A FAST SEA-KEEPING SIMULATION METHOD FOR HEAVY-LIFT OPERATIONS BASED ON MULTI-BODY SYSTEM DYNAMICS

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ABSTRACT

This paper presents a fast numerical method to analyze heavy-lift operations of ships in short crested waves. For this purpose, a sea-keeping simulation method for the coupled motions of a heavy-lift vessel and a freely suspended load is developed. The method considers the motions of the ship in six degrees-of-freedom and the suspended load as a point mass. The coupling of the multi-rigid-body system of the ship and the suspended load is considered by solving the equation of roll motion together with the Euler-Lagrange equations of the load. This approach allows the simulation of several hours of real time motion in short crested waves within only a few seconds. Consequently, the method is particularly suitable when very long or numerous sea-keeping simulations or statistical results are required. Moreover, the method is applied to evaluate the sea-keeping capabilities of a heavy-lift vessel during a lifting operation conducted offshore in 2013.

INTRODUCTION

Nowadays, more and more offshore heavy-lift operations are conducted by heavy-lift vessels. Modern heavy-lift vessels are equipped with high-capacity dynamic positioning systems. Only these advanced dynamic positioning systems make heavy-lift operations under unfavorable environmental conditions possible in the first place. However, the positioning of the ship is not the only challenge for heavy-lift operations with wind and swell. Usually the limitations for controlling the motions of the freely suspended load are more stringent. Compared to semi-submersible crane vessels and crane barges, heavy-lift vessels generally have more slender hulls and less transverse stability (see Fig. 1). Hence, the roll motion of a heavy-lift vessel and its coupling with the motions of the freely suspended load are particularly critical for a heavy-lift operation. Therefore, numerical tools to assess the coupled motions of a heavy-lift vessel and the load are of utmost importance for both the planning of lifting operations and the design ships. For the purpose of developing these fast simulation tools, the research project "HoOK: Hochsee-Operationen mit Kranen" (offshore operations with cranes) has been initiated in spring 2013 [1, 2].
The R&D Project HoOK

HoOK is a joint research project between Hamburg University of Technology and several industry partners. The project is funded by the Federal Ministry of Economics and Technology with the overall objective to support the domestic shipbuilding industry as well as the German energy turnaround from fossil fuels to renewables. The special purpose of the HoOK project within this framework is to develop software tools which can identify and enhance potential cost reductions in the installation of offshore wind farms [1,2]. To this end, a single software package is under development which shall be able to simulate the complete lifting operation of a floating crane in time domain. All motions of the vessel, the crane and its load, and the necessary dynamic positioning and ballasting operations shall be considered in the simulation. In the end, the entire operation shall be simulated as a whole with automated safety evaluation. In order to identify hazards due to transient and non-periodic motions like the rupture of crane rope or load impacts on guides or bumpers or the cargo deck, a time domain approach is chosen.

A further reason for the selection of the time domain approach is the reliable analysis of resonance conditions. These resonance conditions could occur at certain configurations of floating cranes [4,5] and it is known from sea-keeping accidents of ships that resonance or near resonance conditions can cause substantial hazards. During a lifting operation the crane configuration varies strongly and consequently the likelihood to hit the resonance configurations for at least a short duration is high. Schellin et al. showed that only non-linear time-domain analysis can provide reliable quantitative results for crane ships near resonance in general [6].

Within the HoOK project, this paper presents the first step towards the desired simulation goals specified above. So far, ballasting or dynamic positioning operations or operations with multiple hooks or cranes are not yet included in this study. In fact, the presented method is rather an enhancement of the existing sea-keeping method E4-ROLLS by Söding and Kröger [7–9]. Like E4-ROLLS, the focus and the advantage of the presented sea-keeping method lie in the computational efficiency by lean modelling. Hence, the method is particularly suitable when very long or numerous sea-keeping simulations or statistical results are required for a single crane operation.

KINEMATIC MODEL

The underlying kinematic model of the sea-keeping analysis method is explained in this section. Figure 2 shows the model of the heavy-lift vessel and its suspended load as well as the used nomenclature.

