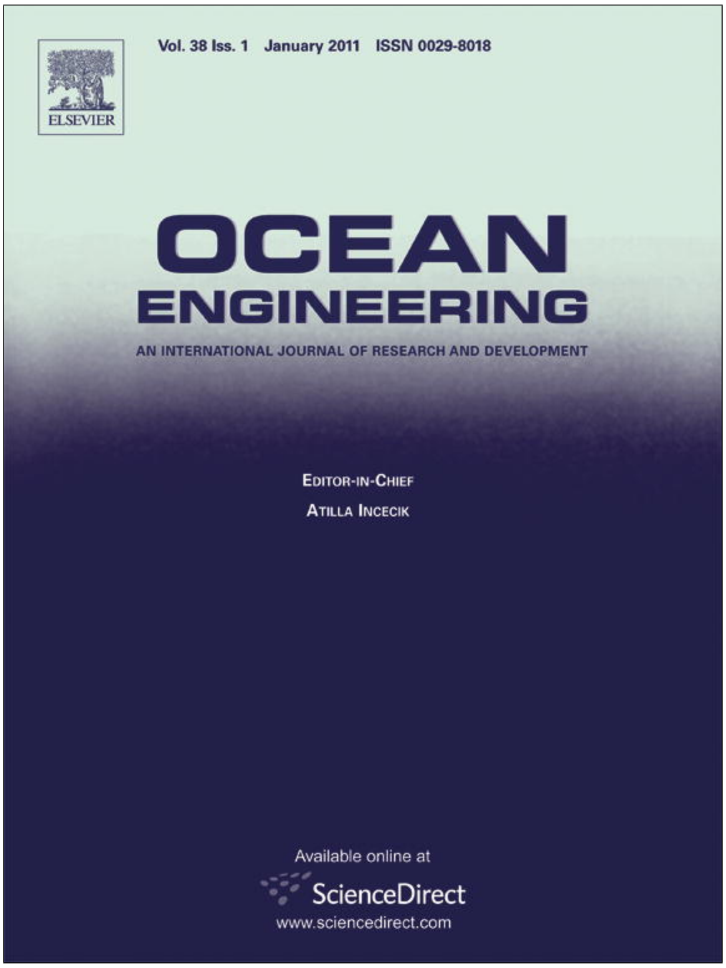
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Numerical and experimental studies of submerged towing of a subsea template

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ABSTRACT

During submerged towing of marine structural modules, the vessel motions will induce dynamic loads on the towing configuration. These dynamic loads are important when evaluating different submerged towing concepts and will have implications on structural design with respect to the relevant limit states. The focus of the present work is accordingly on the dynamic forces, which act in the main lifting wire that connects a subsea structure to the hang-off point above the moonpool of a vessel. Three alternative approaches for estimation of the tension are considered: A simple 1-degree-of-freedom system is integrated in the time domain by means of the Newmark-beta method and comparison is subsequently made with results obtained by application of detailed multibody time domain analysis by the Marintek software SIMO (Simulation Of Marine Operations). The results from the 1-degree-of-freedom system and the calculations by SIMO are also compared with measured response obtained from an experimental investigation. In general it is found that all the three methods show good agreement.

1. Introduction

Subsea templates are traditionally transported to the relevant site either on the deck of a crane vessel or a barge, depending on their size and shape. In both cases the template has to be lifted off from deck and lowered through the splash zone. The lift-off is a critical phase of the operation in which there is imminent danger of large dynamic loads and collision between the template and vessel deck due to relative motions. During immersion, significant wave impact forces (slamming) may also occur. Hence, these potential hazards will imply operational limits for the traditional methods of transportation for subsea equipment.

Another possible way of transporting a subsea template is to perform a submerged tow through the moonpool of an offshore service vessel. It is argued that such an operation will have a wider operational window than traditional methods since all offshore lifts are eliminated and will accordingly be more cost-efficient. A submerged towing operation of a heavy structure also enables maximum utilization of the crane capacity onboard the vessel, which ensures a safe installation process.

During a submerged towing operation, the dynamic behavior of the template and tow arrangement will depend upon the hydrodynamic loads, which act on the components of the system. The magnitude of these forces will affect the deflection angle of the towing wire in the moonpool and influence the probability that slack in the wire will occur. Accordingly, the forces will also set the operational limits both regarding permissible sea states and towing velocity. Hence, it is important that the computational models and procedures are validated before such an operation is carried out.

The purpose of this paper is to study the dynamics in the ship–template system for varying towing velocities, and for different sea states. Since the dynamic system is similar to a simple mass spring system, it is highly relevant to investigate the accuracy of utilizing a simple 1-Degree-of-freedom (1-DOF) approach compared with a more accurate multibody time domain integration approach. Such calculations are presently performed by the computer program SIMO. For the purpose of feasibility studies and preliminary design, there is still a need for simplified methods for estimation of the magnitude of the dynamic loads (Fig. 1).

This paper is organized as follows: Section 2 describes the simplified analytical model. A brief description of the theory and capabilities of the computer program SIMO is made in Section 3, and Section 4 describes the experiments, which are performed. The numerical results are compared to experimental values in

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Nomenclature

M	structural mass of towed object [kg]
A_{33}	added mass of towed object in heave [kg]
B_{33}	damping of towed object in heave [kg/s]
M'	$M+A_{33}$=total mass of towed object [kg]
m	mass per unit length of wire [kg/m]
L	length of wire [m]
E	modulus of elasticity [N/m^2]
A	nominal cross-sectional area of wire [m^2]
C_{Df}	wire longitudinal friction coefficient [dimensionless]
C_{Dz}	vertical drag coefficient of submerged object [dimensionless]
ρ_w	mass density of water [kg/m^3]
A_p	projected area of towed object [m^2]
k	wavenumber [m^{-1}]
$S(\omega)$	wave spectrum
$H(\omega)$	vessel transfer function
ε	phase difference [rad]

Section 5, and conclusions are drawn in Section 6. As an example case, a suspended subsea template, which has already been installed on the Norwegian continental shelf, is applied. The template is 29.3 m in length, 16.0 m in height and 22.1 m wide. The dry mass of the template in air is 300 tons and the added mass caused by the suction anchors is almost half an order of magnitude larger than the mass of the template itself. Therefore, the dynamic lifting wire tension may reach unexpectedly high values.

