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MODEL TEST INVESTIGATION OF THE INFLUENCE OF DAMPING ON THE VORTEX INDUCED MOTIONS OF DEEP DRAFT SEMI-SUBMERSIBLES USING A NOVEL ACTIVE DAMPING DEVICE

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ABSTRACT

Vortex Induced Motions (VIM) of semi-submersibles occur when the vortex shedding frequency is close to the natural frequency of the semi-submersible in the direction transverse to the current. Recent studies suggest that the magnitude of VIM predicted during model tests is higher than that observed in the field. Among others, the damping effect provided by the risers and mooring lines is regarded as one of the reasons for this difference. In this paper the setup and results are presented for model tests to investigate the influence of damping on VIM.

For these model tests an active damping system was developed, which introduces an actively controlled external force mimicking a damping force. This applied damping force is based on the floater sway motion and sway velocity. With this system the introduced damping level can easily be controlled and verified without changing the stiffness of the system. In this paper the advantages and disadvantages of this active damping system are presented.

The VIM tests were conducted for two semi-submersibles: a paired-column semi with eight columns and a four column semi. Reduced velocities ranged from Ur=3 to Ur=10 and different levels of additional linear damping were applied to the floaters in the direction transverse to the current direction. Damping was found to reduce the VIM motions significantly: reductions of more than 60% were observed in the nominal A/D response for 25% equivalent linear damping. This indicates that damping has a significant effect on the global VIM response and thus should be considered in the design phase of the risers and mooring lines of the semi-submersibles.

To improve the understanding of the driving mechanism of VIM and also to provide validation data for CFD analyses, forces were measured on each column of the four column semi. Column force measurements indicate that for the four column semi for 45 degrees heading, i.e. the heading with largest VIM responses, the upstream, the portside and the starboard side columns are exciting the VIM motions. For 22.5 degrees, the downstream, the portside and the starboard side columns excite the VIM motions. For all tested headings the pontoon always damps the VIM response.

INTRODUCTION

VIM predictions from model tests are considered in the design phase of semi-submersible. The magnitude of these VIM responses determines among others the fatigue life of mooring lines and risers. Recent research has laid down that these VIM predictions are conservative [1-3]. This implies that the design of risers and mooring lines is also overly conservative and thereby more expensive than needed.

One of the possible reasons why VIM model testing is conservative is the lack of the hydrodynamic modeling of mooring lines and risers. Instead of model scale mooring and risers, an equivalent soft mooring is applied to model the stiffness properties of the semi-submersible mooring and risers.

A more realistic model test setup including the hydrodynamic modeling of the mooring and risers shall lead to better predictions of the actual VIM responses in the field and eventually to less conservative design of risers and mooring lines. However, depending on the geometry and test velocities, Reynolds effects might play a role. Furthermore, modeling the mooring and risers increases the test setup complexity and model tests become more expensive and time consuming. Another option is to model the damping forces originating from the mooring lines and risers. This can be accomplished with minor adjustments to the standard VIM setup but requires input of damping levels.

Within the RPSEA project a novel setup was developed and model tests were conducted to assess the effect of damping on VIM in a controlled environment. The focus was not on the determination of damping levels of risers, mooring lines and other possible damping sources. Two semi-submersible models were tested, a paired-column semi with eight columns and a four column semi. For the latter model column forces were measured, to better understand the driving mechanisms of VIM and also to have additional validation data for CFD.

In this paper, the developed damping system and its advantages and disadvantages will be discussed. Dedicated tests to verify the correct working of the damping system are also treated. To see the effect of the damping on VIM the model test results are provided as well as the results of the column force measurements.

