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HOW DOES BARGE-MASTER COMPENSATE FOR THE BARGE MOTIONS: EXPERIMENTAL AND NUMERICAL STUDY

Frederick Jaouen
MARIN
Wageningen, the Netherlands

Joris van den Berg
MARIN
Wageningen, the Netherlands

Haite van der Schaaf
MARIN
Wageningen, the Netherlands

Eelko May
Barge Master
Schiedam, the Netherlands

Jim Koppenol
Barge Master
Schiedam, the Netherlands

ABSTRACT

To increase workability and safety in tough environmental conditions, and to create a more economical alternative for jack-up barges, Barge-Master has developed a wave compensating platform for marine and offshore installation barges. To minimize the motions of the crane positioned on top of it, the platform is driven by three hydraulic actuators that compensate for the roll, pitch and heave motions of the barge. To quantify the performance of the platform for different environmental conditions and crane configurations MARIN performed both wave basin model tests and time-domain simulations on the platform. In this paper, the experimental setup is first described and the model test results are presented. Then, the theoretical formulation of the time-domain aNySIM model is described and the numerical results are reported. It is shown that the model tests and time-domain results are in good agreement. The results indicate that the barge motions can be compensated by the platform for more than 90% in 1.2m high sea states. However, it is also shown that the instrumentation, data acquisition and controller system need to be fast and tuned to achieve this optimal compensation.

INTRODUCTION

The workability during offshore installation and building activities while using crane-barges is strongly influenced by both weather conditions and sea conditions. Mainly the so called “swell” (long waves) has a significant impact on the motions of the barge and crane resulting in situations whereby onboard personnel, equipment and installation parts are at risk.

Relatively small roll and pitch rotations of the barge on which a crane is placed result in large movements of the crane boom tip and therefore of the hook/load. These movements relative to the fixed world, where the actual installation work takes place, make working safely impossible and hence operations are ceased. The implications hereof for both project planning and costs are obvious.

To increase workability and safety in though environmental conditions and to create a more economical alternative for jack-up barges, Barge Master has developed a wave compensating platform for marine and offshore installation barges, named “Barge Master” ([1],[2]). The idea is to stabilize the crane without the use of jack-up legs. In a three dimensional world, six degrees of freedom (DoF) need to be compensated to keep the (crane) platform from moving. Three DoF’s are compensated by restraining two translations (Surge and Sway) and one rotation (Yaw) of the barge by using traditional anchors or a dynamic positioning system. The remaining three degrees of freedom, one translation (heave) and two rotations (roll and pitch), are compensated by the Barge Master, a platform driven by 3 hydraulic actuators. By measuring the heave, roll and pitch motions and controlling the three hydraulic cylinders supporting the platform to produce the counteractive motion, the platform stands still relative to the fixed world. A schematic representation of the Barge Master is shown on Figure 2, Figure 3 and Figure 4. The crane will be positioned on top of the hydraulic platform, and the hydraulic cylinders will be founded onto the barge deck.

The Barge Master is a versatile system that can be used as a crane platform or a supply platform. As a crane platform, the Barge Master increases the workability and safety of crane barges. It can be used in combination with a standard crawler crane and a standard flat top barge that can be rented locally (see Figure 1). As a supply platform, the Barge Master can be used to supply items to a fixed crane on for instance a jack-up.

The first Barge Master is designed to accommodate a 400 ton crawler crane or a supply load up to 700 ton. The cylinder to cylinder distance is 15m and the mass of the system itself is 250 ton. The peak power demand during maximum compensation is 1500 kW while the average power demand will not exceed 600 kW. The complete Barge Master system can be containerized and therefore easily mobilised. Compared to a jack-up barge, the Barge Master is an alternative that is cheaper and faster to mobilise.

Figure 1: application of the barge master for offshore installation operations

The target for the design is that a motion compensation performance of at least 90% is achieved in sea states with a significant wave height Hs=1.2m. This means that the platform motion amplitude is reduced with at least 90% compared to the vessel motion amplitude. This would suggest that compared to a jack-up platform, a standard crawler crane placed on a Barge Master platform on a standard barge is a good and probably cheaper alternative. MARIN was commissioned to verify this hypothesis and assess the performance of the Barge Master by carrying out both model tests and time domain simulations. Different environmental conditions and crane configurations were investigated. The controller system was also developed and optimized during the study, and the numerical model was validated based on the experimental results. The model tests results were used to define a working prototype of the barge master, the first one being delivered in July 2012. The results of these studies are presented in this paper.

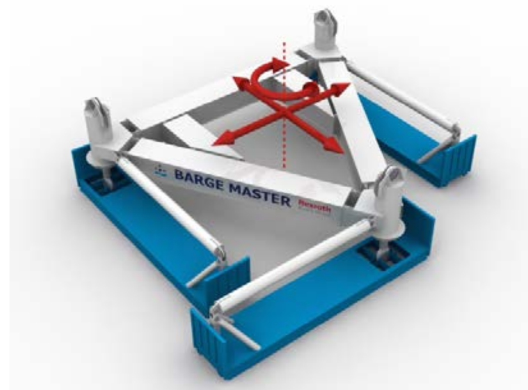

Figure 2: the mechanical constraints restrain the surge, sway and yaw degrees of freedom of Barge Master

Figure 3: the three hydraulic cylinders compensate for the heave, roll and pitch degrees of freedom

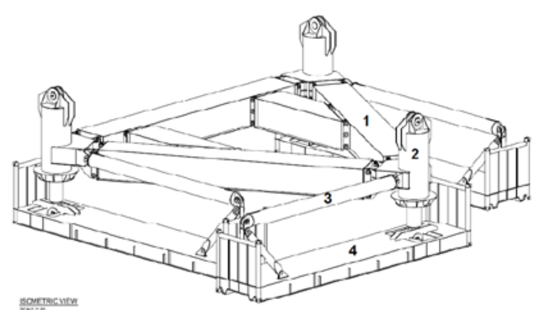

Figure 4: isometric view of Barge Master with 1) platform 2) cylinders 3) constraints 4) foundations

In this paper, the sign conventions and definitions will be first introduced. Then the experimental setup and results will be described. Finally, the time-domain numerical model and its tuning will be explained, and the numerical results will be compared to the experimental results.