2. Simplified analytical model

As an approximation, it is assumed that the dynamic template–ship system can be represented by a 1-DOF system, see Fig. 2.

One of the most important parameters when performing a dynamic analysis is to determine the natural periods of the system. As a first approximation, DNV Recommended practice (DNV-RP-H103) states that when a deeply submerged object is exposed to vertical oscillations, the vertical motion of the object is governed by the motion of the wire fixed to the vessel. The following equation are stated in the recommended practice and serve as an engineering tool to determine the magnitude of dynamic forces occurring during offshore crane operations. However, these equations can also be used for describing a towed object subject to forced excitation loads.

Furthermore, the response amplitude (i.e. η) of the submerged template caused by a forced oscillation at the top of the wire with frequency ω and amplitude η_a can be evaluated by the following equation:

$$\frac{|\eta|}{\eta_a}=\left|\frac{kEA}{kEA\cos(kL)+(-\omega M'+i\omega\Sigma)\sin(kL)}\right| \quad (1)$$

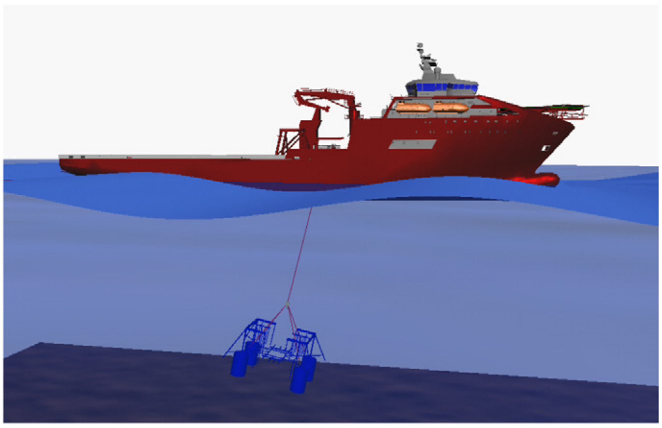

Fig. 1. Submerged towing with offshore service vessel.

where the linear damping coefficient for the motion is defined by

$$\Sigma=\frac{4}{3\pi}\rho C_{Dz}A_p\omega\eta_L \quad [\mathrm{kg/s}] \quad (2)$$

where C_{Dz} is the vertical drag coefficient for the suspended object and A_p is z-projected area. It should be noted that the linearized damping coefficients depend on the amplitude of the motion η_L and an iteration is needed to find the actual response of the suspended structure.

The amplitude of the dynamic axial force $F_D(z)$ in the wire at position Z is given by

$$\frac{F_d(z)}{\eta_a k_E}=\left|\frac{-(kL)^2k_E\sin[k(z+L)]+(kL)h(\omega)\cos[k(z+L)]}{(kL)k_E\cos(kL)+h(\omega)\sin(kL)}\right| \quad (3)$$

where

$$h(\omega)=-\omega^2M'+i\omega\Sigma$$
$$k_E=EA/L \quad (4)$$

where $h(\omega)$ is the complex frequency–response function and EA is the axial stiffness of the wire.

Since the dynamic force varies along the length of the wire (i.e. L), it is important to determine the dynamic force amplitude at all positions. If the dynamic force amplitude exceeds the static tension in the wire, a slack wire condition will occur. For the present example case, formulas (1) and (3) can be plotted with respect to angular oscillation frequency ω as shown in Fig. 3.

From the response/dynamic force curve large dynamic responses are observed close to the natural frequency of

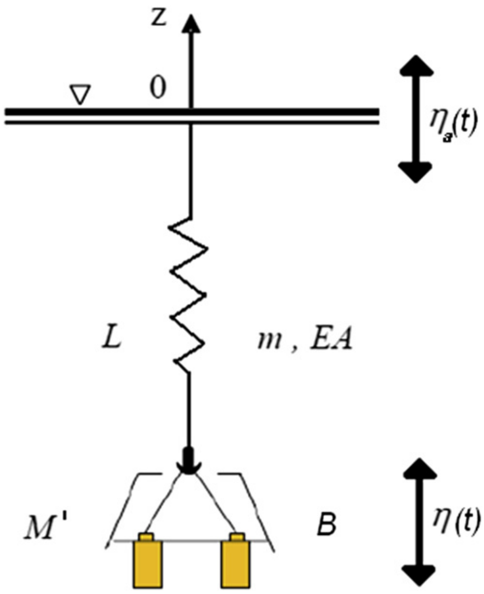

Fig. 2. Forced oscillation of a towed object suspended in a wire.

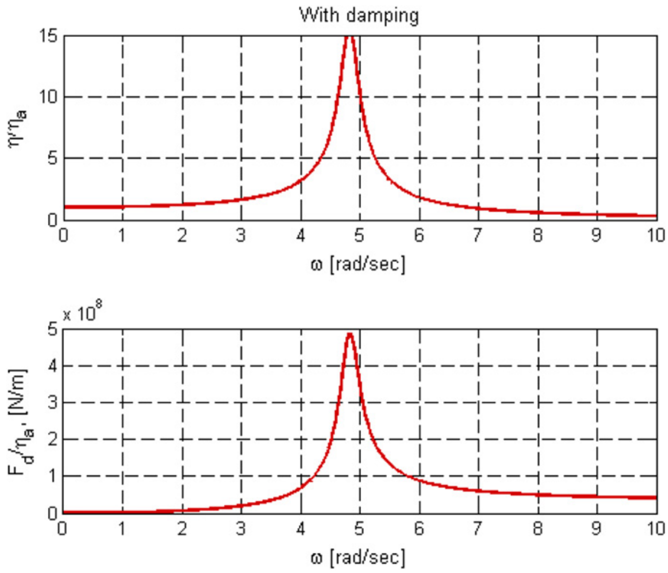

Fig. 3. Response/force amplitude of towed object due to forced excitations.

