the acceleration during the rotation movement is zero.

[0013] Several solutions to the problem to reduce the swing of the suspended load were proposed in the past. But, almost all the solutions are related to sway generated with industrial cranes, where there is not the complication due to the slewing movement.

[0014] With reference to inventions relative to tower cranes, where is considered the slewing of the crane, and as a consequence the effect of the centrifugal force, only few works were obtained in the past.

[0015] We consider really only three works that in the past treated the situation of the slewing in tower cranes.

[0016] The first one is relative to the U.S. Patent 5,908,122 to Robinett et al., where the sway correction is obtained filtering crane input signals in order to dump the payload sway. The filtering method uses a model representing the dynamics of the rotary crane with a matrix of nonlinear equations of motion, and after linearizing the matrix with respect to the radial sway angle and to the tangential sway angle. In this way the filtered inputs are obtained in order to control the sway of the payload.

[0017] The most important limitations of that invention are two.

[0018] The first one is due to the fact that the equations of the movement are linearized. In this way, all the effects relative to the centrifugal force and to the Coriolis force are not considered, so arriving to a solution of the anti-sway model that is correct only in first approximation.

[0019] The second limitation is due to consider always in the final results of the above invention the jib angular acceleration significantly less than the static pendulum natural frequency $\omega = \sqrt{g/L}$, where g is gravity. (9.8 m/s^2) and L is the length of load-line. That approximation carries out as a consequence that the filters of the movement commands can be decoupled. But, often the realistic situation provides comparable values between the above quantities.

[0020] At the contrary, in the present invention these two effects are considered: both the centrifugal force that produces the non-linear terms in the movement equations than the possibility to have comparable values between the jib angular acceleration and the static pendulum natural frequency are considered. So the present invention is really able to consider all the non-linear effects of the crane movements.

[0021] The second work relative to tower (slewing) cranes is in the recent Patent in U.S. Patent Application Publication 2011/0218714 A1 to Stantchev Pentcho et al.

[0022] In this invention the calculation step uses a mathematical model of the pendulum with damping.

[0023] But the limit of this invention is in the fact that the correction coefficients (necessary to define the correct damping of the sway) are defined experimentally according to the different possible lengths of the suspension cables of the load. So, these coefficients have to be obtained only after experimentation, without way to have a complete theory of their computation.

[0024] At the contrary in the present invention the theory is developed without the need to obtain in a certain way the coefficients in the movement equations. So the work is directly applicable on any length of the cable relative to the payload.

[0025] Moreover, in the same invention of Stantchev Pentcho et al., as in the work of Robinett et al., are ignored the effect of all the term of second order (apart the centrifugal effect in the work of Stantchev Pentcho et al.) relative to the Coriolis force. That loads to a loose of the precision of the sway correction.

[0026] In the present invention the same effects are totally considered, developing a mathematical model containing all relevant effects and their coupling. That is just in order to obtain a precise value of the corrections to the sway.

[0027] The third work relative to tower (slewing) cranes is in the recent Paper "High Performances Tracking Control of Automated Slewing Cranes" in "Robotics and Automation in Construction" to Frank Palis et al.

[0028] Also in this work the developed model takes in account the nonlinear effects as Coriolis and Centrifugal forces. Due to the difference in dynamics the general motion of the crane was divided into slow and fast subsystems, which can be optimized independently. The slow system reflects load dynamics and it is used as basis in order to calculate the damping. Nevertheless the limit of this work lies in the fact that they didn't develop an analytic solution of the movement equations, but only they obtained the equation solutions using Automation theory and numerical methods. That loads to don't obtain an exact solution of the anti-sway problem.

[0029] At the contrary, in the present invention the solution of the movement equations is obtained in analytic way, obtaining the exact solution of the filters able to reduce the oscillatory motion of rotary crane payloads.

- As a consequence of the reasons explained above in comparison to the previous inventions, the present invention represents an effective and consistent improvement in the state of the art, relatively to the control of the sway of payloads for slewing cranes.

2. SUMMARY OF THE INVENTION.

[0030] The object of the present invention is to define a system and a method to control the sway of a rotating (building) crane using a computer-controlled system with open-loop control. Particularly, the present invention considers the full dynamic effect of the slewing, trolley and vertical (hoisting) movement, solving in an exact way the fundamental equations of the movement.

[0031] This invention defines a device to control the movement of a load suspended by cables from a suspension point of an hoisting system, the suspension point being able to perform both a rotation movement about a vertical rotation axis (Slewing axis) and a translation movement along a translation axis (Trolley axis).

[0032] The control process does not require to know variable parameters such as the measure of the angle of swing or a measure of the current flowing in the motor.

[0033] For this, the invention requires a device for adjusting the movement of the load suspended by cables to a crane which is movable along a first horizontal axis (Trolley movement). Said device includes means for determining the length of cables for the load suspension and his variation in the time, means for determining the air resistance forces acting on the payload and on the hoisting system and means for determining the damping of the rotation movement due to the structure's elasticity of the hoisting system.

[0034] According to one feature, a drive controller calculates the speed profile of this first movement relative to the trolley and it supplies the information of the speed profile to the drive able to control the corresponding motor. According to another feature, the crane is able to perform a second kind of movement in the horizontal plane (Slewing movement) about a vertical rotation axis (Slewing axis). In this feature, the drive controller calculates the speed profile of this second movement and it supplies the information of the speed profiles to the drives able to control the motors relative to the two considered movements.

3. BRIEF DESCRIPTION OF THE DRAWINGS

[0035] The attached figures illustrate the present invention and, together with the detailed description of the invention, are able to explain the most important characteristics of the invention.

FIG.1 shows a Building (Slewing) crane model corresponding to the present invention. Specifically, the three axes, "z" in vertical direction, "r" along the trolley direction movement and "φ" along the slewing direction movement are put in evidence in the picture.

FIG.2 shows a graph of the two swaying angle module profiles for the crane control system relative to the present invention. The first one is relative to the generated profile of the swaying angle module (in degrees) along the radial direction (direction x, along the r axis), versus time. The second one is relative to the generated profile of the swaying angle module (in degrees) along the direction perpendicular to the radial direction (direction y, perpendicular to the r axis), versus time.

FIG.3 shows a graph of the profile of the derivative (as regard the time) of the trolley position **r** as a function of the time, obtained filtering input commands on both directions (radial and slewing commands) in accordance with the theory of the present invention. The specific values are: $u_{1(C)} = 0$ and $u_{2(C)} = 0.2$

FIG.4 shows the same kind of graph of Fig.3. The specific values are:

$$u_{1(C)} = 0.1 \text{ and } u_{2(C)} = 0$$

[0036] These pictures show clearly the characteristic profile for an anti-sway velocity profile.

[0037] **FIG.5** shows, in analogous way to the previous pictures, a graph of the profile of the product of the position **r** multiplied for the derivative (as regard the time) of the slewing position φ, as a function of the time. In the analogous way to the previous picture, this profile is obtained filtering input commands on both directions (radial and slewing commands) in accordance with the theory of the present invention.

[0038] The specific values are: $u_{1(C)} = 0$ and $u_{2(C)} = 0.2$

[0039] FIG.6 shows the same kind of graph of Fig.5. The specific values are:

$$u_{1(C)} = 0.1 \text{ and } u_{2(C)} = 0$$

[0040] Also these last pictures show clearly the characteristic profile for an anti-sway velocity profile, in this case for the derivative as regard the time of the slewing direction.

[0041] **FIG.7** represents a simplified diagram of the regulating device according to this invention, used to move a load along two horizontal, Trolley and Slewing axes.

4. DETAILED DESCRIPTION OF THE INVENTION

[0042] The following detailed description of the invention includes a description of a slewing crane that is of interest and a derivation of the equations of the motion that represent the dynamics of the defined degree-of-freedom of the slewing crane. The presented embodiment includes a computer-controlled interface (a Plc control) between operator and crane in order to perform movement of the payload by the operator without sway of the same payload.

[0043] Referring to **FIG. 1,** a slewing crane includes a multi-body system with three independent degrees-of-freedom for positioning a spherical-pendulum as the payload of mass **m.** Specifically, the slewing crane includes a translating load-line, having length **L** and a payload of mass **m** attached to a moving or translating trolley with mass **M.** A crane operator or a computer positions the payload using some available commands and changing the load length **L.**

[0044] The crane configuration is characterized by one vertical axis (in Fig.1, axis z), a first horizontal axis, the radial axis (in Fig.1, axis **r)** in the direction of the trolley displacement, and a second horizontal axis, perpendicular to the radial axis **r** (in Fig.1, axis **y**). There are three controlled motions for slewing crane corresponding to these three axes: a trolley translation, a rotation of the crane about his vertical axis **z** (slewing motion) and a vertical motion along the axis **z** with variation of the length **L.**

[0045] Therefore, in order to describe the motion of the slewing crane, an equivalent kinematics scheme with concentrated masses is represented in FIG. 1.

[0046] Dynamics of the generalized slewing crane is represented as a multi-body system with 5 independent degrees of freedom:

$q_1 = r$:	radial position of the trolley
$q_2 = \varphi$:	slewing angle (rotation about the z axis)
$q_3 = l$:	hoisting position of the crane (along the z axis).
$q_4 = \varphi_x$:	sway angle tangential to the radial direction
$q_5 = \varphi_y$:	sway angle normal to the radial direction

[0047] If we refer to the energy, the following balance can be defined.