NOMENCLATURE

ζ	Damping ratio	[-]
a	Acceleration	[m/s]
A	Amplitude	[m]
b	Equivalent linear damping coefficient	[kNs/m]
D	Column diagonal	[m]
f	Frequency	[Hz]
F_y	Cross-flow load	[kN]
$F_{h,y}$	Cross-flow hydrodynamic load	[kN]
K	Structural stiffness	[kN/m]
m_a	Added mass	[kg],[t]
m_s	Mass semi-submersible	[kg],[t]
x	Semi-submersible in-flow direction displacement	[m]
y_a	Basin-fixed motion anchor points springs	[m]
y	Semi-submersible cross-flow direction displacement	[m]
$\dot{y}$	Semi-submersible cross-flow direction velocity	[m/s]
$\ddot{y}$	Semi-submersible cross-flow direction acceleration	[m/s2]
t	time	[s]
U	Tow velocity	[m/s]
U_r	Reduced velocity	[-]
W	Work done in cross-flow direction	kNm
CFD	Computational Fluid Dynamics	
RPSEA	Research Partnership to Secure Energy for America	
VIM	Vortex Induced Motions	

SEMI-SUBMERSIBLE MODELS

Two different models were tested: a paired-column semi-submersible with eight columns and a semi-submersible with four square columns. Both models were symmetric and had a scale factor of 1:54. This model scale allowed for testing at subcritical Reynolds numbers only. Top sides and the above water part of the floaters were not completely modeled. A photo of the semi-submersible models is shown in Figure 1.

Figure 1: Paired column semi (left) and four column semi model

The paired-column semi main characteristics are summarized in Table 1. For this model no appurtenances were modeled, except for the mooring fairleads and chains on the outer columns. One large draft of 53.3m is tested for which a maximum VIM response is expected.

Table 1: Paired-column semi main characteristics

Inner column span (center to center)	50.3	m
Inner column size (length x width)	10.4 x 14.0	m
Outer column span (center to center)	96.0	m
Outer column size (length x width)	13.4 x 14.0	m
Column fillet radius	2.4	m
Pontoon width	12.8	m
Pontoon height	8.2	m
Inner pontoon length	41.9	m
Outer pontoon length	67.5	m
Draft	53.3	m
As built stiffness surge	528.0	kN/m
As built stiffness sway	536.7	kN/m
As built mass	79487	t

The four column semi was a bare hull and its main characteristics are presented in Table 2. Also for this semi a large draft is tested to encounter large VIM responses. Forces on each of the columns were measured using three component force frames on top of the columns, see Figure 2. To minimize possible cross-talk due to pitch and roll moments, each force frames is installed just above the water level. The columns are water tight and have a hole through the middle to house internal tubes that support the pontoon. This hole is also visible in Figure 2. In between the columns and the pontoon a gap of approximately 2mm is present, see Figure 3.

Table 2: Four column semi main characteristics

Column span (center to center)	71.9	m
Column size (length x width)	22.0 x 22.0	m
Column fillet radius	3.7	m
Pontoon width	20.4	m
Pontoon height	10.7	m
Inner pontoon length	45.7	m
Outer pontoon length	86.6	m
Draft	44.2	m
As built stiffness surge	194.0	kN/m
As built stiffness sway	211.5	kN/m
As built mass	109910	t

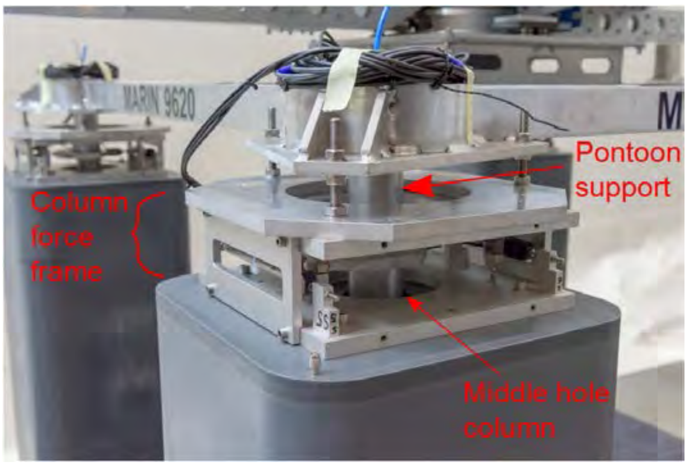

Figure 2: Four column semi: three component column force frames and pontoon supporting frame