$\omega_0=4.8$ [rad/s]. On the contrary, for short oscillation frequencies, the combined system of the wire and template oscillates in a quasistatic manner like a rigid body where the template follows the motion of the top of the wire (i.e. the vessel motion).

An approximation of the vertical translation of the vessel due to irregular waves is obtained by combining the transfer-function in heave, $H(\omega)$, and a sum of harmonic oscillations randomly phase shifted ε relative to each another

$$\eta_a=\sum_n \eta_{3An}\cos(\omega_n t+\varepsilon) \\ =\sum_n \sqrt{|H(\omega_n)|^2 2S(\omega_n)\Delta\omega}\cos(\omega_n t+\varepsilon) \tag{5}$$

when the prescribed displacements at the top of the wire are known (η_a), the equation-of-motion of the 1-DOF system in Fig. 2 can in addition be solved in the time-domain by considering the following equation:

$$\begin{aligned} M'(\ddot{\eta}-\ddot{\eta}_a)+B(\dot{\eta}-\dot{\eta}_a)+k(\eta-\eta_a)&=-M'\ddot{\eta}_a-B\dot{\eta}_a \\ M'\ddot{\eta}_{rel}+B\dot{\eta}_{rel}+k\eta_{rel}&=-M'\ddot{\eta}_a-B\dot{\eta}_a=F(t) \end{aligned} \tag{6}$$

where $\eta_{rel}=\eta-\eta_a$ is the relative displacement, $M'=M+A_{33}$ is the total mass of the system and $F(t)$ is the exciting force. Eq. (6) can be solved in the time-domain using Newmark's-beta method with constant acceleration when all parameters in the equation have been determined. However, the coefficients for the template added mass and damping in Eq. (6) are often difficult to determine but can be approximated based on empirical data or CFD (Computational Fluid Dynamics) analysis. A simplified approach for estimating these values for a subsea template can be to solely consider the circular suction anchors, since it is known that these give rise to the major contribution to the added mass and damping. Coefficients for suction anchors can therefore be retrieved from DNV-RP-H103 and implemented in Eq. (6).

Further simplifications involve the assumption of a vertical wire. This is a simplification because the suspended load will have a horizontal offset angle α caused by hydrodynamic forces. This angle is of great importance since it is crucial that the towing wire does not get into contact with the moon pool edges. If this happens, the wire may be damaged. To derive the formulas for the horizontal offset angle α, small deflections are assumed and the wire is assumed to have infinite axial stiffness and zero bending stiffness.

The top of the lifting wire is located at the hang off point of the structure ($z=0$), and $\eta(z)$ is the horizontal offset measured from the vertical plane ($x=0$). The submerged weight of the template is denoted as W_0. At any vertical position z, the following equilibrium conditions have to be satisfied, according to Nielsen (2007):

$$T_W\cos(\alpha)=W_0+\rho_w gAz_b+mgs(z)-\int_0^{s(z)} q\sin\alpha ds \tag{7}$$

where T_w is the wire tension at the top of the wire and q is the drag force per unit length and the upper limit of integration, $s(z)$ is the length of the wire from the lower end to the cross-section considered. Also, at an arbitrary level z along the wire, equilibrium of horizontal forces can be found from geometrical considerations based on Fig. 4

$$T_w(z)\sin(\alpha)=F_{D0}+\int_0^L q\cos(\alpha)ds-\rho_0(z)A\sin(\alpha) \tag{8}$$

The first term on the right hand side of Eq. (8) is the horizontal drag force, which is acting on the template, and the second term is the integrated effect of the horizontal drag force acting on the wire. The last term on the right hand side is the horizontal component of the hydrostatic pressure.

Since the integrated effect of the drag force and hydrostatic pressure in (8) caused by the wire is small compared to the drag force acting on the template, it can be neglected. Moreover, asumming small angles in (8), the simplifications of $\cos(\alpha)\approx 1$ and $\sin(\alpha)\approx\alpha$ can be made. This yields

$$(T_w(z)+p_0(z)A)\alpha=F_{Do}+\int_0^{s(z)} qds \tag{9}$$

Since the effective tension can be expressed as $T_E=T_W+p_0A$ where T_W is the wire tension and p_0A is the hydrostatic pressure at a considered wire segment, Eq. (9) can be solved for the horizontal offset angle α

$$\alpha=\sin^{-1}\left(\frac{F_{D0}}{T_E}\right) \tag{10}$$

Moreover, the effective tension T_E can be written as

$$T_E=\sqrt{W_0^2+F_{D0}^2} \tag{11}$$

Eq. (10) is therefore simplified to

$$\alpha=\sin^{-1}\left(\frac{F_{D0}}{\sqrt{W_0^2+F_{D0}^2}}\right) \tag{12}$$

In the equations presented here, only the horizontal motion of the ship is considered. During the towing operation however, the ship will also be subjected to vertical forces, F_3, caused by heave and hence the effective tension in the wire will change as a function of time. If heave motion is taken into account, the horizontal offset can according to Nielsen (2007, p. 150) be

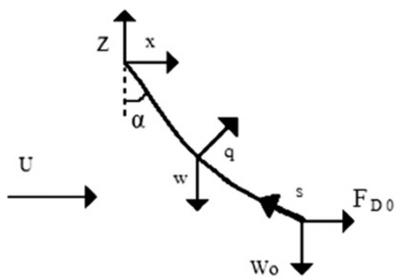

Fig. 4. Force equilibrium in wire Nielsen (2007).