DEFINITIONS AND SIGN CONVENTIONS

All values given in this paper are presented in full scale values unless stated otherwise. The metric system of units (SI) is used. Two right handed coordinate systems are used with the positive z-direction upwards: the basin-fixed coordinate system (BFCS) and the ship-fixed coordinate system (SFCS).

The ship-fixed coordinate system complies with the OCIMF conventions ([6]). The x coordinate is given with respect to Station10, the y-coordinate is given with respect to the centerline and the z-coordinate is given with respect to the keel of the ship. The origin of the *BCFS* coordinate system is located at the free surface, 18m above sea bed. At the starting time of a model test or numerical simulation, the x and y coordinates of the *BFCS* are the same as the x and y coordinates of the *SFCS*. As shown on Figure 5, the Barge Master is located at the stern of the barge, where the motions are the largest.

Two points are of importance because they are used to compare the heave motions of the barge and of the platform. First, the point P is fixed to the platform and is used to quantify the platform heave motion. The x and y coordinates of the point P are such that P is the centroid of the three cylinders. Second, the point BM is fixed to the barge and is used to quantify the barge heave motion. When the Barge Master is in mid-stroke position, the point BM is the projection of the point P on the barge deck. Figure 5 represents the side view of the barge, barge-master and the crane. The sign conventions and points P and BM are also indicated.

The cylinders 1 and 2 are located on portside and starboard respectively, at the same x-coordinate in the *SFCS*. The cylinder 3 is located on the centerline of the barge.

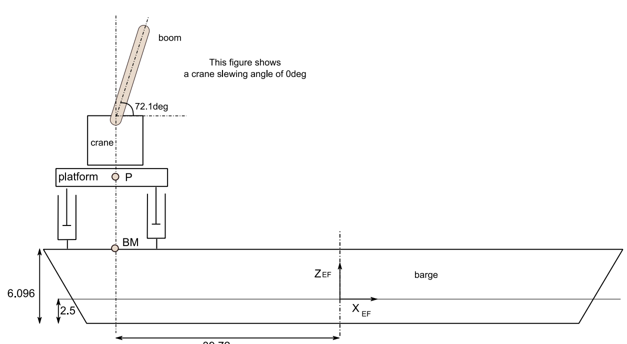

Figure 5: General arrangement of the barge and barge-master platform

EXPERIMENTAL SETUP

Model description

The models used for the test campaign are all new-built models. The model scale is 1:18 and is based on a combination of the environments, instrumentation and weight distribution requirements. The models used for model testing are described in the following paragraphs.

Barge: a new-built wooden scale model of the barge UR108 is used for the model tests. The barge is 91.44m long, 27.4m wide and 6.096m height. The draft of the barge is 2.5m and it has no trim when the barge-master, crane and load are installed. In this condition, the displacement of the barge is 5,590 ton.

Crane: an aluminium scale model of the crane Manitowoc 16000 is used for the test campaign. The crane has a heavy lift main boom with a length of 60m. A schematic load of 70 ton is fixed at the boom tip.

Barge Master: as shown on Figure 4, the barge-master conceptual design consists of the following parts: 1) a triangular platform that forms the support of the crane and connects the three cylinders 2) three vertical cylinders that actuate the platform in three degrees of freedom 3) three horizontal push-pull rods, also named mechanical constraints, that constrain the platform in three degrees of freedom 4) the foundations that form the interface between the barge and the platform. The platform surge and sway translations and yaw rotation are fixed to the barge with the mechanical constraints (3). If the platform compensates a barge movement in roll, pitch and/or heave direction, the orientation of the constraints will change. Due to the rotation of the constraints which have a limited length, the platform will have small parasitic motions.

In the reality, the Barge Master will contain 3 identical hydraulic cylinders which compensate for heave motion at their respective position. For the model tests, the hydraulic pistons are modeled by electrical cylinders that actively steer the vertical position of the crane deck. Figure 6 shows photographs of the Barge Master model manufactured and assembled at MARIN. When the cylinders are in mid-stroke position, the platform has no roll or pitch in the earth-fixed system and all cylinders are vertical. It should be noted that the real cylinders will be much shorter, but the rotation points will be the same as for the model shown on Figure 6. The geometry and kinematics of the barge master are exactly reproduced by the model.

The length of a fully retracted piston is 4.138m while the length of the fully extended piston is 6.063m. It follows that the piston stroke is 1.925m and the length of a piston at mid-stroke is 5.1m. To match the dynamic behavior of the electrical cylinders to the real hydraulic cylinders with finite oil column stiffness, the electric cylinders in the scale model are placed in series with disc springs. The friction and viscous damping are not modeled, so the own damping of the spindles is present only.

Mooring: a soft-mooring is used to keep the barge in position. It consists of four thin steel wire lines with linear springs included. The purpose of the soft-mooring is to keep the model in place and at the correct heading during tests in waves, without affecting the model's wave frequency motions. The surge, sway and yaw stiffnesses are respectively 148 kN/m, 110 kN/m and 3.3E5 kN.m/rad. This leads to surge, sway and yaw natural periods of 42s, 46s and 27s respectively.