Figure 3: Four column semi: gap between column and pontoon

EXPERIMENTAL TEST SETUP

The VIM test campaign was conducted in MARIN's Depressurized Wave Basin (DWB), at atmospheric pressure. The DWB measures 240 x 18 x 8 m with an average steady state tow length of 130 m, allowing between 20-80 VIM oscillations per test. The area blockage ratio is under 1.3%.

Each model is connected to the towing carriage by two pretensioned vertically oriented springs that provide the needed restoring force in the horizontal plane. For each spring the top attachment point is fixed to the carriage and the bottom one to a beam. This beam is connected to a turn-table that can be fixed to the model at prescribed angles providing an easy change of heading. On the turn-table three air-bearings are mounted that allow the model to slide from a smooth granite plate which allows the model to move freely and frictionless in the horizontal plane. This setup is schematically presented in Figure 4. The horizontal stiffness of the vertical mooring system depends on the following parameters: 1) applied pretension in the springs, 2) the spring length, 3) the lever arm at the top side and, 4) the lever arm at the bottom side (for yaw stiffness).

Surge, sway and yaw motions of the semi-submersible model were measured, as well as the horizontal mooring forces at the fairleads, the tension in the springs and the tow velocity. Accelerations were measured using three accelerometers. From these accelerations also yaw accelerations could be deduced.

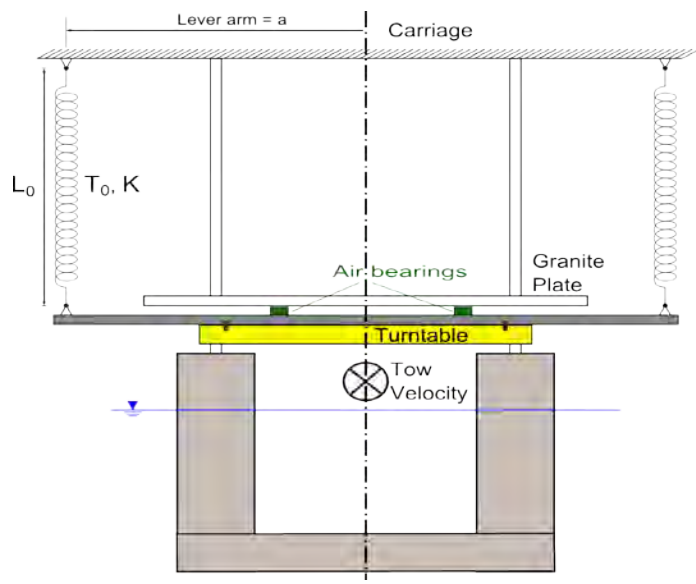

Figure 4: Schematic of the experimental VIM test setup

ACTIVE DAMPING SYSTEM GENERAL DESCRIPTION

In the last decade model tests involving active damping systems were conducted [4-8]. The general idea of these systems is to provide a means of damping in the cross-flow direction. The damping systems make use of additional connections on the model through which a certain damping force is applied. This implies that the damping systems are not fully non-intrusive when the damping system should not add damping.

MARIN developed a novel damping system which is integrated in the standard mooring system and which is as a result non-intrusive when switched off. The damping system consists of a linear actuator which can move the top anchor points of the vertical springs in the cross-flow direction. By imposing motions of the top anchor points, the system can generate a force in-phase or out-phase with the acceleration, thereby introducing an added mass or added damping. The hardware has been chosen such that earlier reported damping levels, see e.g. [8], can be generated by the system. An overview of this new setup is given in Figure 5.