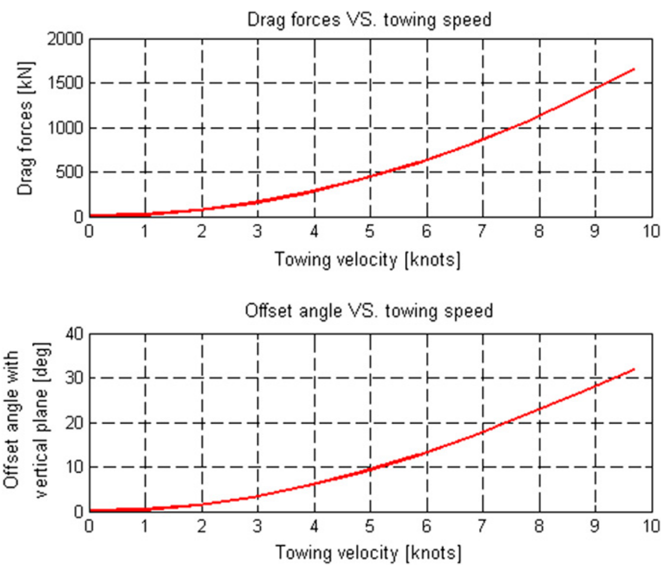

Fig. 5. Horizontal offset angle and induced drag forces.

written as

$$\alpha = \sin^{-1}\left(\frac{F_{D0}(v)}{F_{D0}(v)^2 + F_3^2}\right) \tag{13}$$

Further, by assuming a horizontal current with velocity v acting in the positive x-direction, and knowing that the drag force F_{D0} in cross flow can be expressed as

$$F_{D0} = \frac{1}{2}\rho v^2 C_D A \tag{14}$$

the drag force F_{DO} can be calculated by subdividing the template into 4 components with separate Morison forces. By inserting the value for the drag force into Eqs. (14) and (10), the offset angle α and the force in the lifting wire for the submerged towing system can be plotted versus towing speed in Fig. 5.

When the horizontal offset angle α is known, a conservative approach for estimating the maximum total force in the main lifting wire is to perform a linear superposition of the quasistatic tension in the towing wire caused by drag forces on the template, and the dynamic damping and inertia terms found by solving the single-degree-of-freedom system in Fig. 2 by means of stepwise time-integration.

3. Multi-degree-of-freedom time domain analysis software (SIMO)

The hydrodynamic force exerted on a slender object based on strip-theory is estimated by summing up sectional forces acting on each strip of the object. For slender structural members having cross-sectional dimensions considerably smaller than the wave length (normally when the wave length is at least 5 times the characteristic cross-sectional dimension), hydrodynamic loads may be calculated using Morison's formula. This means that the forces are sum of an inertia force proportional to the acceleration, and a drag force proportional to the square of the velocity.

Hydrodynamic loads on a slender structure can be subdivided into a normal force f_N, a tangential force f_T and a lift force f_L (Fig. 6). In the proceedings the tangential and normal force are omitted while the dominant normal force acting on each section of a slender structure is according to Det Norske Veritas (2009)

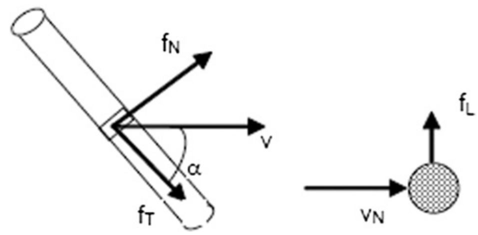

Fig. 6. Hydrodynamic forces on a slender structure (Det Norske Veritas, 2009).

given by

$$f_N = -\rho C_A A \ddot{x}_N + \rho(1+C_A)A\dot{v}_N + \frac{1}{2}\rho C_D D v_{rN}|v_{rN}| \tag{15}$$

where ρ is the mass density of water [kg/m^3], A is the cross-sectional area [m^2], C_A is the added mass coefficient [m^2], C_D is the drag coefficient in an oscillatory flow [dimensionless], D is the cross-sectional dimension [m], $\ddot{x}_N$ is the acceleration of element [m/s^2], v_{rN} is the relative velocity normal to the element [m/s] and $\dot{v}_N$ is the water particle accelerarion [m/s^2].

This expression is implemented into SIMO (Simulation of Marine Operations), which is a time domain simulation program developed by MARINTEK, for dynamic analysis of multibody systems. Long slender elements can be used to model the structural components of a subsea template (represented as a collection of separate subcomponents), where each component is associated with a distinct inertia, damping and added mass term. Rigid connections between all slender elements are assumed, which implies that all forces are calculated and directly transferred to the main body. However, SIMO is not capable of determining hydrodynamic properties of the modeled object and needs explicitly determined hydrodynamic properties of each slender element.

Since Morison's equation is a semi-empirical method, it relies on suitable experimental hydrodynamic coefficients available to determine hydrodynamic loads. Although a large amount of experimental and theoretical data has been published for simple body shapes, very little research has been published for more complex shapes. Furthermore, due to the complex way hydrodynamic coefficients vary (especially at low values of Keulegan–Carpenter numbers), it is often difficult to know whether published coefficients specific for one structure can be applied to another. Therefore using slender elements to represent a structures' hydrodynamic properties is an engineering simplification. However, individually defining coefficients for added mass and quadratic damping for each slender element from empirical data can be an effective approach to represent a dynamic system (Fig. 7).