4.1 Potential Energy

[0048] The Potential energy of the load is:

$$V = -pz + C \quad \text{(Eq.1)}$$

[0049] Considering the position z along the vertical (with reference to the FIG.2), we have:

$$z = l\cos\varphi_x \cos\varphi_y \quad \text{(Eq.2)}$$

[0050] And, therefore, it is obtained:

$$V = -l\left(\cos\varphi_x \cos\varphi_y\right)(p) \equiv \delta L = -mgl\cos\varphi_x \cos\varphi_y \quad \text{(Eq.3)}$$

$$U = -V = mgl\cos\varphi_x \cos\varphi_y \quad \text{(Eq.4)}$$

4.2 Kinetics Energy

[0051] The Kinetics energy is the sum of the corresponding terms for the tower $\mathbf{T_T}$, for the boom $\mathbf{T_B}$, for the trolley $\mathbf{T_R}$ and for the load $\mathbf{T_L}$.

$$T = T_T + T_B + T_R + T_L \quad \text{(Eq.5)}$$

$$T_T = \frac{1}{2} J_T \dot{\varphi}^2 \quad \text{(Eq.6)}$$

$$T_B = \frac{1}{2} m_B r_B^2 \dot{\varphi}^2 \qquad \text{(Eq.7)}$$

$$T_R = \frac{1}{2} m_R (r^2 \varphi^2 + \dot{r}^2) \qquad \text{(Eq.8)}$$

$$T_L = \frac{1}{2} m_L v_L^2 \qquad \text{(Eq.9)}$$

[0052] In order to simplify the final system of Lagrange equations, we take into account only small sway angles. That means the following assumptions:

$$\sin\varphi_x \approx \varphi_x, \sin\varphi_y \approx \varphi_y, \cos\varphi_x \approx 1, \cos\varphi_y \approx 1 \qquad \text{(Eq.10)}$$

[0053] With this assumption, the payload velocity V_L can be obtained defining his expression in terms of the Cartesian coordinates and, after, those coordinates in terms of the Lagrangian coordinates. In this way, it is possible to obtain, for the term $\mathbf{T_L}$, the following expression:

$$T_L = \frac{1}{2} m_L \left\{ \begin{array}{l} \dot{\varphi}^2 (l\varphi_x + r)^2 + (\varphi_x \dot{l} + \dot{\varphi}_x l + \dot{r})^2 + (l\dot{\varphi}\varphi_y)^2 + (\dot{l}\varphi_y + l\dot{\varphi}_y)^2 + \dot{l}^2 + \\ (l\dot{\varphi}_x\varphi_x)^2 + (l\dot{\varphi}_y\varphi_y)^2 + \\ +2\dot{\varphi}(l\varphi_x + r)(\dot{l}\varphi_y + l\dot{\varphi}_y) - 2\dot{\varphi}(\varphi_x\dot{l} + \dot{\varphi}_x l + \dot{r})(l\varphi_y) + \\ -2\dot{\varphi}_x\varphi_x l\dot{l} - 2\dot{\varphi}_y\varphi_y l\dot{l} + 2\dot{\varphi}_x\dot{\varphi}_y\varphi_x\varphi_y l^2 \end{array} \right\}$$

(Eq.11)

4.3 Lagrange Equations

[0054] Establishing the Lagrange function as:

$$L = T + U \qquad \text{(Eq.12)}$$

[0055] it is possible to apply the Lagrange equations

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right) - \frac{\partial L}{\partial q_i} = Q_i \qquad \text{(Eq.13)}$$

[0056] Defining the five components for the generalized forces $\mathbf{Q_i}$ in the following way:

$$
\begin{aligned}
Q_1 &= F_r, \\
Q_2 &= M, \\
Q_3 &= F_z, \\
Q_4 &= F_x, \\
Q_5 &= F_y
\end{aligned}
\qquad \text{(Eq.14)}
$$

where F_r, M, F_z, F_x, Fy are the generalized, non-conservative,
dissipative and active forces working on the system. Particularly F_x and F_y represent the components of air resistance forces acting on the payload and on the hoisting system and the components of forces due to the damping of the rotation movement due to the structure's elasticity of the hoisting system.

[0057] At the end of our calculations, we are able to obtain the five Lagrange equations in the corresponding five Lagrangian coordinates and their derivative as regards the time:

$$
\begin{aligned}
&(m_R + m_L)\ddot{r} - m_L l \varphi_y \ddot{\varphi} + m_L \varphi_x \ddot{l} + m_L l \ddot{\varphi}_x - \\
&\dot{\varphi}^2 \left[m_R r + m_L (r + l\varphi_x)^2 \right] + 2m_L \left(\dot{\varphi}_x \dot{l} - \varphi_y \dot{l} \dot{\varphi} - \varphi_y l \dot{\varphi} \right) = F_r
\end{aligned}
$$

(Eq.15)

$$
\begin{aligned}
&J_H \ddot{\varphi} - m_L l \varphi_y \ddot{r} + m_L r \varphi_y \ddot{l} + m_L r l \ddot{\varphi}_y + \\
&2\left\{ m_L \left(l\varphi_x \dot{r}\dot{\varphi} + l\dot{\varphi}_x r\dot{\varphi} + \dot{l}\varphi_x r\dot{\varphi} + r\dot{r}\dot{\varphi} + \dot{l}\dot{\varphi}_y r \right) + m_R r\dot{r}\dot{\varphi} \right\} = M
\end{aligned}
$$

(Eq.16)

$$
\varphi_x \ddot{r} + \ddot{l} + r\varphi_y \ddot{\varphi} - r\varphi_x \dot{\varphi}^2 + 2\dot{r}\dot{\varphi}\varphi_y - g = F_z / m_L
$$

(Eq.17)

$$
\begin{aligned}
&\left\{ \ddot{r} - l\varphi_y \ddot{\varphi} + l\ddot{\varphi}_x - (r + l\varphi_x)\dot{\varphi}^2 + 2\left(\dot{l}\dot{\varphi}_x - l\dot{\varphi}_y \dot{\varphi} - \dot{l}\varphi_y \dot{\varphi} \right) + g\varphi_x \right\} \\
&- \left(\dot{l}^2 \varphi_x \right) / l = F_x / m_L
\end{aligned}
$$

(Eq.18)

$$\left\{(r+l\varphi_x)\ddot{\varphi}+l\ddot{\varphi}_y-l\varphi_y\dot{\varphi}^2+2\left(\dot{l}\dot{\varphi}_y+\dot{r}\dot{\varphi}+l\dot{\varphi}_x\dot{\varphi}+\dot{l}\varphi_x\dot{\varphi}\right)+g\varphi_y\right\}$$
$$-\left(\dot{l}^2\varphi_y\right)\Big/l=F_y\Big/m_L$$

(Eq.19)

[0058] So, the multiple degree of freedom rotary crane system results defined by the five differential equations where the terms relative to the Centrifugal and Coriolis forces appear.

4.4 Equations Solution

[0059] Because our goal is to obtain a strategy in order to reduce the oscillatory motion of rotary cranes payloads, we'll concentrate on the last equations (Eq. 18) and (Eq. 19) involving the two variables φ_x and φ_y corresponding to the two oscillatory degrees of freedom.

4.4.1 Equations Solution in the first approximation

[0060] To simplify the equations of motion, in the first approximation, the terms in $\dot{\varphi}^2$ (terms relative to the centrifugal force) and in the coefficients of **2nd** grade (terms relative to the Coriolis force) are neglected in the equations (Eq. 18) and (Eq. 19) and so it is obtained the following simplified system, retaining terms of order less than two:

$$\begin{cases}\ddot{\varphi}_x+\left(g/l\right)\varphi_x-\ddot{\varphi}\,\varphi_y=-\ddot{r}/l+F_x/m_L\\ \ddot{\varphi}_y+\ddot{\varphi}\,\varphi_x+\left(g/l\right)\varphi_y=-\left(r/l\right)\ddot{\varphi}+F_y/m_L\end{cases}$$

(Eq.20)

[0061] This system can be represented, in matrix terms, in the following way:

$$\underline{\underline{I}}\cdot\underline{\ddot{\Phi}}(t)+\underline{\underline{K}}\cdot\underline{\Phi}(t)=\underline{u}(t)$$

(Eq.21)

[0062] Where $\underline{I}$ is the Identity matrix, while $\underline{K}$, $\underline{u}$ and $\underline{\Phi}$ are the following matrices:

$$\underline{\underline{K}}=\begin{bmatrix}g/l & -\ddot{\varphi}\\ \ddot{\varphi} & g/l\end{bmatrix}$$

(Eq.22)

$$\underline{u}(t)=\begin{bmatrix}-\ddot{r}/l+F_x/m_L\\ -\ddot{\gamma}(r/l)+F_y/m_L\end{bmatrix}$$