Ship Kinematics

The motions of the ship are considered in six degrees-of-freedom. To describe these motions, two Cartesian frames of reference are used. The earthfixed frame \( \{E, \xi, \eta, \zeta\} \) is located in the center of gravity of the ship in the initial equilibrium position \( E \). The second Cartesian frame of reference is fixed to the ship and is denoted by \( \{S, x, y, z\} \). The origin of \( \{S, x, y, z\} \) always lies in the center of gravity of the ship. Hence, the translatory motions of the ship can be described by the vector

\[
r_{ES} = [\xi, \eta, \zeta]^{T}
\]
which points from \( E \) to \( S \). The rotational motions of the ship are described by the Tait-Bryan angles \( \varphi, \theta \) and \( \psi \), denoting the roll, pitch and yaw angle. Hence the coordinate transformation matrix \( \mathbf{R}_{ES} \) is given by Eqn. (2).

\[
\mathbf{R}_{ES} = \begin{bmatrix}
\cos \psi & -\sin \psi & 0 \\
\sin \psi \cos \theta & \cos \psi \cos \theta & -\cos \psi \sin \theta \\
0 & \sin \theta & 0
\end{bmatrix}
\]

\[
\mathbf{R}_{ES} = \begin{bmatrix}
\cos \varphi & 0 & \sin \varphi \\
0 & 1 & 0 \\
-\sin \varphi & 0 & \cos \varphi
\end{bmatrix}
\]

\[
\mathbf{R}_{ES} = \begin{bmatrix}
\cos \psi \cos \theta & \cos \psi \sin \theta & -\sin \psi \\
\sin \varphi \sin \psi \cos \theta + \cos \varphi \sin \theta & \sin \varphi \sin \psi \sin \theta - \cos \varphi \cos \theta & \sin \varphi \cos \psi
\end{bmatrix}
\]

\[
\mathbf{R}_{ES} = \begin{bmatrix}
\cos \alpha & -\sin \alpha & 0 \\
\sin \alpha & \cos \alpha & 0 \\
0 & 0 & 1
\end{bmatrix}
\]

Here, we use \( c_x \) and \( s_x \) as an abbreviation for \( \cos x \) and \( \sin x \). The order of the three successive rotations is indicated by Eqn. (2) and Eqn. (3). The latter equation transforms ship fixed coordinates to earth fixed coordinates.

\[
E \mathbf{r} = \mathbf{R}_{ES} S \mathbf{r}
\]

Here, the prefix \( E \) of the vector \( \mathbf{r} \) indicates that coordinates are given in the earth fixed frame. Coordinates in the ship fixed frame are labeled with the prefix \( S \) respectively.

**Suspended Load Kinematics**

As shown in Fig. 2, the suspended load is modeled as a point mass \( m \) located in \( M \). In this study we neglect the elasticity of the crane ropes which results in a prescribed, time-dependent distance \( l(t) \) between the location of the mass \( m \) and the crane tip \( T \). The resulting rheonomic, holonomic constraint is

\[
r_{TM}^T r_{TM} = l(t).
\]

During the simulated crane operation not only the suspended length \( l(t) \) can be controlled but also the crane itself. The desired crane position is considered by a given, time-dependent position vector

\[
s \mathbf{r}_{ST}(t) = (x(t), y(t), z(t))^T,
\]

which describes the position of the crane tip in the ship fixed frame \( S \times x, y, z \).

In order to increase the numerical efficiency of the dynamic analysis, we describe the position of the load by general coordinates and introduce the load fixed reference frame \( \{T, x', y', z'\} \). The origin of the load fixed reference frame is always located at the crane tip \( T \) and the frame is fixed to the load in the way that its \( z' \)-axis is always parallel to the crane rope. Hence, the position of the suspended load is

\[
r_{TM} = (0, 0, -l)^T
\]

in the load fixed frame \( \{T, x', y', z'\} \). We obtain the orientation of the load fixed frame by two successive rotations about the \( x \) and the rotated \( y \) axis of the ship fixed frame. Here, the two rotation angles \( \alpha \) and \( \beta \) are the general coordinates and can be interpreted as the roll and pitch angles of the load. Similar to the roll and pitch angles of the ship, we obtain a rotation matrix \( \mathbf{R}_{ST} \) from \( \alpha \) and \( \beta \).