The offshore vessel is represented by force transfer functions in all 6 degrees-of-freedom, which enables a coupled dynamic analysis when the template is connected to the hang off point above the vessel moon pool. The dynamic system can mathematically be represented by the dynamic equation of equilibrium. According to the SIMO theory manual (MARINTEK, 2001), this equation can be written as

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{D_1}\dot{\mathbf{x}} + \mathbf{D_2}\mathbf{f}(\dot{\mathbf{x}}) + \mathbf{K}(\mathbf{x})\mathbf{x} = \mathbf{q}(\mathbf{t},\mathbf{x},\dot{\mathbf{x}}), \tag{16}$$

where $\mathbf{M}$ is the frequency-dependent mass matrix, $\mathbf{m}$ is the body mass matrix, $\mathbf{A}$ is the frequency-dependent added-mass, $\mathbf{C}$ is the frequency-dependent potential damping matrix, $\mathbf{D}_1$ is linear damping matrix, $\mathbf{D}_2$ is quadratic damping matrix, $\mathbf{f}$ is vector function where $f_i = \dot{x}_i|\dot{x}_i|$, $\mathbf{K}$ is hydrostatic stiffness matrix, $\mathbf{x}$ is positive vector and $\mathbf{q}$ is the excitation force vector.

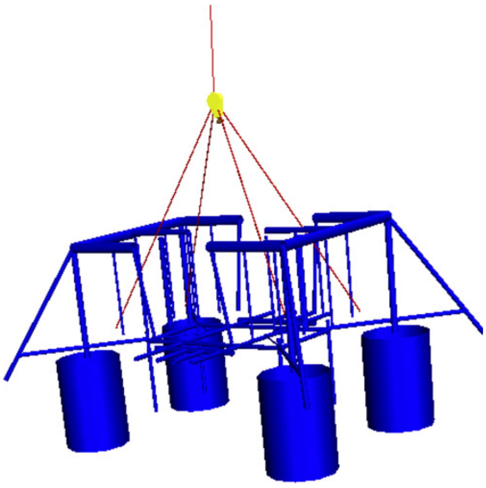

Fig. 7. Subsea template composed of slender elements.

This equation represents a system in 6-degrees-of-freedom, where the frequency dependent mass matrix contains the body mass matrix and frequency dependent added-mass matrix.

The excitation force vector on the right-hand side of Eq. (16) is given by

$$\mathbf{q}(t,x,\dot{x}) = \mathbf{q}_{WI} + \mathbf{q}_{WA}^{(1)} + \mathbf{q}_{WA}^{(2)} + \mathbf{q}_{CU} + \mathbf{q}_{ext}, \tag{17}$$

where, $\mathbf{q}_{WI}$ is the wind drag force, $\mathbf{q}_{WA}^{(1)}$ is 1 order wave excitation force, $\mathbf{q}_{WA}^{(2)}$ is 2 order wave excitation force, $\mathbf{q}_{CU}$ is the current drag force and $\mathbf{q}_{ext}$ is the any other force.

When the external forces, structural mass matrix ($\mathbf{M}$) and the stiffness matrix ($\mathbf{K}$) have been determined in Eq. (17), the equation of motion can be solved by convolution integrals in the time domain with retardation functions, or alternatively by separation of motions. Separation of motions implies that the motions are separated into a high-frequency and low-frequency parts. This allows that the high-frequency motions can be solved in the frequency domain (based on the assumption of linear response of the structure when subjected to the incident waves), and the low-frequency motions are solved in the time-domain. For a submerged towing operation both approaches provide comparable solutions of the dynamic equilibrium equation.

4. Experimental investigation

To verify the analytical models in the previous sections, an experimental investigation is performed for 12 different wave conditions. The forward speed of the towing carriage is used to represent the vessels' horizontal translation and a vertical oscillator is used to model the vertical translation of the vessel (i.e. heave motion) in regular waves in head sea as shown in Fig. 8. The following parameters are examined in the experiment, either by measurements, observations or both:

1. Towing force in the main towing wire
2. Motions of the template
3. Horizontal offset angle α

The experimental facility used in this model test is the Marine Cybernetics Laboratory at NTNU. Dynamic tension forces in the lifting wire are generated by an oscillator and they are measured by a force ring. Since the template is relatively close to the bottom of the test-basin bottom, there will be bias errors caused by bottom proximity effects.

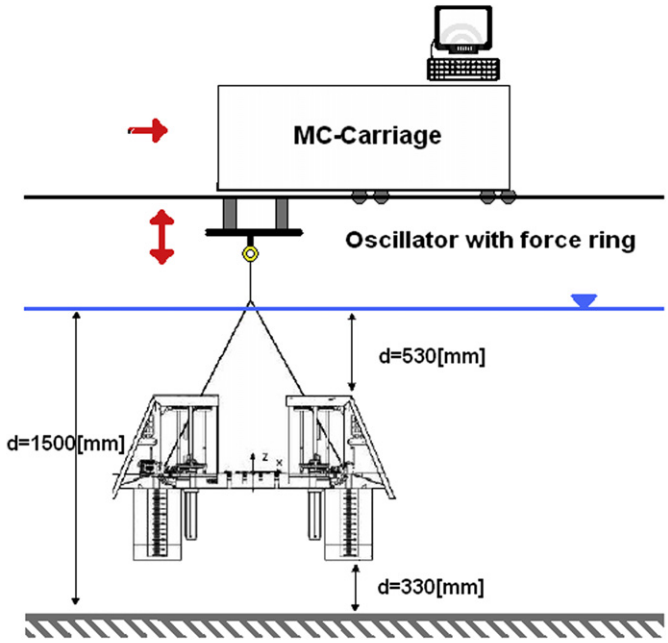

Fig. 8. Experimental setup.

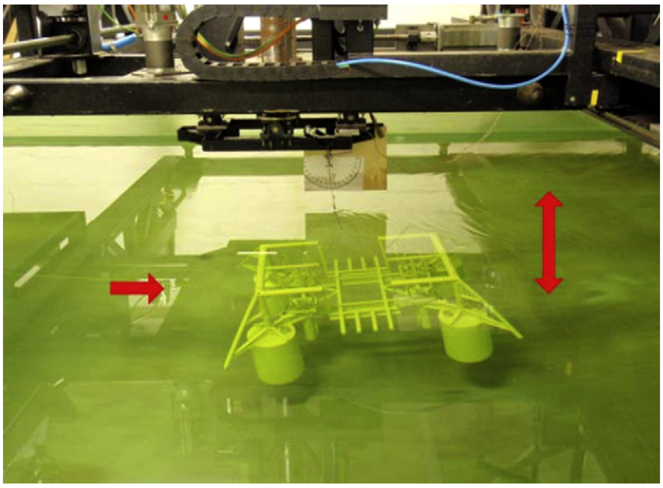

Fig. 9. Towing experiment in the longitudinal direction.