(Eq.23)

$$\underline{\Phi}(t)=\begin{bmatrix}\varphi_x\\ \varphi_y\end{bmatrix},\ \underline{\ddot{\Phi}}(t)=\begin{bmatrix}\ddot{\varphi}_x\\ \ddot{\varphi}_y\end{bmatrix} \qquad \text{(Eq.24)}$$

[0063] The eigenvalues are obtained by solving the following homogeneous equation:

$$\det\left[\underline{\underline{K}}-\lambda\underline{\underline{I}}\right]=0 \qquad \text{(Eq.25)}$$

[0064] In this way, we can obtain:

$$\lambda_{1,2}=\left(g/l\right)\pm i\ddot{\varphi} \qquad \text{(Eq.26)}$$

[0065] While the matrix of the eigenvectors is obtained, known the eigenvalues, by solving the homogeneous equation associated to the equation (Eq.21):

$$\underline{\underline{Z}}=\left[\underline{Z}_1,\underline{Z}_2\right]=\begin{bmatrix}1 & 1\\ -i & i\end{bmatrix} \qquad \text{(Eq.27)}$$

[0066] With the obtained eigenvectors, being the transformation defined as:

$$\underline{\Phi}(t)=\underline{\underline{Z}}\,\underline{\Psi}(t)\ , \qquad \text{(Eq.28)}$$

the equation (Eq.21) can be transformed in the following equation:

$$\underline{\underline{I}}\cdot\underline{\ddot{\Psi}}(t)+\underline{\underline{\Lambda}}\cdot\underline{\Psi}(t)=\underline{U}(t) \qquad \text{(Eq.29)}$$

[0067] In the last equation, the matrix $\underline{\Lambda}$ of the eigenfrequencies associated to the eigenvectors results diagonalized, as must be after the defined linear transformation, in the new modal coordinates $\Psi_i(t)$:

$$\underline{\underline{\Lambda}}=\begin{bmatrix}\left(g/l\right)+i\ddot{\varphi} & 0\\ 0 & \left(g/l\right)-i\ddot{\varphi}\end{bmatrix} \qquad \text{(Eq.30)}$$

[0068] Considering the eigenfrequencies defined in (Eq.30), it is a natural consequence to define a set of path notch filters able to change the profile of the speed commands from the operators to the trolley and slewing axes, in order to reduce the sway of the payload.

[0069] In fact, in the Laplace domain, the decoupled system of equations (Eq.29) becomes:

$$\Psi_1(s) = \frac{U_1(s) + [s\,\Psi_1(0) + \Psi_1'(0)]}{(s^2 + \Gamma_1)}$$

$$\Psi_2(s) = \frac{U_2(s) + [s\,\Psi_2(0) + \Psi_2'(0)]}{(s^2 + \Gamma_2)} \qquad \text{(Eq.31)}$$

[0070] Where $\Psi_i(s)$ are the Laplace transforms of the modal coordinates $\Psi_i(t)$, $U_i(s)$ are the Laplace transforms of the operator commands $U_i(t)$ according to the modal coordinates, and $\Psi_i'(0)$ are the derivative of the Laplace transforms for the modal coordinates in correspondence to $s = 0$.

[0071] In the hypothesis to start with:

$$\Psi_i(0) = 0\,;\,\Psi_i'(0) = 0\;, \qquad \text{(Eq.32)}$$

from (Eq.31) it follows then the following equation, considering also (Eq.30) and assuming that the jib angular acceleration $\ddot{\varphi}$ and load-line length l are constant over each sample period of the programmable logic controller:

$$\Psi_1(s) = \left[\frac{1}{s^2 + \left(\frac{g}{l} + i\ddot{\varphi}\right)}\right] U_1(s)$$

$$\Psi_2(s) = \left[\frac{1}{s^2 + \left(\frac{g}{l} - i\ddot{\varphi}\right)}\right] U_2(s) \qquad \text{(Eq.33)}$$

[0072] In order to obtain unity steady-state system gain of the filtered inputs, these filtered inputs are chosen in the following way:

$$\underline{U_F}(s) = \underline{\underline{P}}\,\underline{U_{(C)}}(s)\;, \qquad \text{(Eq.34)}$$

where

$$U_F(s) = \begin{bmatrix} U_1(s) \\ U_2(s) \end{bmatrix} \qquad \text{(Eq.35)}$$

and

$$\underline{\underline{P}}=\begin{bmatrix}\dfrac{K_1\left\{s^2+\Lambda_1\right\}}{\left(s+\alpha_1\right)^3} & 0\\ 0 & \dfrac{K_2\left\{s^2+\Lambda_2\right\}}{\left(s+\alpha_2\right)^3}\end{bmatrix}. \qquad \text{(Eq.36)}$$

[0073] In (Eq.36) K_1 and K_2 are defined as:

$$K_1=\frac{\left(\alpha_1^{\ 3}\right)}{\left(\Lambda_1\right)}U_{1(C)}$$
$$K_2=\frac{\left(\alpha_2^{\ 3}\right)}{\left(\Lambda_2\right)}U_{2(C)} \qquad \text{(Eq.37)}$$

[0074] In the previous (Eq.37) $U_{1(C)}$ and $U_{2(C)}$ are the not-filtered input, that is they are the constant profiles of the jib angular acceleration and of the trolley acceleration usually used for the drives that control the motors relative to the two above defined movements.

[0075] Practically, $U_{1(C)}$ and $U_{2(C)}$ are the sum of the commanded quantities (commanded by the crane operator) for the jib angular acceleration and of the trolley acceleration and of the air resistance forces acting on the payload and on the hoisting system and of the forces due to the damping of the rotation movement due to the structure's elasticity of the hoisting system. Load-length l and trolley position r can be continuously measured (for example) by the use of encoders.

[0076] The parameters α_1 and α_2 in (Eq.37) are used in order to define the profiles of the filters and so the amount of the roll-off after the notch.

[0077] The functions defined in (Eq.33) are able to generate two notch filters corresponding to the modal frequencies.

[0078] From the vector $\underline{U_F}(s)$ can be obtained the corresponding anti-transformed of Laplace vector $\underline{u_F}(t)$.

[0079] At the end, integrating on the time, the profiles of the jib angular velocity and of the trolley velocity can be obtain.

[0080] Furthermore, from the linear transformation in modal coordinates expressed in (Eq.28), and being worth (Eq.24), can be obtained the equations representative of the two components of the vector $\underline{\Phi}(t)$, that is the equations relative to the two sway angles, respectively tangential and normal to the radial direction **r:**

$$\begin{cases}\varphi_x=\alpha^3\left(g/l\right)(1-i)\left\{\dfrac{t^2}{2}e^{-\alpha t}\right\}\\ \varphi_y=\alpha^3\left[\left(g/l\right)-\ddot{\varphi}\right](1+i)\left\{\dfrac{t^2}{2}e^{-\alpha t}\right\}\end{cases} \qquad \text{(Eq.38)}$$

[0081] Where i is the imaginary unit. Computing the absolute value of these quantities, it is possible to have the profiles of the two sway angles:

$$\begin{cases} |\varphi_x| = \sqrt{2}\, A \\ |\varphi_y| = \sqrt{2}\, B \end{cases} \qquad \text{(Eq.39)}$$

where *A and B* are defined as:

$$\begin{cases} A = \alpha^3 \left(g/l \right) \left\{ \frac{t^2}{2} e^{-\alpha t} \right\} \\ B = \alpha^3 \left[\left(g/l \right) - \ddot{\varphi} \right] \left\{ \frac{t^2}{2} e^{-\alpha t} \right\} \end{cases} \qquad \text{(Eq.40)}$$

4.4.2 Equations Solution: not linear terms

[0082] In order to consider also the contribution of the not linear terms in (Eq. 18) and (Eq. 19), we start from the solution obtained in first approximation (only linear terms) and defined from the vector $\underline{\Phi}(t)$ of (Eq. 28) and from the vector $\underline{U_F}(s)$ of (Eq.34), as described in the previous paragraph.

[0083] Starting from the vector $\underline{U_F}(s)$ can be obtained the corresponding anti-transformed of Laplace vector $\underline{u_F}(t)$. The components of this vector are:

$$\begin{aligned} u_{F(1)}(t) &= \ddot{\varphi}_{(0)}(t) \\ u_{F(2)}(t) &= \ddot{r}_{(0)}(t) \end{aligned} \qquad \text{(Eq.41)}$$

where $\ddot{\varphi}_{(0)}(t)$ and $\ddot{r}_{(0)}(t)$ are the second derivative respect to the time of the angle slewing rotation and of the radial trolley position, obtained with the linear approximation defined in the previous paragraph. Integrating on the time, the corresponding first derivative $\dot{\varphi}_{(0)}(t)$ and $\dot{r}_{(0)}(t)$ respect to the time of the angle slewing rotation and of the radial trolley position can be obtained.