\[
\mathbf{R}_{ST} = \begin{bmatrix}
\cos \beta & 0 & \sin \beta \\
0 & 1 & 0 \\
-\sin \beta & 0 & \cos \beta
\end{bmatrix}
\]

\[
\mathbf{R}_{ST} = \begin{bmatrix}
c_\beta & s_\beta c_\alpha & c_\alpha c_\beta \\
p_\alpha & p_\beta c_\alpha & s_\alpha c_\beta \\
-\sin \beta & \cos \beta & \cos \alpha
\end{bmatrix}
\]

In line with Eqn. (3), the coordinate transformation between the ship fixed frame and the load fixed frame is performed by Eqn. (8).

\[
S \mathbf{r} = \mathbf{R}_{ST} T \mathbf{r}
\]

Hence, the position of the load in the inertial frame is

\[
E \mathbf{r}_{EM} = E \mathbf{r}_{ES} + R_{ES} S \mathbf{r}_{ST} + R_{ST} (0, 0, -l)^T.
\]

**DYNAMIC ANALYSIS**

The concept of the dynamic analysis is based on the sea-keeping code \( E4-ROLLS \) (E4-roll simulation) which was first established by Söding [7] and further improved by Kröger [8,9]. A brief and comprehensive summary of the method can be found in [10,11].

**E4-ROLLS Method**

\( E4-ROLLS \) was developed to simulate accurately roll motions and capsize events in steep waves. In order to compute capsize frequencies and roll motions for a large number of different sea states, ship headings and speeds, the sea-keeping code was designed to be extremely fast and focussed on the roll motion. The other degrees-of-freedom of the ship are computed in \( E4-ROLLS \) as well but less accurate, because only their influence on the roll motion are of interest. The focus on efficient roll motion computations led to the following separated treatment of the
degrees-of-freedom of the ship: The heave, sway, yaw and pitch motions are assumed to be small and less influenced by non-linear effects than the surge or the roll motion. Therefore these motions are computed in the frequency domain by means of a linear strip method. Their response amplitude operators (RAOs) are then used to create motion history signals in irregular seaways by superposition of reactions in regular waves. The roll and surge motions, by contrast, depend strongly non-linearly on the wave and motion amplitude and thus are computed non-linearly. The equations of motions are separately solved and integrated in time domain. The following equation is used to describe the roll motion:

\[
\ddot{\phi} = \frac{n_{e,y} + n_d - m_i (g + \ddot{z}) h + I_{xx} \left( (\dot{\psi} + \psi \dot{\phi}) c_p - (\ddot{\vartheta} + \ddot{\phi} \dot{c_p}) s_p \right)}{I_{xx} - I_{xz} (\dot{s_p} + \dot{c_p})}.
\]  

(10)

In this equation \(n_{e,y}\) denotes the linear wave exciting roll moment computed in frequency domain. It also includes the linear coupling roll moment due to sway and yaw motions. \(n_d\) is the roll damping moment, \(m_i\) the mass of the ship, \(g\) the gravity acceleration, \(h\) the instantaneous righting leverarm, \(I_{xx}\), the product of inertia of the roll and yaw motion, and \(I_{xx}\) the moment of inertia of the roll motion.

**Enhancement**

In contrast to *E4-ROLLS*, our study does not only focus on the sole roll motion of the ship but also on the motion of the suspended load and its coupling to the ship motions. Hence, we do not solve the equation of roll motion separately but together with the load motions. One possibility to obtain the coupled equations of motion is to set up the Newton-Euler equations for both rigid bodies and solve them together with the rheonomic, holonomic constraint of Eqn. (4). This procedure requires the consideration of six degrees-of-freedom for the ship and three degrees-of-freedom for the suspended load as well as Lagrangian multipliers \(\lambda\) for the unknown constraint force. In addition, the algebraic constraint (4) and the ordinary differential equations of the Newton-Euler equations form a system of differential-algebraic equations (DAE) which requires special integration schemes. In short, the Newton-Euler approach is computationally intensive and not in line with the initial focus of the underlying method on computational efficiency and speed.