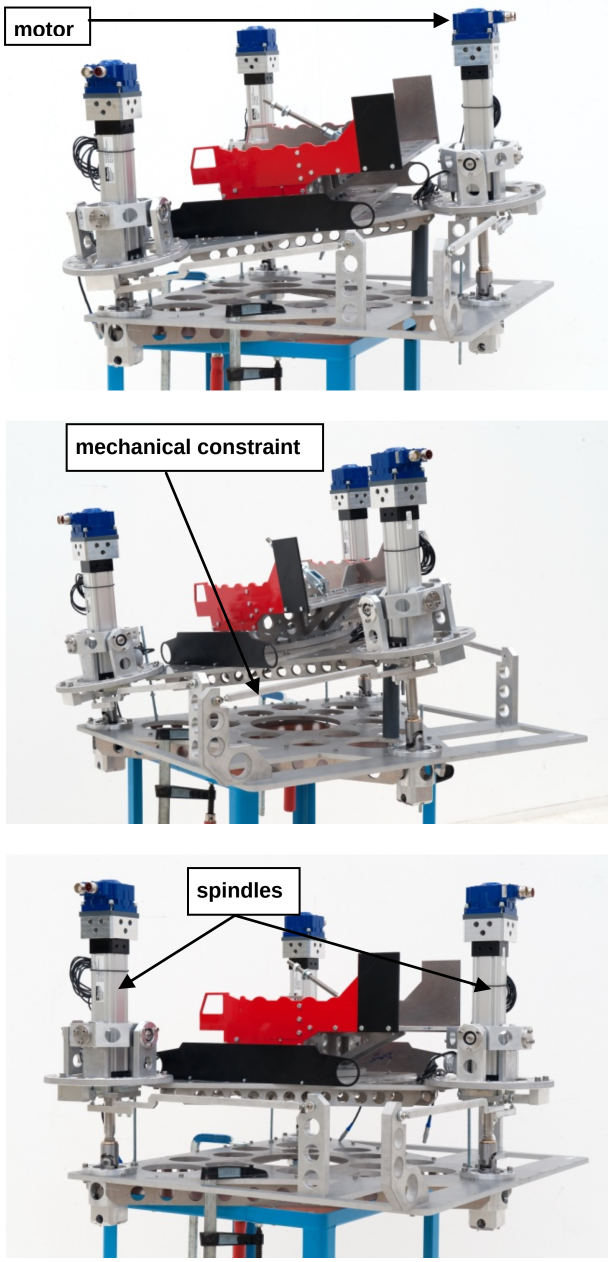

Figure 6: photographs of the Barge Master model

Instrumentation

All signals are measured at a 500Hz sampling frequency (model scale) during the test campaign. which corresponds to 118 Hz full-scale or 0.008s each two subsequent samples. A high sampling frequency is used, because minimizing the time delay in the system is critical to achieve a good motion compensation performance. This is possible both at model and full scale. On the real barge master, modern digital controllers are used which work with high sampling rate as well. The irregular wave tests are all one-hour long. The instrumentation that is used is shown on Figure 7 and is described in the following paragraphs.

Accelerometers: two sets of three 3DOF accelerometers are installed on the barge, at the (x,y) coordinate of the cylinders. One set is installed just under the cylinders, fixed to the barge-master foundation. The second set is installed on the bottom of the barge. These two sets are shown in green on Figure 7.

Krypton measurement system: an optical system measures the 6DOF of the barge and platform using triangular target frames with 3 optical markers. The measuring update rate is 350 Hz and the resolution and accuracy better than 1 mm. These measurements have a high bandwidth and are available at very high update rates. The measurements are performed with respect to the *BFCS*. They are free of drift. The two krypton targets are shown in yellow triangles on Figure 7.

Linear motor: Three linear electromotors are used to model the hydraulic cylinders. Each linear motor consists of a high accuracy spindle in combination with a brushless servomotor, both parts can be seen on Figure 6. This servomotor is driven by a PI controlled velocity mode drive system which accepts new velocity set points at a rate of 200Hz. The relative motor position and therefore the stroke of the cylinders are measured.
In order to avoid damage of the spindles, their stroke and velocity are limited. Above a certain excursion of the cylinders the speed requested from the actuator is scaled such that after 18 cm over-travel the speed has reduced to 0. Furthermore, the spindle speed is limited. Above this maximum speed limit, the speed set point for the actuators is clipped. The spindles are shown in yellow rectangles on Figure 7.

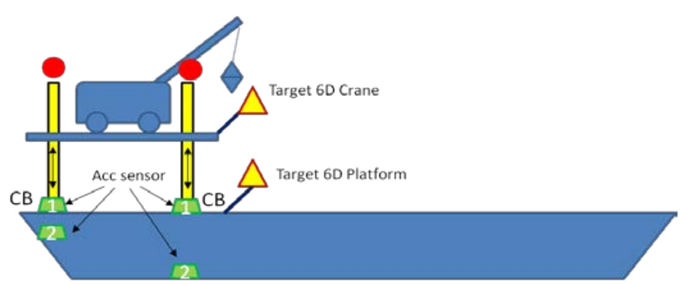

Figure 7: overview of the instrumentation

Environments and setup

The model tests are carried out in the Shallow Water Basin of MARIN, Wageningen. This basin offers the highest wave generation capacity in a water depth up to 1.0m. It is 220m long and 15.8m wide. It has a piston type wave maker at the short side of the basin. The opposite side of the basin is equipped with a beach to absorb the wave energy and minimize the wave reflection. The full scale water depth is 18m for all tests.

Three long-crested one-hour irregular waves are calibrated and tested. The JONSWAP formulation is used with an enhancement factor of 3.3. The significant wave height Hs is 1.2m and four peak periods of 6s, 8s, 10s and 12s are tested. Because the wave pistons in the shallow water basin can generate waves in the direction 180 deg only, the heading of the barge is varied to simulate 180deg and 210deg waves.

Two wave directions and three crane slewing angles are tested during the test campaign, i.e. a total of four setups. These setups are shown on Figure 8.