[0084] With the goal to obtain the complete solution of (Eq. 18) and (Eq. 19), the terms relative to the centrifugal force and to the Coriolis force cannot be neglected in the above equations. Therefore, we define:

$$\begin{aligned} \dot{\varphi}(t) &\equiv \dot{\varphi}_{(0)}(t) + \Delta\dot{\varphi}(t) \\ \dot{r}(t) &\equiv \dot{r}_{(0)}(t) + \Delta\,\dot{r}(t) \end{aligned} \qquad \text{(Eq.42)}$$

where $\Delta\dot{\varphi}(t)$ and $\Delta\,\dot{r}(t)$ are the solutions of the following equations, obtained from (Eq. 18) and (Eq. 19), considering only the not linear terms:

$$\left(r + l\varphi_x\right)\left[\dot{\varphi}_{(0)}(t) + \Delta\dot{\varphi}(t)\right]^2$$
$$+ 2\left(l\dot{\varphi}_y + \dot{l}\varphi_y\right)\left[\dot{\varphi}_{(0)}(t) + \Delta\dot{\varphi}(t)\right] - 2\,\dot{l}\dot{\varphi}_x = 0$$

(Eq.43)

$$l\varphi_y\left[\dot{\varphi}_{(0)}(t) + \Delta\dot{\varphi}(t)\right]^2$$
$$-2\left\{\left[\dot{r}_{(0)}(t) + \Delta\dot{r}(t)\right] + l\dot{\varphi}_x + \dot{l}\varphi_x\right\}\left[\dot{\varphi}_{(0)}(t) + \Delta\dot{\varphi}(t)\right] - 2\,\dot{l}\dot{\varphi}_y = 0$$

[0085] The term l, having a slow variation on the time as regards the other term $\Delta\dot{\varphi}$ and $\Delta\dot{r}$, can be considered constant during the fast variation of these variables.

[0086] Besides, the terms containing $\dot{\varphi}_{(0)}(t)$ and containing $\dot{r}_{(0)}(t)$ can be considered known, being obtained from the integration of (Eq.41). Therefore, with the following assumptions:

$$A \equiv \left(r + l\varphi_x\right)$$
$$B \equiv 2\left(r + l\varphi_x\right)\dot{\varphi}_{(0)}(t) + 2\left(l\dot{\varphi}_y + \dot{l}\varphi_y\right)$$
$$C \equiv \left(r + l\varphi_x\right)\left[\dot{\varphi}_{(0)}(t)\right]^2 + 2\left(l\dot{\varphi}_y + \dot{l}\varphi_y\right)\left[\dot{\varphi}_{(0)}(t)\right] - 2\,\dot{l}\dot{\varphi}_x$$
$$D \equiv l\varphi_y$$
$$E \equiv 2\left(l\varphi_y\right)\left[\dot{\varphi}_{(0)}(t)\right] - 2\left[\dot{r}_{(0)}(t) + l\dot{\varphi}_x + \dot{l}\varphi_x\right]$$
$$F \equiv -2\left[\dot{\varphi}_{(0)}(t)\right]$$
$$G \equiv -2$$
$$H \equiv \left(l\varphi_y\right)\left[\dot{\varphi}_{(0)}(t)\right]^2 - 2\left[\dot{r}_{(0)}(t) + l\dot{\varphi}_y + \dot{l}\varphi_y\right]\left[\dot{\varphi}_{(0)}(t)\right] - 2\,\dot{l}\dot{\varphi}_y$$

(Eq.44)

it follows that the system of (Eq.43) assumes the following expression:

$$\mathrm{A}\left[\Delta\dot{\varphi}(t)\right]^2 + B\left[\Delta\dot{\varphi}(t)\right] + C = 0$$

(Eq.45)

$$\mathrm{D}\left[\Delta\dot{\varphi}(t)\right]^2 + E\left[\Delta\dot{\varphi}(t)\right] + F\left[\Delta\dot{r}(t)\right] +$$
$$+ G\left[\Delta\dot{\varphi}(t)\right]\left[\Delta\dot{r}(t)\right] + \mathrm{H} = 0$$

[0087] In the hypothesis that the two sway angles in the tangential and in the normal direction are defined from (Eq.38), the above equations system (Eq.45) can be considered as a system of two quadratic equations in the unknown variables $\Delta\dot{\varphi}$ and $\Delta\dot{r}$.

[0088] Solving this system as regards of the unknown variables $\Delta\dot{\varphi}$ and $\Delta\dot{r}$, we obtain the additional terms necessary in the system of (Eq.42) in order to have the exact solution of the (Eq. 18) and (Eq. 19), in this way incorporating the effect of the Centrifugal and Coriolis forces.

[0089] The device, defined in this invention, is therefore able to be used to control, with the use of a programmable logic controller (plc) device, the sway of the payload during all the movements of the crane, both jib rotation and trolley translation, using equation (42) to define the velocity profile for each movement along the two horizontal axes.

[0090] The control device does not include, in advantageous way, neither any preliminary modelling step nor the measurement of specific physical parameters such as the sway angle or a measurement of the current flowing in the motor.

[0091] The control device above described is designed to be installed in an automation system which is responsible to control the movement of the load. This automation system includes a variable speed drive for the trolley (translation) movement and a variable speed drive for the slewing (rotation) movement. The control device can be installed or directly in the variable speed drives or can include also a programmable logic controller which is used to supply the speed references to the variable speed drives.

5. DESCRIPTION OF THE DEVICE AND AN EXAMPLE OF

EMBODIMENT OF THE INVENTION

[0092] As previously described, the control process relative to this invention does not require to know variable parameters such as the measure of the angle of swing or a measure of the current flowing in the motor.

[0093] For this, the present invention is realized with a device that includes means for determining the length of cables for the load suspension and his variation in the time, means for determining the position of the suspension point of the payload along the translation movement, means for determining the position of the suspension point of the payload along the rotation movement, means for determining air resistance forces acting on the payload and on the hoisting system and means for determining the damping of the rotation movement due to the structure's elasticity of the hoisting system.

[0094] According to one feature, a drive controller calculates the speed profile of the first movement relative to the trolley (Trolley movement) and it supplies the information of the speed profile to the drive able to control the corresponding motor.

[0095] According to another feature, the crane is able to perform a second kind of movement in the horizontal plane (Slewing movement) about a vertical rotation axis (Slewing axis). In this feature, the drive controller calculates the speed profile of this second movement and it supplies the information of the speed profiles to the drives able to control the motors relative to the two considered movements.

[0096] **Fig. 7** shows a simplified diagram of the control device according to this invention relative to the movement of a load along two horizontal axis (slewing and trolley). The control device **10** here defined is been implemented in a hoist such as a tower crane.

[0097] The control device **10** includes means to have information representative of the length L of the suspension cable of the payload. These means can be, as in the realized example, an encoder associated with the hoisting motor. But other means could be, in example, an encoder associated directly with the drum of the cable or several limit switch sensors on the entire run of the cable having predetermined level values as a function of the limit switch positions.

[0098] Again, the control device **10** includes also means to have information representative of the air resistance forces acting on the payload and on the hoisting system and means for determining the damping of the rotation movement due to the structure's elasticity of the hoisting system. The first means (relative to the information about air resistance forces) can be, as in the realized example, obtained with an anemometer able to measure direction and velocity of the wind.

[0099] The second means (relative to the information about damping of the rotation due to the elasticity of the structure) can be, as in the realized example, obtained with calculation means for determining the elasticity of the structure.

[0100] In general the control device **10** is made with an estimator module **11** connected with an input-output module **12.**

[0101] Practically, the estimator module **11** receives the inputs from the input-output module **12** (in example the length L of the cable and the set speeds $V_{Set\_Trolley}$ and $V_{Set\_Slewing}$ from the operator) and computes the speed profiles necessary to obtain the anti-sway functionality to send to the devices able to command the motors relative to the movements of the tower crane along the trolley and slewing directions.

[0102] Concretely, the control device **10** is been realized using a Plc where is computed the velocity profiles for the two movements (Trolley and Slewing) using the method described above.

[0103] Obviously, it is necessary that the used Plc has sufficient computation velocity in order to perform the complex calculations with a short cycle time (not more of 20 ms) in order to obtain an high precision of the velocity profiles to give to the drives that control the motors. Nevertheless, at today it is quite easy to find a Plc able to satisfy this requirement.

[0104] Concerning the command of the movement along the direction of the trolley, it is possible to say that it can be performed using a variable speed drive $D_{Trolley}$ that commands the motor $M_{Trolley}$. In turn the drive $D_{Trolley}$ receives the speed profile reference $\dot{r}(t)$ computed from the control device **10.**

[0105] Corresponding, the command of the movement along the direction of the slewing, it is performed using a variable speed drive $D_{Slewing}$ that commands the motor $M_{Slewing}$. In turn the drive $D_{Slewing}$ receives the speed profile reference $\dot{\varphi}(t)$ computed from the control device **10.**

[0106] At the end, the hoisting movement along the axis **Z** of the **Fig. 1** is performed using an hoisting motor (not shown in **Fig. 10)** that can be controlled by a variable speed drive.

[0107] The operator, typically, provides to define a reference signal speed set-point $V_{Set\_Trolley}$ for the Trolley direction and $V_{Set\_Slewing}$ for the Slewing direction. That concretely is realized using a combiner as a joystick in the radio-command instrument. That was realized in the experimental apparatus on the tower crane in order to realize the control of the movements.

[0108] Sometimes, however, there are applications where the tower crane is controlled automatically and, therefore, the set-point $V_{Set\_Trolley}$ for the Trolley direction and $V_{Set\_Slewing}$ for the Slewing direction come directly from an automation equipment, that can coincide or not with the Plc used to control the anti-sway functionality.