Froude scaling is used to transfer the measured values to full-scale values (scaling ratio $\lambda = 25$). This implies that the Reynolds numbers for the submerged towing operation in full-scale and model-scale will differ. These differences in Reynolds numbers mean that the drag coefficients are not equal in full-scale and model-scale, thus resulting in the drag forces not scaling correctly according to λ. To achieve correct drag forces, the diameters of the template model should be adjusted, but this is presently not possible. The application of Froude scaling is therefore an additional source of bias errors.

In Fig. 9 a picture of the experimental model is shown. Since the template can either be towed in the longitudinal or transverse direction, it is up to the marine contractor to choose the towing configuration. Usually the configuration with the lowest drag forces is chosen.

Since the main contribution to the drag force is caused by the suction anchors, and the turbulent flow caused by the forward suction anchors reduces the drag force on the rear anchors, one could easily conclude that a tow in the transverse direction is the best configuration with respect to drag forces. However, it is

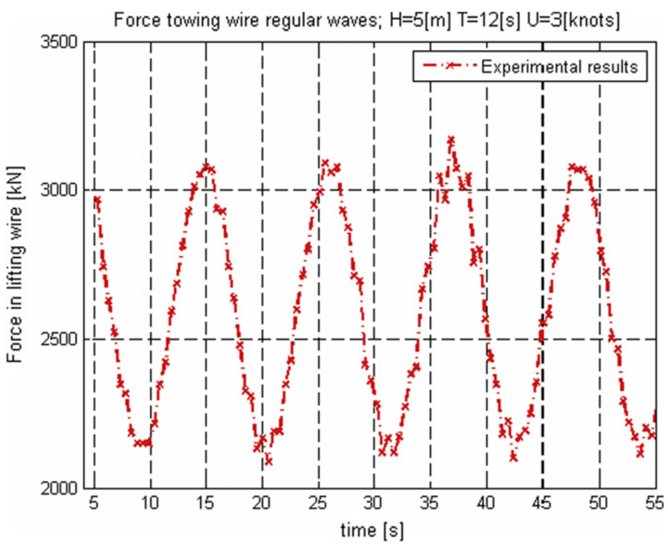

Fig. 10. Sample force in lifting wire from experimental results.

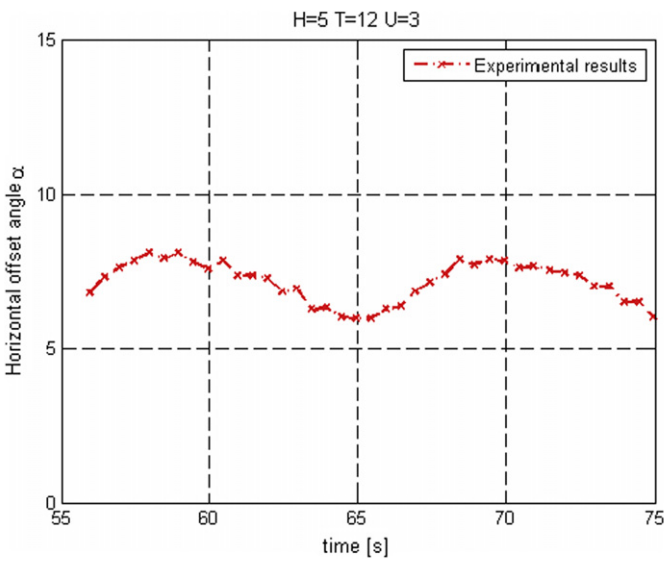

Fig. 11. Sample horizontal offset angle α from experimental results.

important to keep in mind that the projected area of the template incident to the current also increases, which may cause an increase of the drag forces. It is thus important to perform a detailed CFD analysis, and/or carry out model experiments to decide which towing configuration is to be preferred. In this paper, only towing in the longitudinal direction is considered.

A sample plot of the force in the towing wire for head sea, and regular waves with wave height $H=5$ [m], period $T=12$ [s] and towing speed $U=3$ [knots] is shown in Fig. 10.

The time dependent horizontal offset angles α are calculated from the measured forces in the vertical and horizontal direction of the force ring. An example plot for the experimental horizontal offset angle for regular waves with $H=5$ [m], $T=12$ [s] and $U=3$ [knots] is shown in Fig. 11.

From the figure it can be seen that the horizontal offset angle varies harmonically with time for a given sea state with regular waves. Especially the maximum horizontal offset angle is of interest, since this a limiting design criterion for the marine operation.

5. Comparative results

A sample comparison of the forces in the towing wire based on experimental results, analysis by SIMO and the analytical model for a sample regular wave condition with $H=5$ [m], $T=12$ [s] and $U=3$ [knots] can be seen in Fig. 12.

From the figure it can be observed that all three methods give comparable results for the amplitude of the force in the lifting wire. In particular, the SIMO analysis results and the experimental results for the maximum values agree well. However, the experimental results under-predict the minimum force in the lifting wire as compared to SIMO. This deviation is most likely caused by the influence of bottom proximity effects of the added mass in heave, A_{33}. These effects represent an issue due to the limited water depth of the test basin. Another possible reason for this deviation is that the SIMO model over-predicts the quasistatic drag forces and under-predicts the dynamic forces. This can be evaluated by towing the structure with no forced excitations and measuring the drag forces and comparing them with the SIMO results. For this case the hydrodynamic coefficients for the SIMO model were in accordance with the experimental results, thus implying that the deviation in dynamic forces is mainly caused by bottom proximity effects.