Many commercial multi-body sea-keeping programs overcome the problem of solving DAEs by substituting the rheonomic, holonomic constraint (4) by a stiff spring element. Hence, the constraint becomes an ordinary differential equation (ODE). The drawback of this approach is the requirement for small time integration steps to avoid numeric instabilities when solving ODEs with very stiff elements. Moreover, this formulation of the constraint does not any longer reduce the degrees-of-freedom of the multi-body system. Hence, if the goal is to obtain a computationally efficient method, then this approach is inferior.

Therefore, we applied another approach. In a first step and similar to in *E4-ROLLS*, we separate the degrees-of-freedom of the system in two groups. The first group consists of the degrees-of-freedom, whose motions are small and which do not underlie strong non-linear effects. The second group is formed by the degrees-of-freedom, whose motions are relatively large and which are strongly governed by non-linear effects. According to *E4-ROLLS* the sway, heave, pitch, and yaw motions of the ship are in the first group which is combined in the vector \(x\). Because of their linear characteristics, these motions are computed in frequency domain by a radiation-diffraction strip method. The coupling effect of the suspended load motion on them is small therefore neglected. For this reason we denote \(x\) as the vector of uncoupled motions and apply the following linearization to its components \(x_i\), and their derivatives:

\[
\begin{align*}
\cos x_i & \approx 1, \\
\sin x_i & \approx x_i, \quad \forall i, j \in \{1,...,5\} \\
x_i \cdot x_j & \approx 0
\end{align*}
\]

Here, the surge motion is treated in a distinctive manner. In *E4-ROLLS*, the surge motion is treated non-linearly in order to account for surf-riding and strong added wave resistance. Both of these non-linear effects primarily occur at higher ship speeds in steep waves. In contrast, heavy-lift operations are usually conducted in calm and moderate sea states and at zero forward speed. Consequently, the surge motion can be treated linearly in heavy-lift simulations. The only reason why the surge motion is computed non-linearly in this method is to take advantage of the existing software architecture of *E4-ROLLS*. Nevertheless, we assign the surge motion to the vector of uncoupled motions \(x\).

The roll and the suspended load motions form the second group, which is denoted by the vector \(y = (\phi, \alpha, \beta)^T\). These motions are relatively large and depend strongly non-linearly on the wave and motion amplitude. Therefore, these motions are computed non-linearly and all coupling effects are considered between these motions. Additionally, the coupling of \(x\) on \(y\) is considered in this direction. In the following, we denote \(y\) as the vector of coupled motions.

Furthermore, we assume that crane motions are slow compared to wave induced ship and load motions. Hence, the time derivatives of the time dependent prescribed variables \(s_{RF}(t), l(t)\), etc. vanish in the equations of motion.

In a second step, we solve the Euler-Lagrangian equations of the coupled generalized motions \(y\). Thus, we obtain the equations of motions (11) in minimal coordinates:

\[
M(x, y)\ddot{y} + k(x, x, \dot{x}, y, y) = q(x, y).
\]  

(11)
Here, $M$ denotes the mass matrix, $k$ the vector of the generalized non-linear and damping terms, $q$ the vector of the generalized external forces. The mass matrix and the vector of the generalized external forces are presented in detail in Eqn. (12) and (13).

$$M = \begin{bmatrix} M_{1,1} & M_{1,2} & M_{1,3} \\ ml (y_s s\alpha c_\beta + l_c \beta - z_a c_\alpha) & ml^2 & 0 \\ ml s\beta (y_s + l_s \alpha) & 0 & ml^2 c_\alpha \end{bmatrix},$$ \hspace{1cm} (12)

$$q = \begin{pmatrix} n_{e, sy} + n_d - m_s (g + \ddot{\zeta})h + g m (z_a s\phi - x_a \psi - y_a c\phi) \\ g ml (\dot{\phi}s\alpha s\beta + s\alpha c\beta c\phi + c\alpha s\phi) \\ -g ml (\dot{\phi} c\beta + s\beta c\phi) \end{pmatrix}.$$ \hspace{1cm} (13)

The elements $M_{1,1}$, $M_{1,2}$, and $M_{1,3}$ can be found in the appendix. For no suspended load ($m = 0$) Eqn. (11) becomes the roll motion equation (10) of the well validated sea-keeping code E4-ROLLS with the enhancement of the consideration of non-symmetric mass distributions.