[0109] Specifically, the physical values provided in Table 1 were used in the experimental system realized for the embodiment of this invention.

TABLE 1

Physical quantities used for the experimental apparatus of the embodiment

PARAMETERS	UNITS	VALUE
L (cable length)	m/s	11.1
$\ddot{r}$(t) (Trolley acceleration set)	m/s^2	0.2
$\dot{\varphi}$(t) (Slewing acceleration set)	$rad/\ s^2$	0.1

[0110] The most relevant technical problems met were:

1. A relevant technical problem is that if no translation (trolley) movement is requested by the automation system (or by the operator), the final position of the suspension point on the trolley must be identical to its final position. In other words the final distance $r(t)$ at the time t at the end of the rotation movement must be equal to the initial distance $r(0)$ at the time 0 at the start of the rotation. For this reason, the estimator module 11 of the control device 10 must store the initial position $r(0)$ at the time 0 and, at the end of the rotation movement, applies a suitable correction to the position on the trolley, in order to return the suspension point to its initial position.
2. Another relevant practical problem was the correct definition of the set speed of the two movements, trolley and slewing. In fact, the correct development of the anti-sway profile is extremely depending from the exact value of the set speed in terms of m/s or rad/s. If the translation of unit of measure from the values in Hz (typically used in industrial environment) for the speed command of the variable speed drives to the.international units (m/s or rad/s) is wrong (at level of thousandths) the generated profile will be not sufficiently précised. So, it is necessary to have particular attention to these values.
3. Concerning the precision with the length of the cable must be known, it is possible to say that the precision it doesn't be high. So, if it is optimal the use of an encoder, as described above, also the use of several limit switch sensors on the entire run of the cable having predetermined level values is possible with sufficient precision and correctness of the result in the anti-sway functionality.

Claims

1. Control system for controlling the sway produced from the movement of a payload suspended by a variable length load-line to a suspension point of an hoisting system, the suspension point being able to perform both a rotation movement in the horizontal plane about a vertical rotation axis and a translation movement in the horizontal plane, the payload experiencing a sway defined by a tangential angle to the translation axis and by a normal angle to the translation axis, **characterized in that** the control system is comprising:

a) means for generating a signal relative to the said length of the cable connected to the suspension point of the payload;
b) means for receiving said signal relative to the said length of the cable connected to the suspension point of

the payload in the control system;
c) means for generating a signal relative to the said vertical speed of the variable length of the cable connected to the suspension point of the payload;
d) means for receiving said signal relative to the said vertical speed of the variable length of the cable connected to the suspension point of the payload in the control system;
e) means for generating a signal relative to the said position of the suspension point of the payload along the rotation movement in the horizontal plane;
f) means for receiving said signal relative to the said position of the suspension point of the payload along the rotation movement in the control system;
g) means for generating a signal relative to the said position of the suspension point of the payload along the translation movement in the horizontal plane;
h) means for receiving said signal relative to the said position of the suspension point of the payload along the translation movement in the control system;
i) means for generating a signal relative to the air resistance forces acting on the payload and on the hoisting system;
j) means for receiving said signal relative to the said air resistance forces in the control system;
k) means for receiving data relative to the dynamic characteristics of the hoisting system;
l) calculation means for determining:

l.1) the sway angle tangential to the translation axis of the said suspension point of the payload,
l.2) the speed of said sway angle tangential to the translation axis,
l.3) the sway angle normal to the translation axis of the said suspension point of the payload,
1.4) the speed of said sway angle normal to the translation axis;

m) calculation means for determining the damping of the rotation movement in the horizontal plane due to the structure's elasticity of the hoisting system;
n) calculation means for determining:

n.1) the speed of the said suspension point of the payload along the rotation movement in the horizontal plane,
n.2) the speed of the said suspension point of the payload along the translation movement in the horizontal plane;

o) means **characterized by** motor controller for generating a signal relative to the speed of the said suspension point of the payload along the rotation movement in the horizontal plane;
p) means **characterized by** motor controllers for generating a signal relative to the speed of the said suspension point of the payload along the translation movement in the horizontal plane.

2. Control system of claim **1, characterized in that** the calculation means determine:

a. the sway angle tangential to the translation axis of the said suspension point of the payload,
b. the speed of said sway angle tangential to the translation axis,
c. the sway angle normal to the translation axis of the said suspension point of the payload,
d. the speed of said sway angle normal to the translation axis,
e. the speed of the said suspension point of the payload along the rotation movement in the horizontal plane,
f. the speed of the said suspension point of the payload along the translation movement in the horizontal plane,

solving a system of differential equations that take into account also the physical effects of second order relative to the centrifugal force and relative to the Coriolis force acting on the said hoisting system.

3. Control system of claim **2, characterized in that** the said means for determining air resistance forces acting on the payload and on the hoisting system further includes an anemometer able to measure direction and velocity of the wind.

4. An automation system configured to control the movement of a payload suspended by a variable length load-line to a suspension point of the hoisting system, **characterized in that** the said automation system includes a control system as claimed in claim 2.

5. Method for controlling the sway produced from the movement of a payload suspended by a variable length load-line to a suspension point of an hoisting system, the suspension point being able to perform both a rotation movement in the horizontal plane about a vertical rotation axis and a translation movement in the horizontal plane, the payload experiencing a sway defined by a tangential angle to the translation axis and by a normal angle to the translation axis, **characterized in that** the method is comprising the following steps:

a) generating a signal relative to the said length of the cable connected to the suspension point of the payload
b) receiving said signal relative to the said length of the cable connected to the suspension point of the payload in the control system;
c) generating a signal relative to the said vertical speed of the variable length of the cable connected to the suspension point of the payload;
d) receiving said signal relative to the said vertical speed of the variable length of the cable connected to the suspension point of the payload in the control system;
e) generating a signal relative to the said position of the suspension point of the payload along the rotation movement in the horizontal plane;
f) receiving said signal relative to the said position of the suspension point of the payload along the rotation movement in the control system;
g) generating a signal relative to the said position of the suspension point of the payload along the translation movement in the horizontal plane;
h) receiving said signal relative to the said position of the suspension point of the payload along the translation movement in the control system;
i) generating a signal relative to the air resistance forces acting on the payload and on the hoisting system;
j) receiving said signal relative to the said air resistance forces in the control system;
k) receiving data relative to the dynamic characteristics of the hoisting system;
l) calculating:

l.1) the sway angle tangential to the translation axis regarding the said suspension point of the payload,
1.2) the speed of said sway angle tangential to the translation axis,
l.3) the sway angle normal to the translation axis regarding the said suspension point of the payload,
l.4) the speed of said sway angle normal to the translation axis;

m) calculating the damping of the rotation movement in the horizontal plane due to the structure's elasticity of the hoisting system;
n) calculating:

n.1) the speed of the said suspension point of the payload along the rotation movement in the horizontal plane,
n.2) the speed of the said suspension point of the payload along the translation movement in the horizontal plane;

o) generating by using motor controller a signal relative to the speed of the said suspension point of the payload along the rotation movement in the horizontal plane;
p) generating by using motor controller a signal relative to the speed of the said suspension point of the payload along the translation movement in the horizontal plane.

6. Control method of claim 5, **characterized in that** the calculations that determine:

a. the sway angle tangential to the translation axis of the said suspension point of the payload,
b. the speed of said sway angle tangential to the translation axis,
c. the sway angle normal to the translation axis of the said suspension point of the payload,
d. the speed of said sway angle normal to the translation axis,
e. the speed of the said suspension point of the payload along the rotation movement in the horizontal plane,
f. the speed of the said suspension point of the payload along the translation movement in the horizontal plane,

are obtained solving a system of differential equations that take into account also the physical effects of second order relative to the centrifugal force and relative to the Coriolis force acting on the said hoisting system.