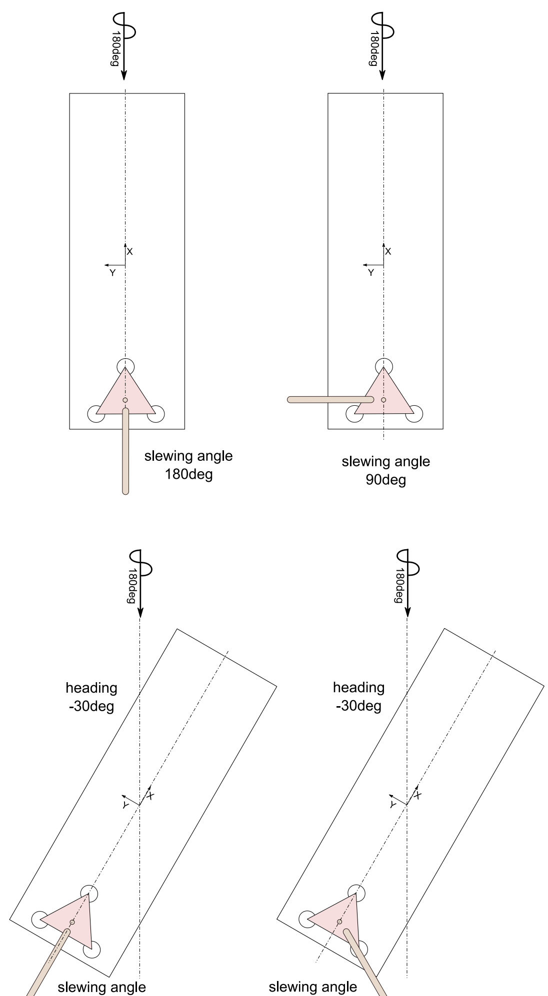

Figure 8: scenarios tested in basin

Controller

The objective of the Barge Master is to compensate the barge motions automatically, using a control system. The main task of the control system is to control the position of the crane platform to compensate the barge heave, roll and pitch motions. The vertical positions and velocities at the locations where cylinders connect to the barge, are used as set-points for a linear servo system consisting of a position feedback loop and a velocity feed-forward loop. A very simplified representation of the control system used per cylinder is shown on Figure 9. On this Figure, $z_{required}$ and $z_{measured}$ represent respectively the required (zero) and measured position of the cylinder base in the *BCFS*. The heave velocity of the barge at the cylinder base is noted v_{barge}. The velocity set-point prescribed to the cylinder is noted $v_{control}$. This section describes the control system that is used during model testing.

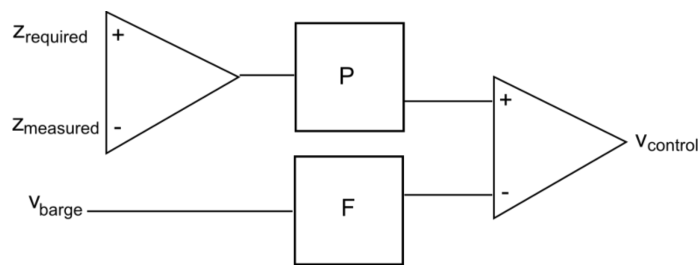

Figure 9: simplified representation of the control system

Background: the application of a servo system by means of a conventional feedback system requires an error to provide dynamic control actions from an actuator system. By increasing the proportional gain to very high values a fast tracking system can be implemented. This normally works for numerical simulations but in practice this results in instability due to latencies, noise and/or backlash in the real construction. Performance of tracking systems can be improved by using feed-forward terms for generation of control actions. These terms are determined independent from system response. In combination with a speed controlled actuator system a feed-forward of required speed gives a huge improvement of servo response without the necessity of high feedback controller gains. For the Barge Master application it was foreseen by Barge Master, Bosch-Rexroth and MARIN that the combination of position feedback and velocity feed-forward is the preferred way to go. This control strategy is applied during the project.

Controller strategy: to achieve an optimal compensation of motions the cylinder motions should track the barge heave without any delay due to e.g. phase shift or bandwidth limitations. This controller strategy is illustrated on Figure 10 and Figure 11, assuming a regular heave motion of the barge at 0.7Hz model scale, close to the heave natural frequency of the barge. On these Figures, the red and blue curves represent the barge and platform heave motions respectively. The pink curve represent the opposite motion of the cylinder. In Figure 10 a phase shift of 18deg is assumed and the residual error due to this tracking error is 30%. If we increase the bandwidth of the servo system and make the tracking faster to about 3 degrees of tracking error the residual error motion is still 6%. This is shown on Figure 11. For a period motion at 0.7Hz model scale, the phase shift of 3 degrees corresponds to a latency of only 10msec model scale.

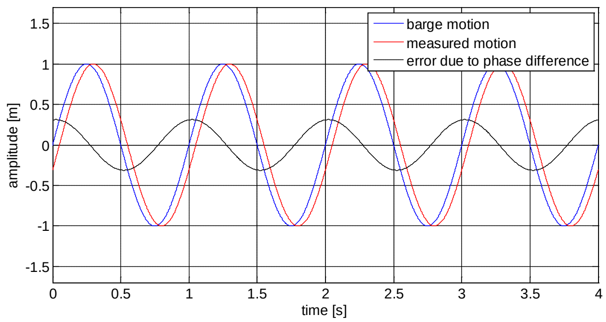

Figure 10: residual motion for a 18 degree tracking error

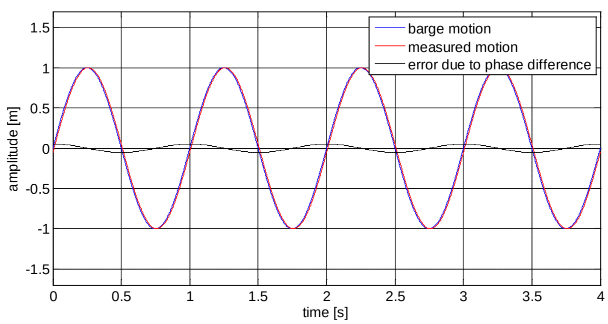

Figure 11: residual motion for a 3 degree tracking error

The theoretical possible compensation as a function of the phase shift (in degrees) between required motion and actual motion is shown in the picture below. The horizontal axis represents the phase shift in [deg] while the vertical axis represents the percentage of motion compensation, which is the ratio of the platform heave by the barge heave. For a zero phase shift, the compensation is perfect; in other words, the heave motion of the platform is null.

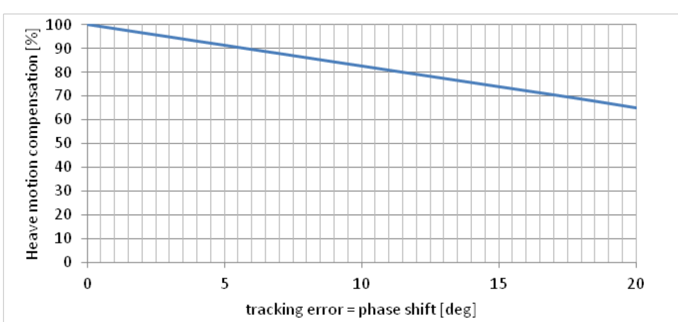

Figure 12: theoretical compensation vs. tracking error

Position and velocity measurements: both at full scale and model scale, the control strategy described above requires the availability of both heave positions and velocities at the position where the three cylinders connect to the barge.

To obtain the heave positions at the cylinder bases, the measured 6DOF barge motions are transformed to obtain the heave positions of the cylinders.