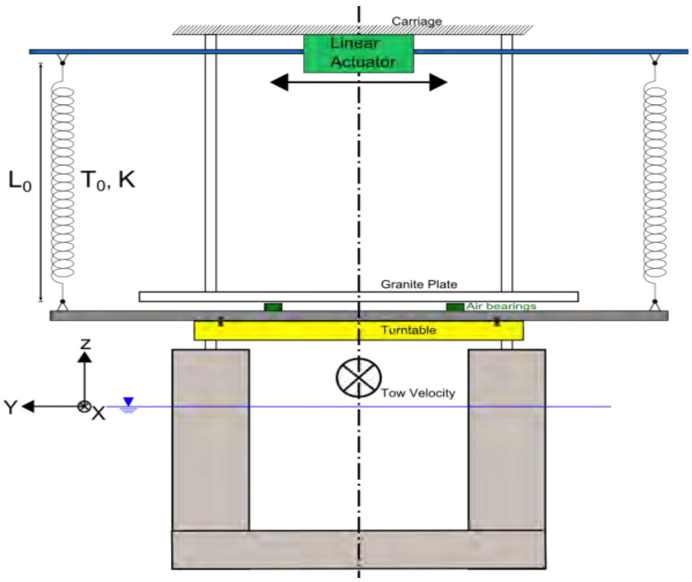

Figure 5: Schematic of the new experimental VIM test setup with optional cross-flow damping

The working principle can be best understood by observing the simplified schematic of the damping system in Figure 6

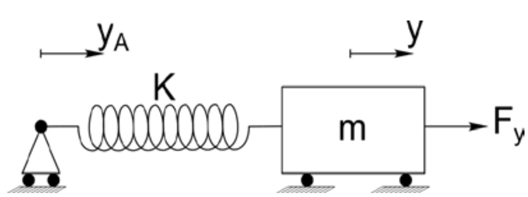

Figure 6 Simplified 1 DOF model of the active damping system, the mooring and the semi-submersible subjected to hydrodynamic loads.

In this schematic y_a is the spring anchor point displacement in the cross-flow direction, K is the equivalent horizontal mooring stiffness, y the model displacement in the cross-flow direction, m_s the semi-submersible mass and F_y the hydrodynamic force acting on the model in the cross-flow direction. The equation of motion for such a system is given by:

$$m_s \ddot{y} + K(y - y_a) = F_y$$

When the displacement of the top anchor points is given by

$$y_a = -\frac{b}{K}\dot{y},$$

the equation of motion can be rearranged as

$$m_s \ddot{y} + b\dot{y} + Ky = F_y$$

yielding an equivalent linear damping with a positive or negative damping coefficient b. In a similar fashion a positive or negative added mass m_a can be accomplished when the displacement of the top anchor points is given by

$$y_a = -\frac{m_a}{K}\ddot{y}.$$

ACTIVE DAMPING CONTROL SYSTEM

The linear actuator of the active damping system is steered by a setpoint position and a setpoint velocity using a sophisticated control algorithm. The control algorithm uses as base input the cross-flow position, velocity and acceleration of the semi-submersible, as visualized in Figure 7.