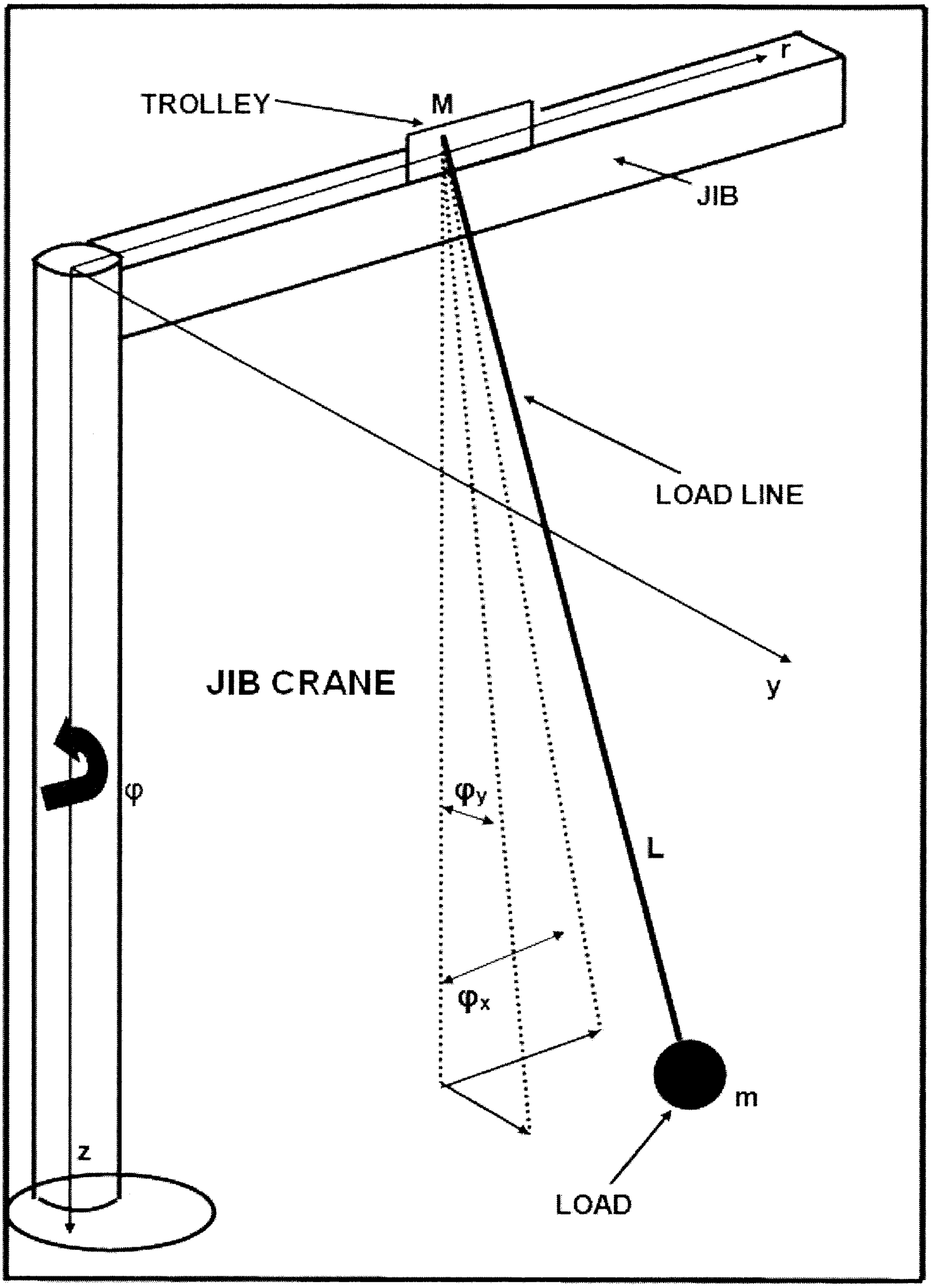


FIG. 1

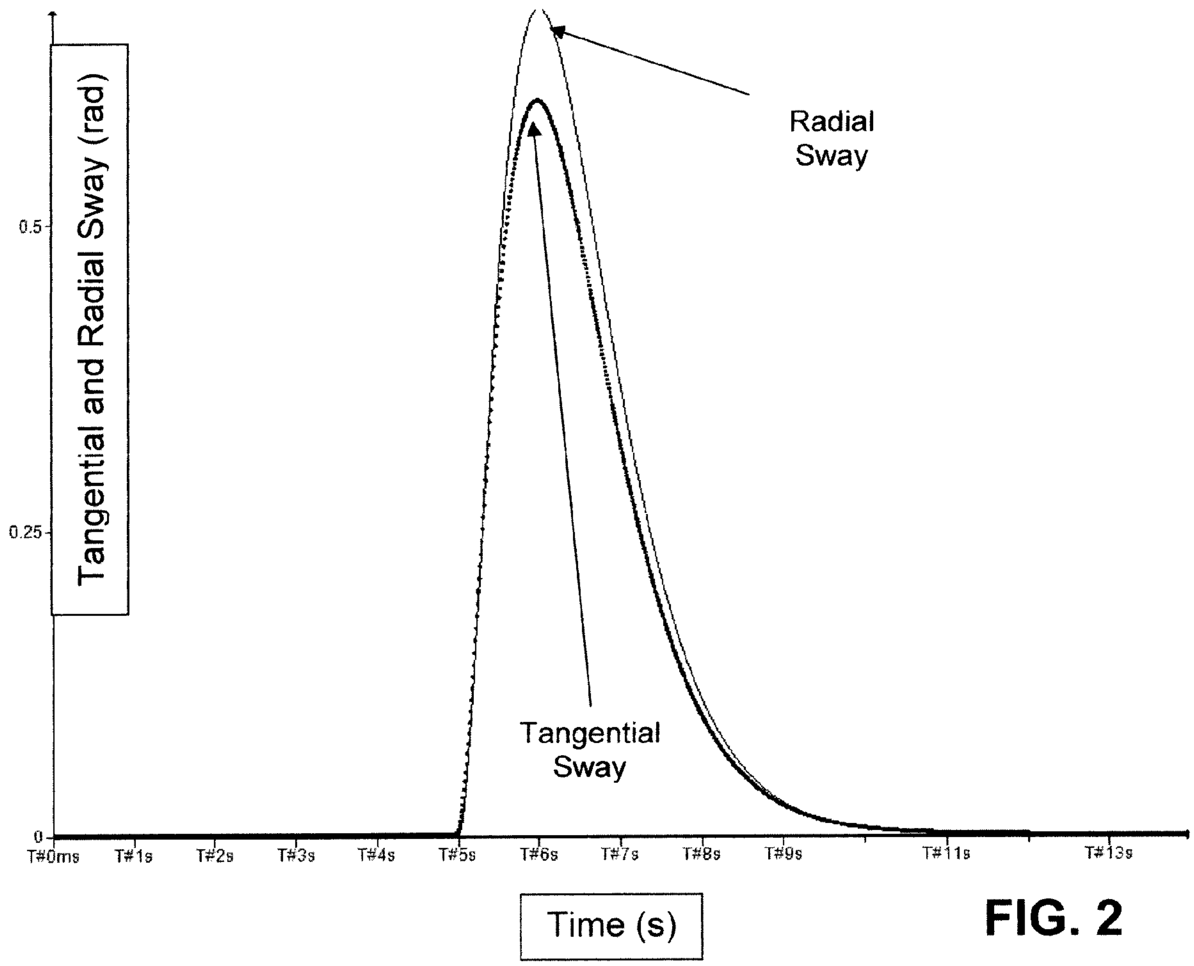
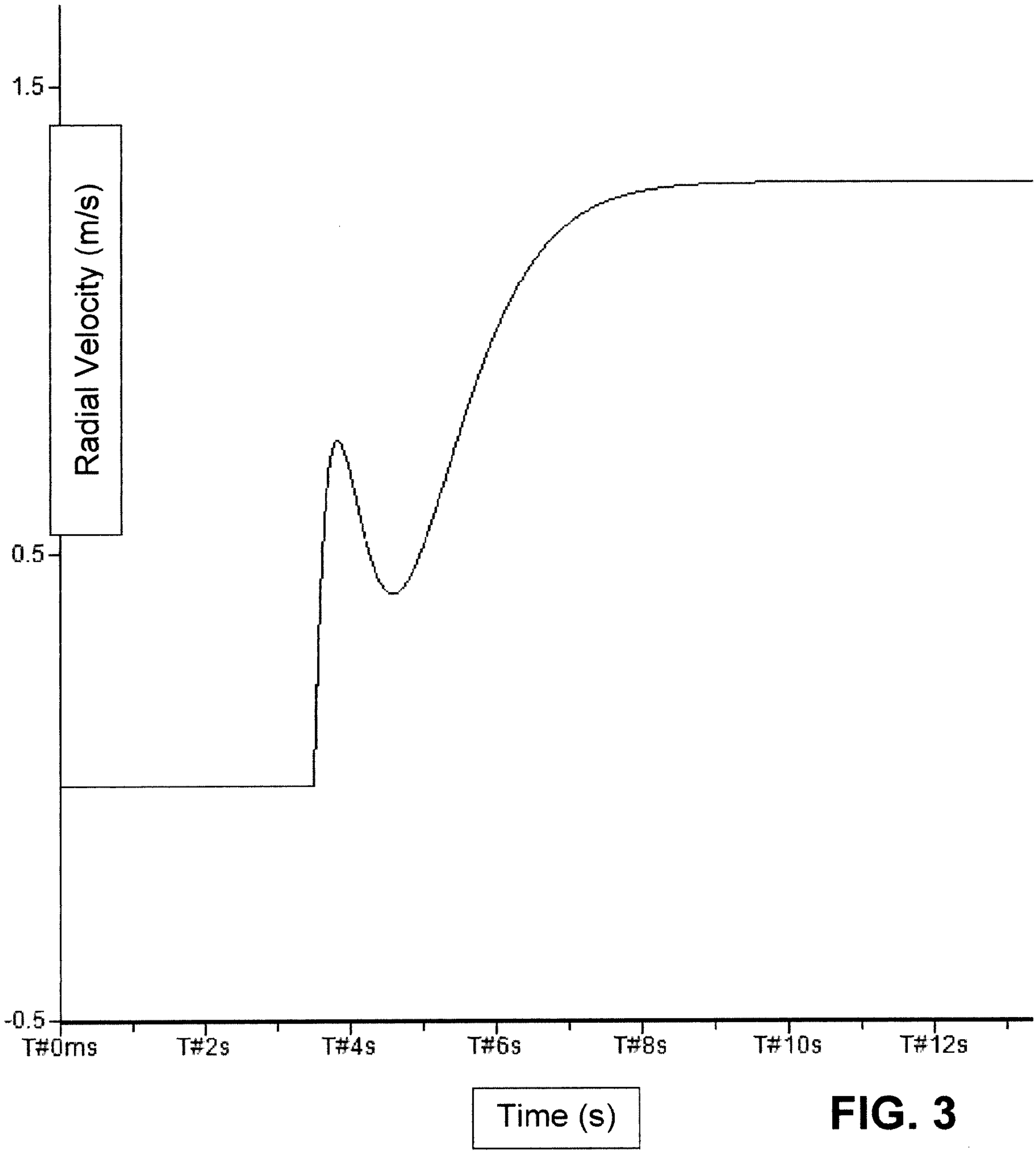