**Internal Damping Forces**

It is common practice to use tugger lines during an offshore heavy-lift operation. These tugger lines are ropes which are connected nearly horizontally between the ship and the suspended load. The length or rather the tension of the tugger line is controlled by a winch with the overall purpose to dampen the motions of the suspended load. In the simulation we use a simplified modelling approach to consider the tugger lines. The damping effect of the tugger lines is taken into account by adding 30% of critical linear damping to the suspended load motions described by $\alpha$ and $\beta$. This tugger line damping is included in Eqn. (11) by an additional damping term $B\dot{y}$.

**External Forces**

In this study, we consider gravitational, hydrostatic and hydrodynamic forces acting on the multi-rigid-body system. Apart from the gravitational forces on the load, we used similar external roll moments as Söding and Kröger used in E4-ROLLS (see. [7–9]) The external forces and moments are in detail:

- **Gravitational force acting on the suspended load $m$:**
  
  $$E = (0, 0, -g m)^T.$$ \hspace{1cm} (14)

- **Gravitational force acting on the ship:** This force does not result in a roll moment because the reference point $S$ lies in the center of gravity of the ship.

- **Vertical ship acceleration:** There exists a hydrodynamic coupling moment between the heave and the roll motion. Thus, the heave motion results in a roll moment. We assume that the hydrodynamic coupling moment can be approximated by the hydrodynamic mass for the heave motion $m_{33}$ acting in the center of buoyancy of the instantaneous displacement. Hence, the roll moment is
  
  $$E_{nT} = (m_{33} \ddot{h}, 0, 0)^T.$$ \hspace{1cm} (15)

  The instantaneous leverarm $h$ in irregular waves is computed very efficiently by the equivalent wave concept. This concept is a further development of Grim’s effective wave concept [12], which approximates the irregular water surface at the midship plane by a harmonic wave whose length is equal to the length of the ship. Hence, the a leverarm lookup table can be computed prior to the simulation. This procedure reduces the numerical effort for leverarm computation during the simulation to simple interpolation.

- **Froude-Krylov pressure acting on the hull:** The roll moment due to the undisturbed hydrodynamic pressure is
  
  $$E_{nFK} = (-\rho \nabla g h, 0, 0)^T.$$ \hspace{1cm} (16)

  Here, $\rho \nabla$ is the instantaneous displacement of the ship. As well as the leverarm $h$ the instantaneous displacement can be computed by the equivalent wave concept. However, Söding avoids this computation by using a simplified equation of heave motion:

  $$E_{nFK} = \left( -m_s \ddot{\zeta}, \rho \nabla g - m_s g \right).$$ \hspace{1cm} (17)

  Here, $m_s$ denotes the mass of the ship. Thus, Eqn. (15-17) are combined to Eqn. (18) which does not include the instantaneous displacement anymore.

  $$E_{nFK} + E_{nT} = \left( -m_s (g + \ddot{\zeta}) h, 0, 0 \right)^T.$$ \hspace{1cm} (18)

- **Linear wave exciting moment and linear coupling roll moment due to sway and yaw motions:** By using the equivalent wave concept to determine the instantaneous leverarm $h$, the Froude-Krylov moment is separated into two parts. The part in $E_{nFK}$ is due to longitudinal waves whereas the roll moment due to transversal waves is included in the linear wave exciting roll moment $n_{e, sy}$. The latter moment is computed by means of a radiation-diffraction strip method in frequency domain. In the frequency domain strip method we
use an approach by Söding and Bertram [13] to account for the coupling between the suspended load and the ship. This approach considers the pendulum motion of the suspended load in one degree-of-freedom and neglects damping of the load. In this study we extend the approach to consider the pendulum motion in two degrees-of-freedom and to include damping of the load.

- Roll damping moment: Equation (19) for the roll damping moment includes a linear and a quadratic part. The linear roll damping coefficient $d_L$ considers the roll damping of the bare hull and is based on systematic model experiments by Blume [14]. The quadratic roll damping coefficient $d_Q$ considers the roll damping of appendages and the hull and is based on an approach of Gadd [15] and Blume [14]. A comprehensive summary of both approaches can be found in [9].

$$n_d = -d_L \dot{\phi} - d_Q \dot{\phi} |\dot{\phi}|.$$ (19)

**CASE STUDY**

The presented sea-keeping analysis method is applied to simulate an offshore lifting operation conducted in 2013.