The analytical calculations also show conservative estimates of the dynamic forces compared to the SIMO analysis and experimental results. As previously mentioned, these results originate from a time integration of the equation of motion in heave where constant and conservative values for the added mass A_{33} and the damping B_{33} were applied. The static forces in the lifting wire caused by drag forces were directly added to the solution causing a vertical shift of the static equilibrium force. This can be observed in Fig. 12.

It can therefore be concluded that in this case, the analytical results overestimate the maximum force in the lifting wire (caused by conservative estimates of added mass and damping), and under-predict the minimum wire tension. These results are undesirable since the minimum wire tension is of importance when evaluating the possibility for slack in the wire, and analytical formulas should therefore only serve for the purpose of a preliminary study of the wire tension.

Also the horizontal offset angle α can be compared based on the calculations performed by means of SIMO and the experimental results. From Fig. 13 with $H=5$ [m], $T=12$ [s] and $U=3$ [knots], it can be seen that the experimental results yield the

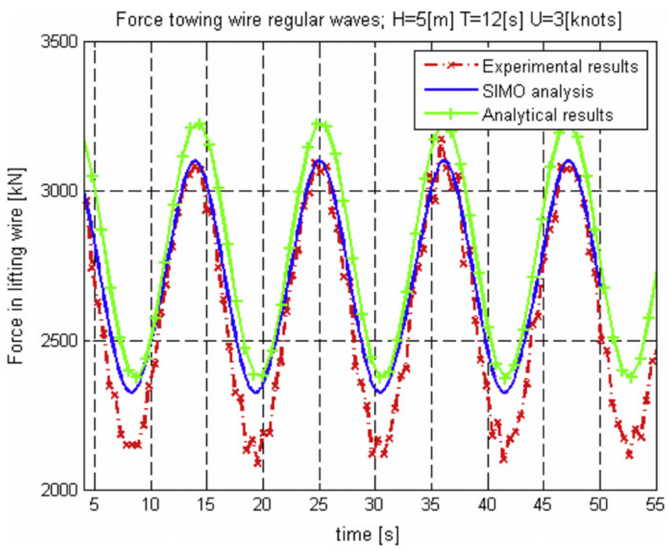

Fig. 12. Lifting wire force comparison.

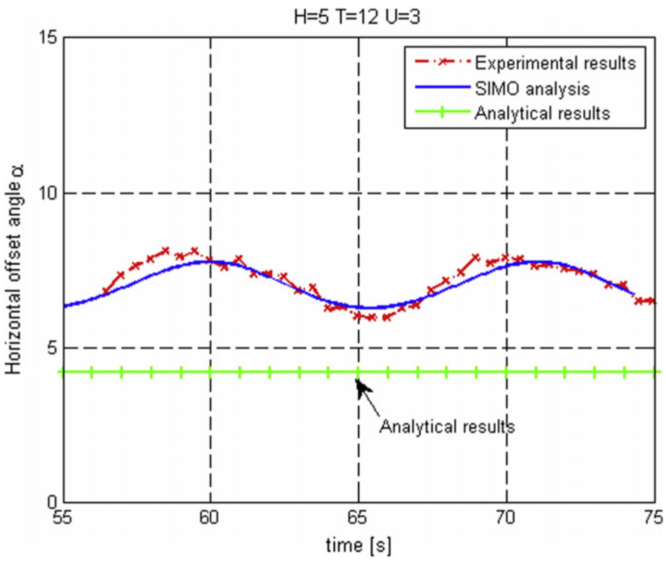

Fig. 13. Horizontal offset angle comparison.

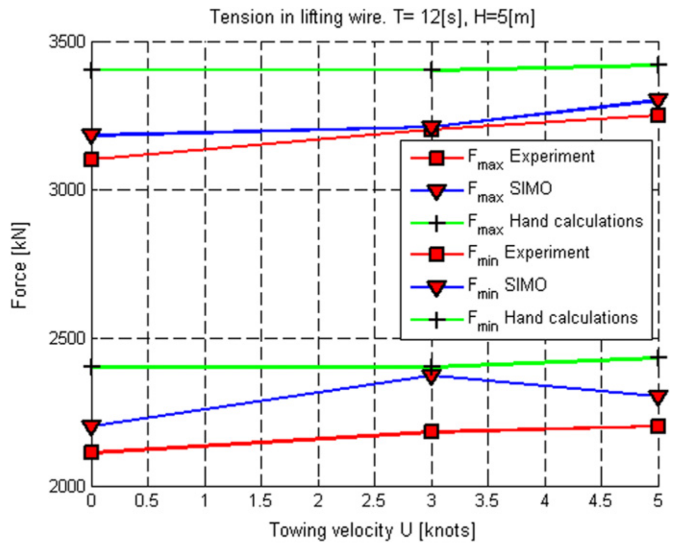

Fig. 15. Tension for regular waves, $T=12$ [s], $H=5$ [m].

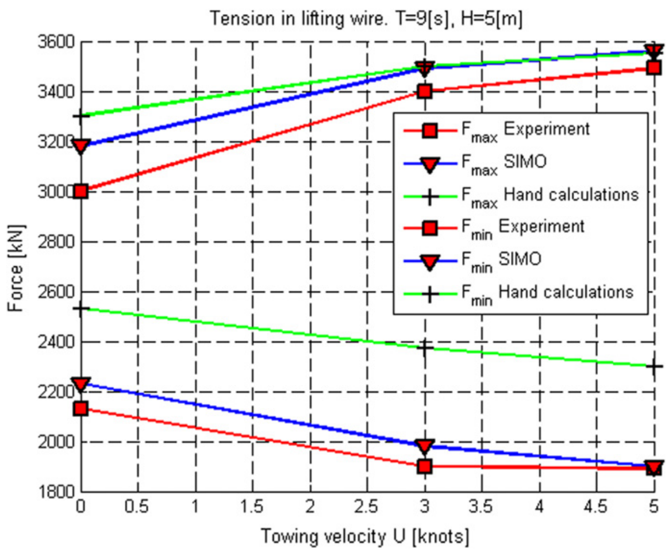

Fig. 14. Tension for regular waves, $T=9$ [s], $H=5$ [m].