Tangential and Radial Sway (rad)
0.5
0.25
0
Radial Sway
Tangential Sway
T#0ms
T#1s
T#2s
T#3s
T#4s
T#5s
T#6s
T#7s
T#8s
T#9s
T#11s
T#13s
Time (s)

FIG. 2



1.5
0.5
-0.5
Radial Velocity (m/s)
T#0ms
T#2s
T#4s
T#6s
T#8s
T#10s
T#12s
Time (s)

FIG. 3

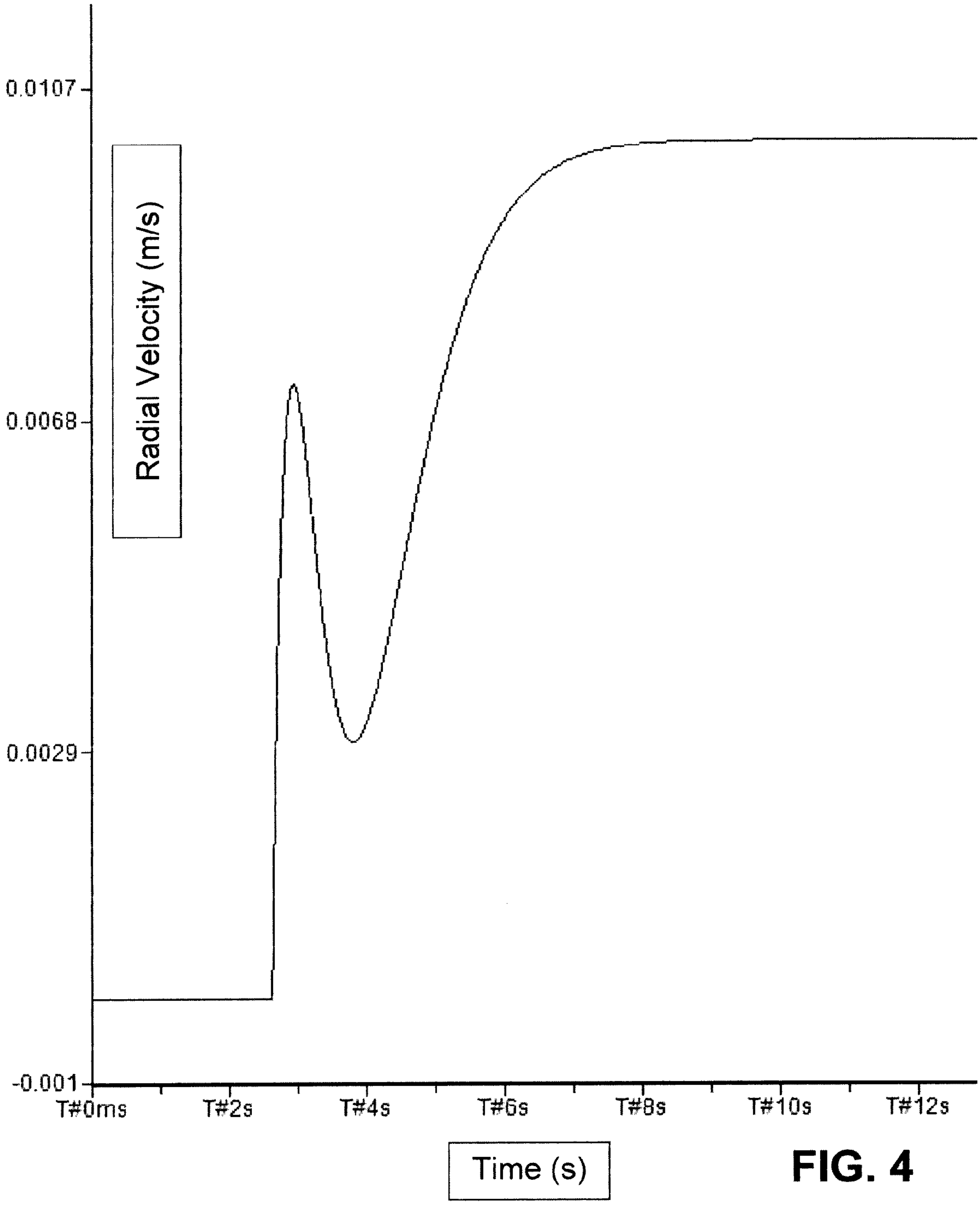

Radial Velocity (m/s)
0.0107
0.0068
0.0029
-0.001
T#0ms
T#2s
T#4s
T#6s
T#8s
T#10s
T#12s
Time (s)

FIG. 4

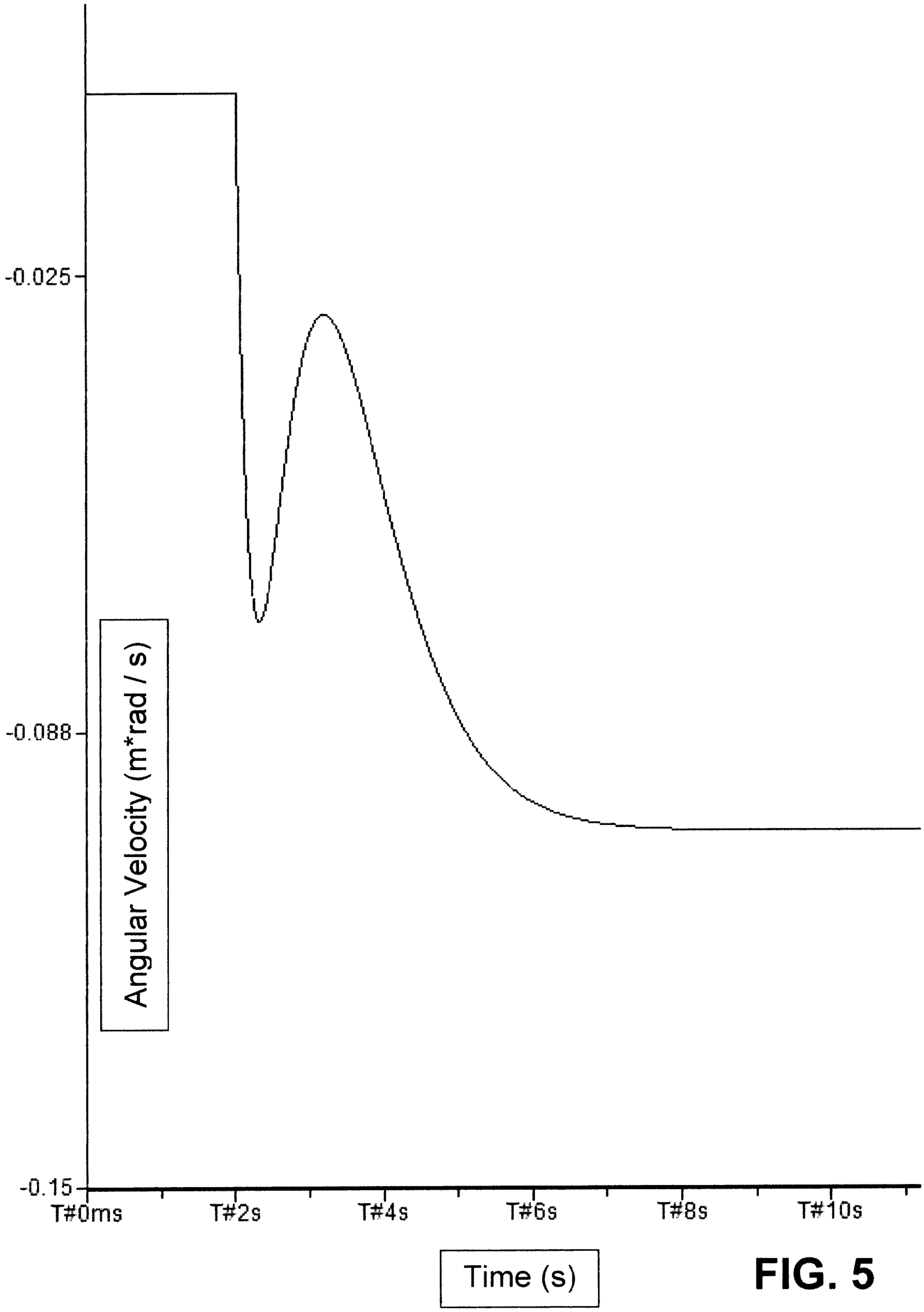

-0.025
-0.088
-0.15
Angular Velocity (m*rad / s)
T#0ms
T#2s
T#4s
T#6s
T#8s
T#10s
Time (s)