Determining the heave velocity is less straight forward. Differentiating the measured motions to get velocities while maintaining highest dynamics is not useful due to the resulting noise. This noise is a caused by quantization effects in time and value in the measured position signals. Another option is the integration of accelerations to velocities. From the noise point of view this is better but in practice acceleration signals are not offset free and this leads to drift in the integrated velocities.

Therefore, a dedicated circuit is used to estimate heave velocities based on the optical motion measurements with a feed-forward of actual accelerations measured at the three cylinder locations. Two sets of acceleration sensors are installed for this purpose.

Optimization: during the test campaign, the performances of the control system were improved from a hardware and software point of view. The objective of the optimization is to minimize the tracking error to increase the motion compensation as shown on Figure 12. A combination of filters were used in the control loop to provide some positive phase shift to compensate for delays in other parts.

MODEL TEST RESULTS

Influence of the barge-master on the barge

One of the objectives of the model tests is to make sure that the Barge Master did not influence the behavior of the barge in a negative way, i.e. excite the barge when it is supposed to compensate the barge motions. In addition to the model tests in irregular waves, the test matrix includes roll decay tests with an active and inactive Barge Master. It is found that the roll natural period drops by 1s when the platform is active. For the tested barge and sea states, it followed that the roll natural period was brought further away from the wave energy range, and this had therefore a positive effect on the roll motion. On the whole, this effect depends of the sea states and the vessel considered. Furthermore the damping of the barge remains similar. From these observations, it is concluded that the Barge Master does not affect negatively the behavior of the barge in the tested conditions.

Maximum stroke and velocity

Another objective of the model tests is estimate the strokes and velocities that are reached by the cylinders to compensate the barge motions. Table 1 summarizes the tests that have been carried out. It also shows the cylinder stroke and velocities that have been measured as a percentage of their maximum values. A 100% value means that the maximum value has been reached at least one time during the one-hour test.

It shows that the maximum speed is reached for cylinders 1 and 2 only, while the maximum speed for the cylinder 3 is always under the limit. This is because the heave motions at the cylinders 1 and 2 are larger due to the influence of roll component. For the cylinders 1 and 2, the maximum limit is reached at least one time per hour in the 8s and 10s waves, depending on the crane configurations. The cylinders 1 and 2 reached the limit for tests 305002 and 304004 only.

This Table also shows that the maximal stroke was exceeded during the tests for cylinders 1 and 2. Here again, the cylinder 3 has more margin. No stroke exceedance is measured in the 6s waves.

It should be noted that cylinder stroke and velocity exceedances are expected and are handled by the controller. The real question is to if the motion compensation remains acceptable despite these exceedances. The following paragraph will show that this is the case.

Test No.	Wave direction [deg]	Hs [m]	Tp [s]	Crane slewing angle [deg]	Controller Status	CYL1 Stroke [%]	CYL1 Speed [%]	CYL2 Stroke [%]	CYL2 Speed [%]	CYL3 Stroke [%]	CYL3 Speed [%]
Barge Heading = 0deg – Crane = 180deg											
303001	180	1.2	6.0	180	ON	47%	47%	46%	45%	31%	29%
304001	180	1.2	8.0	180	OFF	0%	0%	0%	0%	0%	0%
304002	180	1.2	8.0	180	ON	96%	74%	96%	75%	68%	48%
305001	180	1.2	10.0	180	ON	100%	75%	100%	75%	83%	52%
Barge Heading = 0deg – Crane = 90deg											
303002	180	1.2	6.0	90	ON	50%	51%	48%	47%	32%	31%
304003	180	1.2	8.0	90	ON	96%	73%	92%	70%	65%	48%
305002	180	1.2	10	90	ON	100%	100%	100%	100%	85%	61%
Barge Heading = -30deg – Crane = 180deg											
303003	210	1.2	6.0	180	ON	60%	65%	62%	67%	41%	42%
304004	210	1.2	8.0	180	ON	100%	100%	100%	94%	78%	62%
304007	210	1.2	8.0	180	OFF	0%	0%	0%	0%	0%	0%
Barge Heading = -30deg – Crane = 180deg – New network card											
305003	210	1.2	10.0	180	ON	100%	87%	100%	78%	89%	58%
Barge Heading = -30deg – Crane = 240deg – After controller optimization and new network card											
304011	210	1.2	8.0	240	ON	100%	79%	100%	75%	76%	54%
305004	210	1.2	10.0	240	ON	100%	86%	100%	76%	90%	56%
306001	210	1.2	12.0	240	ON	100%	78%	100%	73%	100%	56%

Table 1: extreme measured cylinder stroke and speed

Compensation performances

One of the objectives of the study is to estimate the motion compensation performance of the Barge Master system. This section summarizes the performance measured during testing. The performance of the Barge Master is quantified by comparing the barge motions to the platform motions, defined respectively at point BM and point P. The motion compensation achieved by the barge master is defined as the ratio of the platform motion standard deviation with the barge motion standard deviation.

It should be noted that the model tests results shown in Table 1 were performed in a chronological order. Towards the end of the test campaign, the controller was improved in order to reduce the tracking error, and therefore improve the compensation performance. First, the hardware was improved by changing a network card. Then the software was improved by adding filters. As shown in Table 1, the model tests performed with a wave direction 210deg and a crane slewing angle 240deg (304011, 305004 and 306001) were performed with the optimized controller.

The motion compensation results measured in these tests are illustrated in Figure 13. The horizontal axis of the bar chart indicates the wave condition. The vertical axis represents the compensation of the barge motion. This Figure shows that the platform compensation performance exceeds 90% in most of the situations. The test with a wave peak period of 12 seconds shows a slightly lower motion compensation performance due to more frequent stroke exceedance events. On the whole, very satisfying results are obtained and would certainly offer a significant increase of the workability during offshore operations.

Figure 13: bar chart of the compensation performance measured with the optimized controller

Model tests with a suspended load

At the end of the test campaign, model tests were carried out with a hanging load. Basically, the 70ton load was not fixed at the boom end anymore, but was hanging to a stiff line (Figure 14). These tests are not used for model verification and are therefore not presented in this paper. This effect of the hanging load is further studied using the numerical model.