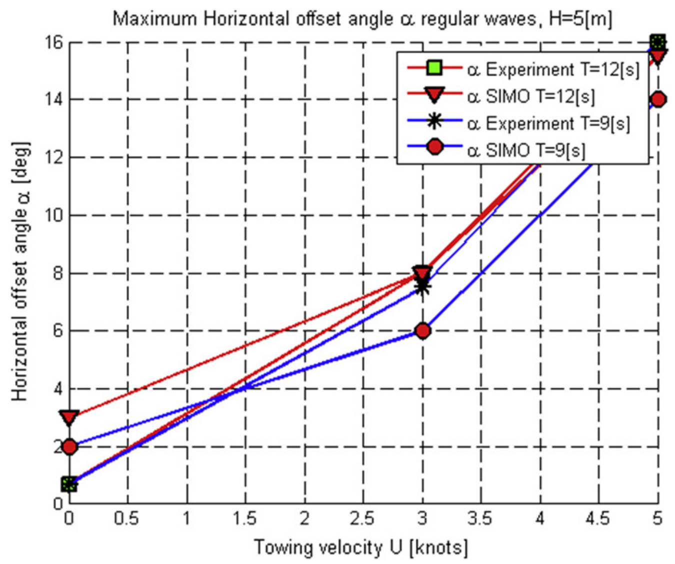

Fig. 16. Offset angle for regular waves $H=5$ [m].

smallest minimum horizontal angle values, but generally both methods show good agreements for the maximum horizontal offset angle.

The fact that there is a deviation of the minimum offset angle is again most likely due to bottom proximity effects. However, this is acceptable since the parameter of particular interest is the maximum horizontal offset angle α.

A summary of all the results from the analytical model, the SIMO analysis and the experimental tests are presented in Figs. 14–16. Only regular waves $H=5$ [m] are considered since these provide the maximum dynamic forces and offset angles. Furthermore, only the maximum horizontal offset angle is considered.

Incident regular waves with wave period $T=9$ [s] represent the worst condition with respect to vessel translation in the vertical direction, see Fig. 14. This is because the natural period T_0 for the vessel in heave occurs at $T=9$ [s]. However, the period of wave encounter T_e changes with increasing towing speed, and this effect is important to be aware of when evaluating limiting sea states for the operation. According to (Faltinsen, 1990) the period of wave encounter can be written as

$$T_e = \frac{2\pi}{\omega_0} + \frac{2\pi g}{\omega_0^2 U}\frac{1}{\cos(\beta)} \tag{18}$$

where β is the general heading angle between the vessel and the direction of wave propagation. The general heading angle is defined as $\beta=0^\circ$ for head sea. $\beta=90^\circ$ for beam sea and $\beta=180^\circ$ in following sea. U represents the towing speed of the vessel [m/s]. The maximum dynamic forces occur when the period of encounter is equal to the natural period of the vessel in heave, but it is important to remember that the total tension in the lifting wire also depends on the drag induced forces, which are acting on the suspended structure. This can be observed in Fig. 14. As the towing speed increases, the period of wave encounter decreases below the natural period of the vessel. Therefore depending on the transfer function of the vessel, dynamic excitation is reduced for periods lower than the natural period in heave. Drag forces on the other hand increase with towing speed, and this is the main reason for the slight increase in dynamic load amplitude in Fig. 14.

A comparison of maximum and minimum values for the lifting wire tension is made in Fig. 15. These results correspond to regular incident waves with wave period $T=12$ [s], and wave height $H=5$ [m].

The maximum and minimum tension obtained with SIMO are in accordance with experimental values for all test cases. The analytical model differ somewhat (up to 15%) from experimental values. This is the same pattern that could be observed in Fig. 12, meaning that the analytical model over predicts the force and the experimental values are influenced by bottom proximity effects.

The results for the maximum observed horizontal offset angle α are given in Fig. 16. Since the analytical model under-predicted the horizontal offset angle, these results are disregarded in order to simplify the plot.

As a general observation, there is a good agreement between experimental values and SIMO results for lifting wire tension and offset angle α in regular waves. This implies that the SIMO model provides good results and can be further used in more detailed hydrodynamic analysis including stochastic wave conditions.

6. Conclusions and recommendations for further work

The tension in the towing wire can be estimated by superposition of quasistatic drag forces and dynamic forces obtained by solving a single-degree-of freedom equilibrium equation. Also the horizontal offset angle with the vertical plane can be estimated by approximate expressions when the drag forces and excitation forces for the system are known. However, since the analytical formulas are based on approximations of the dynamic quantities, these results may deviate somewhat from the real values and should therefore only be used for feasibility studies at an early design stage.

More accurate results for horizontal offset and tension in the lifting wire are obtained by a more extensive model given as input to the multi-degree-of-freedom analysis program SIMO. This also enables analysis with varying environmental loads, and can be used to simulate a complete submerged towing operation. Time-traces of the dynamic loads can be extracted from SIMO and can be used in a more detailed fatigue and ultimate limit-state analysis for design of the hang off structure. From previous experience with submerged towing operations, especially fatigue is of crucial interest and should be further investigated when the dynamic forces have been successfully quantified.

To verify the accuracy of the numerical simulation model developed, an experimental investigation is a helpful resource. However model tests are often expensive and time demanding, and should therefore only be performed if results from analytical formulas and numerical models are uncertain and exceed theoretical values.

Finally it can be concluded that numerical time integration of multi-degree-of-freedom models can be applied for the purpose of accurate and efficient response estimation in relation to submerged towing operations. By utilizing such numerical tools, more cost efficient marine operations may accordingly be achieved.

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