FIG. 5

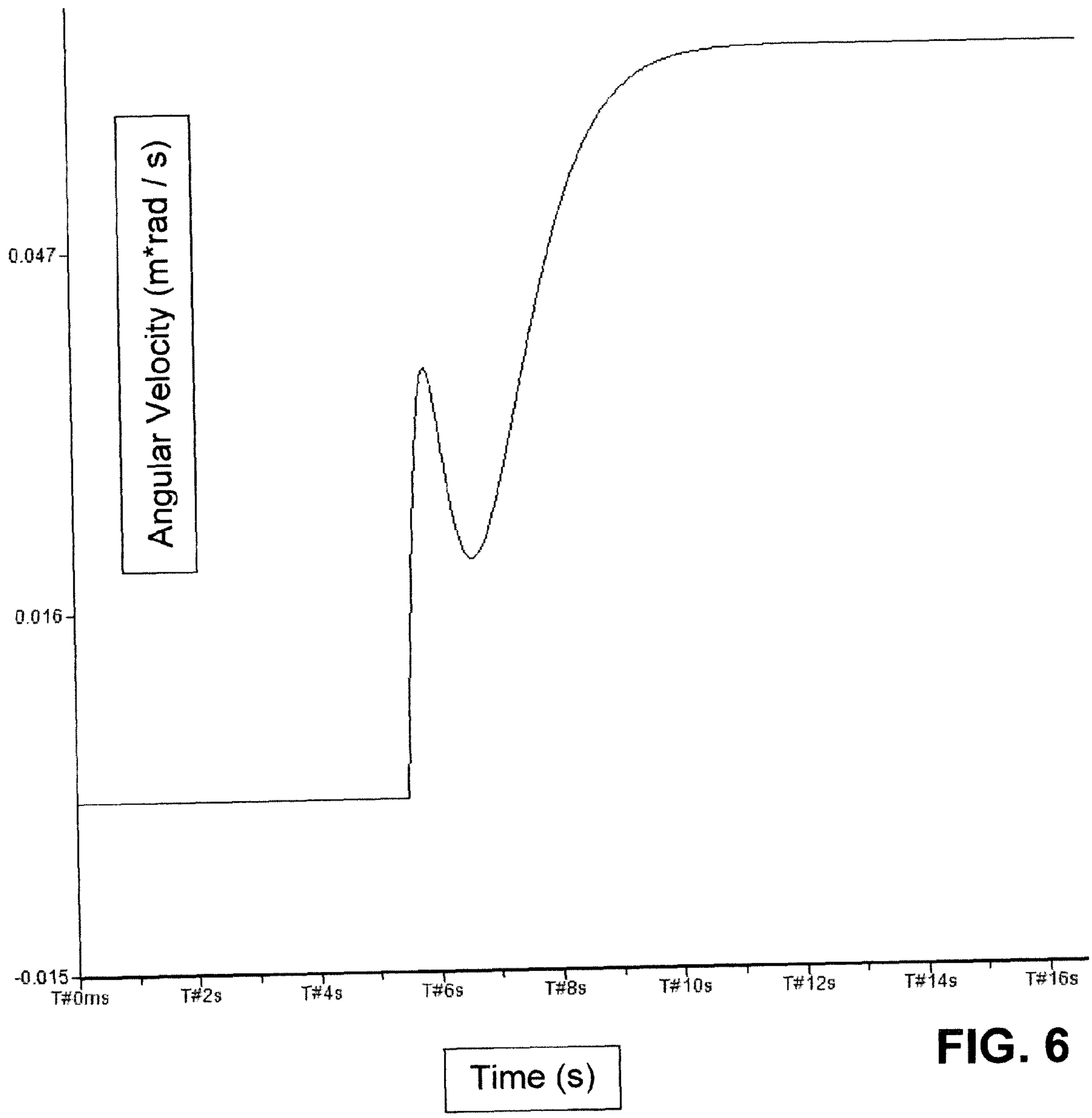
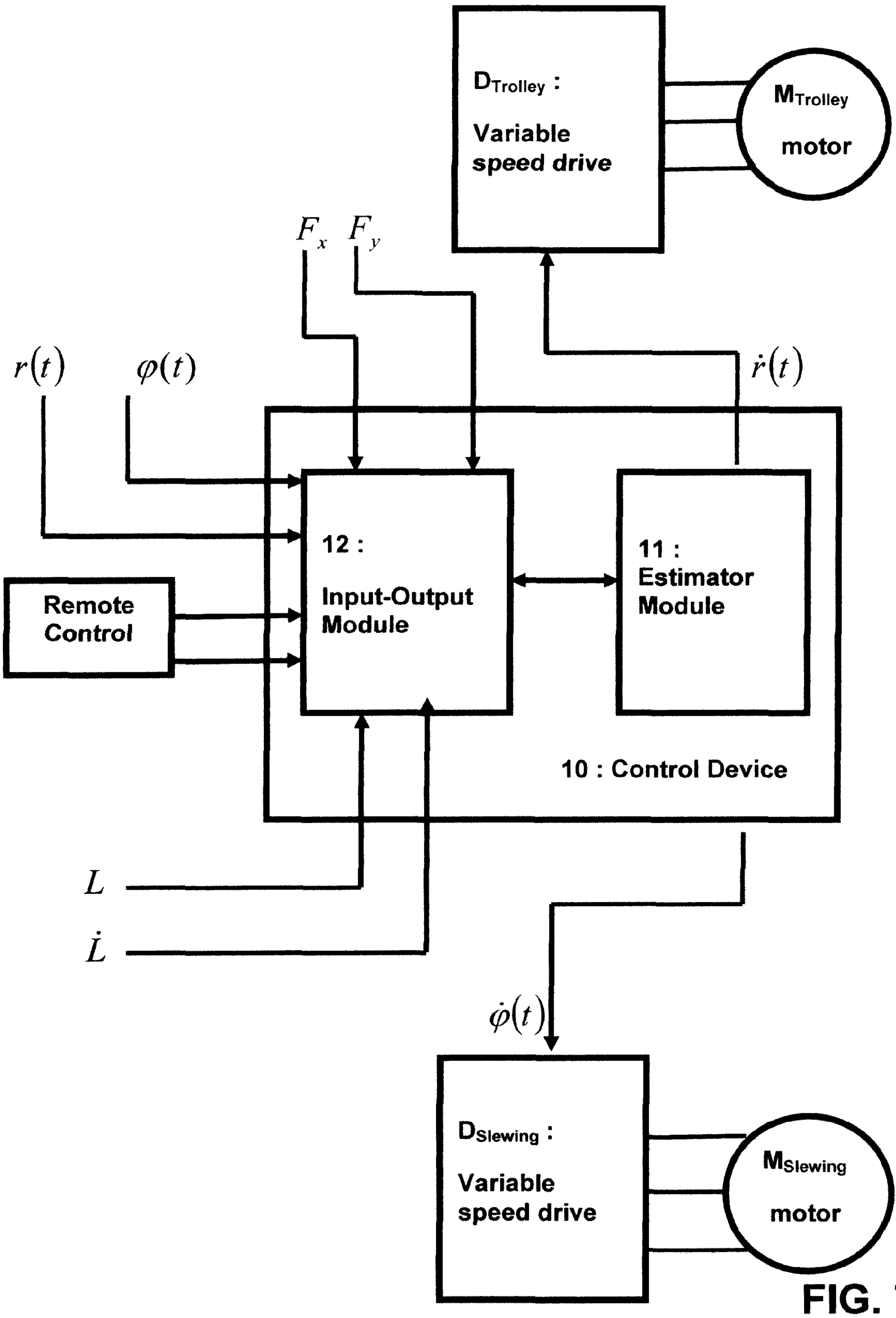

Angular Velocity (m*rad / s)
0.047
0.016
-0.015
T#0ms
T#2s
T#4s
T#6s
T#8s
T#10s
T#12s
T#14s
T#16s
Time (s)

FIG. 6



$D_{Trolley}$:
Variable speed drive
$M_{Trolley}$
motor
F_x F_y
$r(t)$
$\varphi(t)$
$\dot{r}(t)$
Remote Control
12 :
Input-Output Module
11 :
Estimator Module
10 : Control Device
L
$\dot{L}$
$\dot{\varphi}(t)$
$D_{Slewing}$:
Variable speed drive
$M_{Slewing}$
motor

FIG. 7

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EUROPEAN SEARCH REPORT

Application Number
EP 14 42 5148

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Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	WO 97/45357 A1 (SIEMENS AG [DE]; WIENKOP UWE [DE]; MEIER CHRISTOF [DE]) 4 December 1997 (1997-12-04) * page 3 - page 4 * * page 6 - page 7 * * figure 1 * -----	1-6	INV. B66C13/06
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TECHNICAL FIELDS SEARCHED (IPC)

B66C

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	15 May 2015	Serôdio, Renato

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15-05-2015

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