**Heavy-lift Vessel**

For this operation the heavy-lift vessel shown in Fig. 3 was used. The 12,500 tdw ship was built in 2011 by the shipyard J.J. Sietas KG, Schiffswerft GmbH u. Co. as yard type 183. It is equipped with two cargo cranes of 1000 t SWL which can be jointly used for tandem lifts up to 2000 t. The vessel is equipped with a high capacity dynamic positioning DP2 system. The main particulars of the ship are shown in Table 1.

**TABLE 1. MAIN PARTICULARS OF THE HEAVY-LIFT VESSEL SIETAS TYPE 183.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length over all</td>
<td>160.50 m</td>
</tr>
<tr>
<td>Breadth moulded</td>
<td>27.50 m</td>
</tr>
<tr>
<td>Draught at summer freeboard</td>
<td>9.01 m</td>
</tr>
<tr>
<td>Speed @ 85% MCR</td>
<td>18.00 kn</td>
</tr>
</tbody>
</table>

**Offshore Lifting Operation**

The purpose of the lifting operation was the installation of a structure at an offshore location. First, the structure was stored on the cargo deck of the ship and transported to the site. At the site location, the seafastenings were removed, the main hook and the tugger lines were attached and the structure was lifted up by the rear cargo crane. Subsequently, the structure was swung out to portside over the water surface and lowered down to the seabed. Here, the structure was detached and installed successfully. During the entire operation no mooring lines were deployed and the ship was solely positioned by its DP2 system. In this study we simulate the lifting operation for the time period when the structure was lifted from the cargo deck of the ship as shown in Fig. 4. In order to keep this first model straightforward, we consider the crane tip position $s_{ST}(t)$ and the suspended length $l(t)$ as constant during the simulation and use the constant values shown in Table 2. Compared to its crane capacity of

**TABLE 2. PARAMETERS OF THE LIFTING OPERATION AT THE TIME WHEN THE STRUCTURE WAS LIFTED FROM THE CARGO DECK OF THE SHIP.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crane tip position $x_{st}$, $y_{st}$, $z_{st}$</td>
<td>-46.0 m, -1.0 m, 50.1 m</td>
</tr>
<tr>
<td>Suspended length $l$</td>
<td>31.0 m</td>
</tr>
<tr>
<td>Mass of the suspended load $m$</td>
<td>85.0 t</td>
</tr>
<tr>
<td>Radii of gyration $r_{xx}$, $r_{yy}$, $r_{zz}$</td>
<td>12.36 m, 44.02 m, 44.34 m</td>
</tr>
<tr>
<td>Displacement of the ship $\Delta$</td>
<td>19478 t</td>
</tr>
<tr>
<td>Metacentric height GM</td>
<td>2.71 m</td>
</tr>
<tr>
<td>Center of gravity of the ship</td>
<td>x: 70.85 m fr. AP, y: -0.07 m fr. CL, z: 10.36 m fr. BL</td>
</tr>
</tbody>
</table>
2000 t the analyzed offshore lift has a relatively small suspended load weight of 85 t. Hence, the coupling effects and in particular the influence of the suspended load motion on the roll motion is small. Nevertheless, we focussed our research on this operation because measurements of the ship and crane motions as well as of the hook load have been recorded during the lift. Moreover, this operation was one of very few offshore operations where seaway was high enough to be practically measureable. These measurements shall be used in further studies for comparison to simulation results.