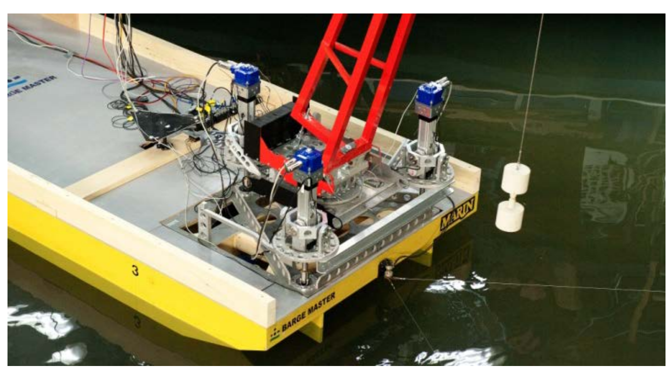

Figure 14: overview of the hanging load

NUMERICAL SETUP

Time-domain software: aNySIM

The program aNySIM is an in-house time-domain simulation software of MARIN. It can simulate the coupled motion behavior of multiple floating and non-floating bodies including effects such as mooring systems and hydrodynamic interactions ([4],[5]). The program integrates the equation of motion taking into account the own inertia, added inertia, wave loads, damping loads and restoring forces. It is able to simulate multi-bodies interactions. The time-domain code aNySIM integrates the linear equation of motions for the barge and barge-master platform, taking into account the own inertia, added inertia, wave loads, damping loads and hydrostatic restoring forces:

$$\sum_{j=1}^{6}(M_{kj}+m_{kj})\ddot{X}_j + \int_{-\infty}^{t} R_{kj}(t-\tau)\dot{X}(\tau)d\tau + C_{kj}X_j = F_k(t) \quad with\ k = 1,2\ldots 6$$

where:

- xj = motion in j-th mode
- Fk(t) = arbitrarily in time varying external force in the k-th mode of motion
- M = inertia matrix
- m = added inertia matrix (frequency independent)
- R = matrix of retardation functions
- C = matrix of hydrostatic restoring forces

The inertia matrix consists of the mass of the ship and the distribution of the masses with respect to the centre of gravity (radii of inertia). The right-hand side forcing function F includes the wave forces, mean wave drift forces, slowly varying wave drift forces, restoring forces of mooring lines. Additional external forces such as the forces applied by the cylinders on the barge and the platform are not a standard component of aNySIM and are added through a user interface, described later in this paper.

It should be noted that any viscous or flow effects cannot be calculated by the aNySIM program directly and need to be estimated (such as damping values) or neglected. In this study, viscous damping values are determined from the experimental data for roll, pitch and yaw degrees of freedom.

For the barge-master project, two bodies are modeled, referred to as *body1* and *body2*. This is shown on Figure 16. On one hand, *body1* consists of the barge. On the other hand, *body2* consists of the barge-master platform, the crane and the load attached to the boom, all together considered as one rigid body. The mechanical constraints shown on Figure 4 are represented by stiff compressive lines. The hydraulic cylinders and controller are modeled in the interface, described later in this paper.

Diffraction analysis software: DIFFRAC

The motion response, wave loads and wave drift forces of the ninety-meter long barge are calculated with MARIN's diffraction program DIFFRAC ([3]). DIFFRAC solves the linearised velocity potential problem, using a three-dimensional source distribution technique. The mean wetted part of the hull is approximated by a large number of panels. The applied panel distribution for the barge is depicted in the Figure 15. The distribution of source singularities on these panels forms the velocity potential describing the fluid flow around the hull. The pressure distribution on the hull is calculated from the velocity potential. The added mass and damping coefficients, as well as the wave forces are then determined from the pressure distribution and written in a hydrodynamic database, used as input for aNySIM.

All these calculations in DIFFRAC are carried out in the frequency domain. The added mass, damping coefficients and the wave load coefficients (exciting force) are used for calculating the motion RAO's for the six components of the barge. The program DRIFTP uses the results from DIFFRAC (added mass coefficients, damping coefficients, wave load coefficients, velocity potential) to calculate the second order wave drift forces. The mean wave drift forces (in regular waves) and the low frequency wave drift forces (in regular wave groups) are calculated through a direct pressure integration method. The results of the calculations are presented in the form of quadratic transfer functions (QTFs), of which the P-matrices contain the real part and the Q-matrices contain the imaginary part. DIFFRAC incorporates shallow water effects using the so-called Pinkster approximation.

The 3-D diffraction analysis calculates the potential damping only and cannot handle viscous flow effects in a direct manner. Therefore, empirically determined viscous roll damping effects has to be added separately in the input file of aNySIM. In this study empirical values of linear and quadratic damping coefficients based on the model decay tests are used.

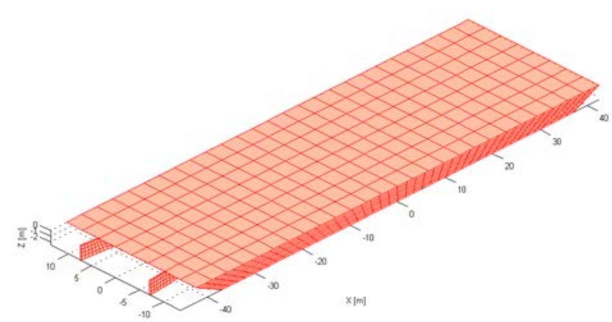

Figure 15: panel distribution of the barge 90m

Barge-Master user-interface

As described earlier, the time-domain software aNySIM integrates the equation of motion of each body simulated. The right-hand-side consists of the external forces applied on a given body, such as the mooring line forces, the mechanical constraints, or the wave forces. A user-defined interface is implemented to model both the controller and the hydraulic pistons. The design of the interface is as close as possible to the controller and actuator used in the model tests. A simplified representation of the two-body system simulated in aNySIM is represented on Figure 16.