**VERIFICATION**

In this study we concentrate on the verification of the wave exciting moment from the frequency domain analysis. The validation of the method is planned by comparisons to full scale measurements and model test in future studies. For verification purpose, the response amplitude operators (RAO) from the frequency domain strip method are shown in Fig. 5. Here, the RAO is made non-dimensional with the wave slope \( k \) and it is valid for the lifting operation specified in the case study. In order to verify the system response, Fig. 5 shows the RAO of the roll motion of the ship with freely suspended load and without suspended load respectively. The RAO of the ship with the suspended load indicates two eigenfrequencies of the crane vessel at \( \omega = 0.37 \text{s}^{-1} \) and at \( \omega = 0.59 \text{s}^{-1} \). The RAO of the roll motion of the ship without suspended load shows only one eigenfrequency peak which can be estimated by

\[
\omega = \sqrt{\frac{g GM \Delta}{I_{xx}}}. \tag{20}
\]

By using the common approximation for the radius of the hydrodynamic roll moment of inertia of 0.05 \( r_{xx} \), very good agreement is achieved with the natural roll period of the ship without suspended load of 0.40 \text{s}^{-1} from the frequency domain strip method. However, the natural roll period of the ship with suspended load cannot be estimated by Eqn. (20). In the same manner the second resonance peak of the RAO is not at 0.56 \text{s}^{-1} which is the natural frequency of pendulum given by

\[
\omega = \sqrt{\frac{g}{T}}. \tag{21}
\]

As it was already found by Grim [4], the eigenfrequencies of the coupled system are shifted compared to the eigenfrequencies of
the uncoupled systems. For symmetric ships and $x_\text{st} = y_\text{st} = 0$ the eigenfrequencies $\omega_{1,2}$ of the coupled system are given by

$$0 = g^2 \left( \frac{GM \Delta}{m} - l \right) + 2g \left( l^2 - l z_\text{st} - \frac{GM \Delta}{2m} - l_{xx} \right) \omega_{1,2}^2 - l \left( l^2 - 2l z_\text{st} + z_\text{st}^2 - \frac{l_{xx}}{m} \right) \omega_{1,2}^4. \tag{22}$$

Equation (22) can be verified by the limit value for $I_{xx} \to \infty$ which is the pendulum eigenfrequency given by Eqn. (21) and the limit value for $l \to 0$ which is the roll eigenfrequency of the ship with a fixed load given by Eqn. (20). As Fig. (5) shows, the agreement of the strip theory results and the eigenfrequency predictions for $\omega_{1,2}$ by Eqn. (22) is very good.

RESULTS, APPLICATIONS AND DISCUSSION

The major question during the planning phase of an offshore heavy-lift operation is whether the operation can be conducted safely at a given sea state. One possibility to give a reliable answer to this question is a sea-keeping capability assessment by means of sea-keeping simulations. For this purpose, quantitative safety criteria can be defined and checked subsequently. Here, we define two generic safety criteria for the motion of the load. The first safety criterion is fulfilled if the root mean square deviation of the relative suspended load motion does not exceed 0.2 m/s. Relative suspended load motion means in this context the motion of the suspended load relative to the ship. Hence, this criterion is appropriate when the load is lifted from the cargo deck of the ship. The second safety criterion assesses the absolute motion of the load with respect to the earth fixed frame. It is fulfilled if the root mean square deviation of the absolute suspended load motion does not exceed 0.2 m/s. An assessment of the absolute load motion is for instance of interest if the load need to be placed at a fixed position. An example of such an operation is the placing of a transition piece on a pre-installed monopile.

For both safety criteria, capability diagrams are computed and shown in Fig. 6 and Fig. 7. In these capability diagrams, the results are presented in polar coordinates. The main encounter angle of the short crested waves is plotted over the angular coordinate. An encounter angle of zero degrees means that waves are coming from abaat. An encounter angle of ninety degrees means beam waves from starboard. In the capability diagrams, the significant wave height $H_s$ of the short crested waves is plotted over the radial coordinate. The graphs plotted in the capability diagrams are found in the following way. For a seaway with given significant wave period $T_s$ and given encounter angle a simulation with a given significant wave height is performed. After each simulation the underlying criterion is checked. To achieve statistical certainty the simulations are repeated five times with different discretizations of the irregular seaway. This procedure is repeated for a set of different significant wave heights until the wave height is found where the criterion is just fulfilled. When the so-called limiting wave height is found, the procedure is repeated with the subsequent encounter angle. For all simulations required to generate the diagram, the mean forward speed of the ship is zero. The graph through all limiting wave heights of one significant wave period $T_s$ is then plotted in the capability diagram and labeled with the significant wave period which corresponds to the significant wave height. Hence, the diagrams
can be used to assess the actual existing sea state for a given encounter angle. If the actual significant wave height is inside the graph, the safety criterion is fulfilled and the operation is assessed as safe. If the actual significant wave height is outside of the graph, the safety criterion is violated.