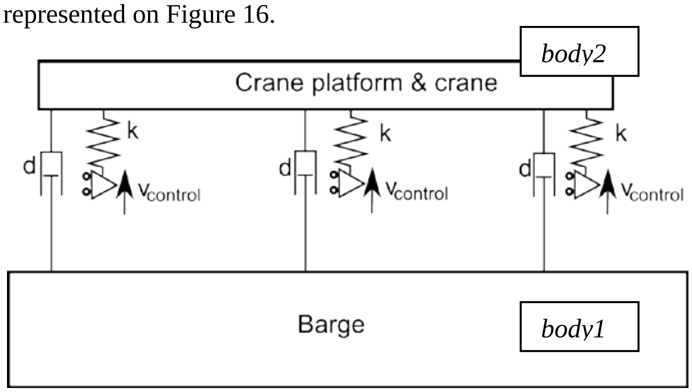

Figure 16: schematic representation of Barge Master

In the model tests, a control velocity is prescribed to the spindles based on the heave position error of the points where the cylinders connect to the barge, and based on the barge velocity. In the numerical model, forces can be applied on the bodies only, i.e. no position of velocity can be directly prescribed.

At each time-step, the Barge Master controller, similar to the one tested in the basin, determines the control velocity to be applied to a cylinder. The theoretical formulation is as follows:

$$V_{control} = P \cdot \left(Z_{required} - Z_{measured}\right) - F \cdot V_{barge}$$

From the control velocity $V_{control}$, an actual velocity V_{actual} is calculated to model a first order response of the cylinder, similar to what was observed in the basin. This models the actual response time of the actuators., whichcan be adjusted using the parameter τ.

$$\dot{V}_{actual} = (V_{control} - V_{actual}) \cdot \tau$$

From this first-order differential equation, the analytical equation of the cylinder position is determined, as a function of the initial cylinder position and speed (at previous time step), the control speed and the parameter τ. Based on the cylinder position and the actual distance between the platform and the barge, the cylinder force can be computed using the hydraulic stiffness.

Tuning of the numerical model

Before performing the final time-domain simulations, the aNySIM model is tuned to so that the properties of the numerical model are as close as possible to the properties of the experimental setup. The tuning consists in adjusting a selection of the tested setup properties based on the model test measurements. It mainly uses the results of both active and inactive decay tests. This section describes the parameters that are tuned.

Wave elevation: in order to enable a 1:1 comparison between model tests and aNySIM simulations, the wave elevations used in the simulations are taken directly from the wave calibration measurements in the basin. In this manner the wave signals in the simulations are identical to those generated during the model tests.

Mass matrix: prior to model testing, the weight distribution of the barge, platform and crane are measured. This includes mass, location of the centre of gravity and principle moments of inertia of these elements. Using the weight distribution of these elementary components, the mass matrix of the whole *body 1* and whole *body 2* are calculated and used for the numerical simulations.

Damping matrix: the diffraction calculations do not include any viscous contributions in the damping of the barge. Based on the analysis of the decay tests, linear pitch B_{θ} and yaw damping B_{ψ} are added to the barge. A quadratic roll damping B_{φ} is also added to the barge.

Stiffness matrix: the stiffness of the barge for the heave, roll and pitch is based on static approach and is correctly modeled in the diffraction analysis. In the tank, a soft-mooring system is applied so that the barge remains in position. Therefore, the linear surge, sway and yaw stiffnesses are adjusted in aNySIM.

Controller coefficients: in order to enable a 1:1 comparison between model tests and aNySIM simulations, the same proportional gain P and feed-forward controller coefficient F as used for model testing are applied for the simulations. In this manner the control velocities resulting from a position error and barge velocity in the simulations are similar to those generated during the model tests.

Response time of the actuators: the global response time of the actuators can be seen as the sum of two different terms: the reaction time (or communication time), and the actuator response time. First, the communication time is the time needed for the acquisition system to get the measured positions and velocities and is basically the time-step itself in the numerical model. Second, the response time of the actuator is the time needed for the actuator to achieve 63% of the requested input and it is modeled in the numerical model as a first-order response system with the parameter τ.

NUMERICAL RESULTS

The results of four one-hour time domain simulations are presented in this section. These simulations correspond to the tests 303003, 304004, 304001 and 305004, mentioned in Table 1. These tests are all performed for a 210deg wave, before and after the controller optimization. The results presented in this section focus on the motions of the barge and platform. The motion time traces and heave RAOs of the measurements and calculations are compared.

As an example, the time traces of the heave and pitch motions of the barge and platform are compared for test 304004 on Figure 17 and Figure 18. A reasonable agreement is found between the experimental and numerical results.

On Figure 19, the RAO of the barge and platform heave motions is also analyzed for the four time-domain simulations. This Figure compares for each case the heave phase and amplitude RAOs analyzed from the experiments and from the calculations. These RAOs represents the ratio of the platform heave at point P over the barge heave at point BM. The full line accounts for the measured heave RAO during experiment while the dotted line accounts for the computed heave RAO. The horizontal axis represents the cyclic frequency in [rad/s] while the vertical axis represents the phase or amplitude RAO [-].

Figure 19 shows that a good agreement is obtained between the experimental and numerical results of the heave amplitude RAOs. It also shows that the heave RAO is a linear function of the frequency. In other words, the lower the frequency, the better the compensation.

For tests 303003 and 304004, the phase RAO are very similar. Small differences can be observed for the tests 304001 and 305004. As shown in Table 1, these tests are performed after the controller optimization. As earlier mentioned, the controller for basin model tests was optimized using filters that affect the phasing. These filters were not included in the numerical model. It is therefore normal to find these differences.

On the whole, the numerical results agree reasonably well with the experimental results. This allows improving the controller further by means of time-domain simulations. A similar study for other ships can also be performed quicker and cheaper than by additional model testing.