Figure 6 shows that the relative load motion or rather the load velocity is generally smaller in shorter waves. In beam seas the relative load motion is generally larger than in head or stern seas. Furthermore, a comparison with Fig. 7 clearly demonstrates that the absolute load motion is in general larger. Therefore, from this perspective the placing of the load on a fixed position is more critical than the lifting of the load from the cargo deck of the ship. Moreover, Figure 7 shows that the velocities of the absolute load motion do not differ very much with respect to the significant wave period.

Even with advanced numerical tools to increase the convergence of the search for the limiting wave height, both capability plots in Fig. 6 and Fig. 7 require more than one year of simulated time. The required computation time is less than 15 minutes on a standard CPU. This fact illustrates the clear need for fast sea-keeping methods.

**Dynamic Amplification Factor Estimation**

Another application of the presented sea-keeping method is the determination of dynamic loads within the crane and the crane ropes due to pendulum motions of the load. Usually the dynamic loads are not computed by first principle methods but rather estimated with dynamic amplification factors given by class rules or other literature. Often the maximum dynamic loads are of interest. The quantification of these maximum loads is in particular difficult because they are rare events. Like capsize frequency computations, the simulation of rare events requires very long simulation times. Hence, a fast sea-keeping method is in particular suitable for these type of computations. Here, the presented sea-keeping method is used to quantify the dynamic amplification factor $DAF$ of the crane foundation bending moment. It is defined as the ratio of norm of the maximum dynamic crane bending moment $n_{\text{dyn, max}}$ to the norm of the static crane bending moment in the upright floating position $n_{\text{stat, upright}}$:

$$DAF = \frac{\|n_{\text{dyn}}\|}{\|n_{\text{stat, upright}}\|}. \quad (23)$$

Here, $n_{\text{dyn, max}}$ is the maximum value of the crane bending moment which is observed during five simulations with different irregular seaways of 10,000 s each. The bending moment is defined at the foundation of the crane which is located at $(-34.45 \text{ m}, 10.18 \text{ m}, 3.44 \text{ m})^T$ in the ship fixed frame $\{S, x, y, z\}$.

The results of this analysis are shown in Fig. 8. The rather trivial result of ascending dynamic amplification factors with an increasing wave height is clearly observable. Furthermore, shorter waves result in higher dynamic loads compared to longer waves of the same height. This result is in line with the capability plot in Fig. 7. The capability plot shows that the absolute load velocities are quite similar for long and short waves. However in short waves, the entire multi-body system generally oscillates with higher frequencies than in long waves. Hence, the accelerations and internal forces are generally higher for larger frequencies provided that the velocities of all frequencies were of similar magnitude.

**CONCLUSION AND OUTLOOK**

A numerical tool to assess the coupled motions of a heavy-lift vessel with a suspended load was presented and applied within a case study. This tool is an enhancement of the well validated sea-keeping code $E4-ROLLS$ and uses the same lean modeling approach. Due to these simplifications, crane operations in irregular seas with one crane and one hook can be simulated very fast and computationally efficiently. However, multiple hook or multiple crane operations cannot be simulated and require more complex methods. The results from the case study show that the method can accurately determine resonance frequencies of the coupled motions of the heavy-lift vessel. The case study clearly showed that the natural roll frequency of the ship can be significantly shifted even when the suspended load is less than 0.5% of the displacement which is not reflected by actual class rules [16]. Moreover, sample results showed that the method is particularly suitable when very long or numerous simulations or statistical results are of interest.

The presented method was developed within the ongoing re-
search and development project HoOK (see above). Within the future phase of the project the following improvements of the simulation method are planned:

- Further validation of the method.
- Consideration of ballasting and dynamic positioning operations within the simulation.
- Simulation of joint heavy-lift operations by multiple cranes and multiple hooks.
- Consideration of the non-linearity of the tugger winch damping.

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