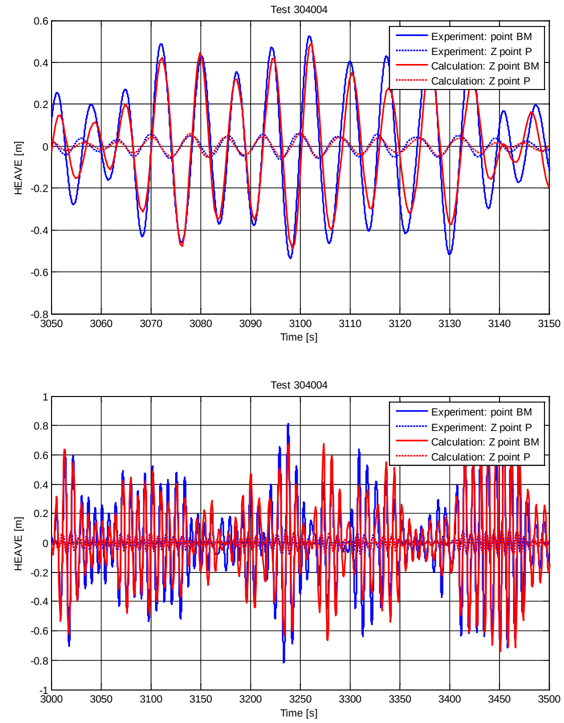

Figure 17: comparison of the heave time trace : experimental v.s. numerical results

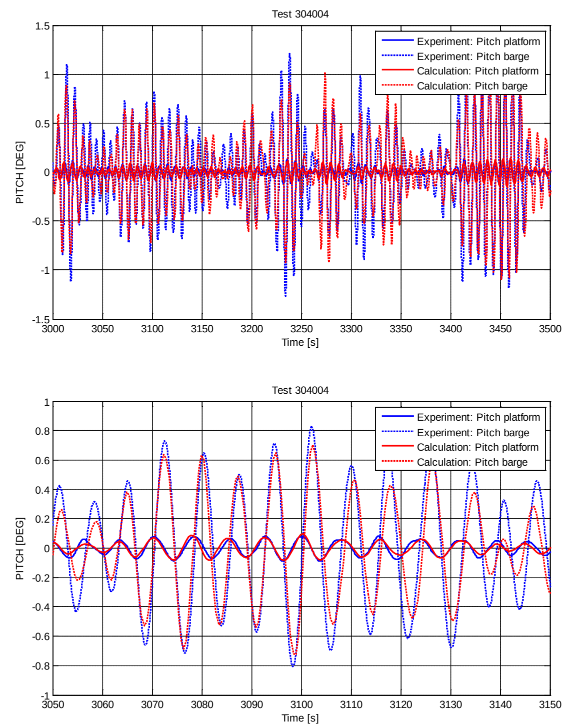

Figure 18: comparison of the pitch time traces: experimental v.s. numerical results

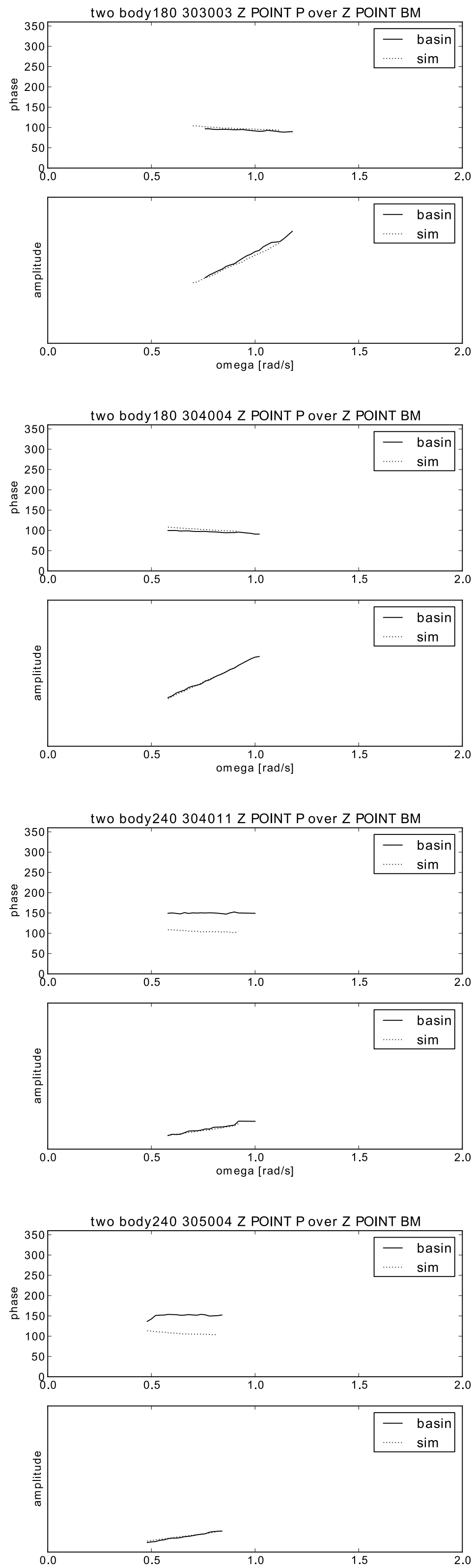

Figure 19: RAO between barge heave and platform heave motion

CONCLUSIONS AND RECOMMENDATIONS

In this paper, the experimental and numerical study results on the compensation platform Barge Master are presented and discussed. It is shown that a motion compensation performance exceeding 90% is achieved with the Barge Master in the tested irregular sea states. This shows that barge-master can be a good alternative for jack up barges. The performances of the barge master in other sea states will be investigated using the numerical model.

The model test results showed that the Barge Master did not influence the barge behavior in a negative way for the tested conditions. The natural periods and damping coefficients of the barge are measured for an active and inactive controller. It is found that the active Barge Master leads to a shorter roll natural period and a similar roll damping compared to the passive Barge Master.

Although the cylinder stroke and speed limits are reached for some wave conditions and crane configurations, the motion compensation performance of the Barge Master is very satisfying and meets the expectations from theory. However, the model tests also showed that the controller plays a crucial role and needs to be finely tuned to achieve optimal performances. In the model tests carried out with the optimized controller, the motions of the compensated platform are lower than 7% of the barge motions. This is valid for roll, pitch and heave. In other words, it is confirmed that the motion compensation performance achieved by the barge master can exceed 90% in the tested sea states.

Furthermore, an aNySIM based time-domain numerical model of the Barge Master is developed at MARIN and is validated based on the experimental results. It is shown that the measured and calculated barge and platform motion time traces agree reasonably well. Similarly, the ratio between the barge heave motion and the platform heave motion agree well. The motion compensation is also shown to be a linear function of the motion frequency, i.e. the lower the frequency the better the compensation.

Based on the experimental and numerical results, Barge Master decided to increase the maximum stroke of the hydraulic cylinders, so that the design target criteria are fulfilled. This should reduce the probability of stroke exceedance and improve the compensation performance further, especially in the long waves. The first full scale Barge Master motion compensation system is expected for July 2012.

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