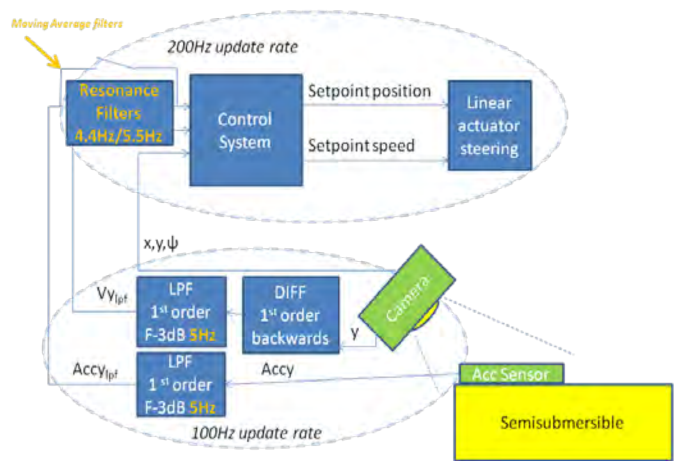

Figure 7: Overview working principle active damping system

The input is updated at a rate of 100Hz. Real time first order, low pass filtering is applied on the velocity and acceleration signals to reduce measurement noise. When filtering is applied with cut-off frequencies much larger than the typical VIM frequency, no significant phase lags enter the actual system output. For filtering with cut-off frequencies just a few Hz above the VIM frequency the phase lags observed in the system output do still not hamper the desired working of the damping system. When the noise in input signals has a frequency close to the VIM frequency and hampers active damping, hardware measures need to be taken to reduce the noise or shift the noise frequency to a higher level such that filtering can be applied. Optionally, also moving average filtering can be applied on the resulting velocity and acceleration signals.

The control system consists of three modules that are updated at a rate of 200Hz. A first module determines the instantaneous horizontal mooring system stiffness based on the installed spring stiffness. This instantaneous stiffness can be a constant specified value or a calculated mooring stiffness dependent on the floater position, which incorporates non-linear effects in the mooring system stiffness. The second module determines the setpoint position of the linear actuator. Hereto, the instantaneous force to be delivered by the active damping system is calculated. This force is determined from a combination of the semi-submersible instantaneous cross-flow position, velocity, acceleration and the amount of damping or added mass to be realized. This instantaneous force combined with the instantaneous horizontal stiffness is translated into a new anchor point position y_a. The third module determines the feed-forward setpoint velocity of the linear actuator. The setpoint value is determined from the time rate of change of the instantaneous force and instantaneous horizontal stiffness. For the calculation of the time rate of change of the instantaneous force, the semi-submersible instantaneous cross-flow velocity and acceleration are needed.

The linear actuator steering block in Figure 7 is a control loop in which the actual measured and determined (control system) setpoint positions and velocities are used to steer the linear actuator. Details of this control loop, which is updated with at a rate of 200Hz, are provided in Figure 8.

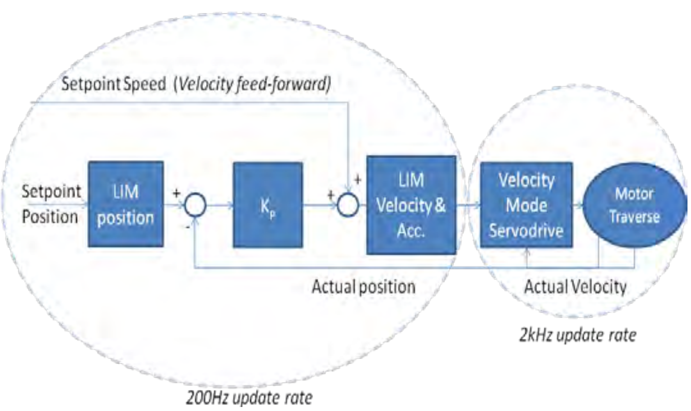

Figure 8: Control loop for linear actuator (LIM) steering

ACTIVE DAMPING SYSTEM INTEGRITY TESTS

The accuracy of the modeling of the added mass or damping is dependent on several factors. Among others the following factors are important for the active damping system performance: 1) accuracy of measurements of kinematics, 2) hardware limitations, 3) filtering and filtering range, 4) accuracy setup related input (stiffness, dimensions, etc.).

To assess the overall performance of the system regarding damping, dedicated integrity tests were conducted. For these tests the semi-submersible model was locked at various positions underneath the granite plate. The active damping system was programmed to deliver a certain amount of damping based on prescribed, fictitious model motions. Two representative frequencies and amplitudes were assessed. An overview of the test conditions and settings is provided in Table 3.

Table 3: Damping system integrity tests parameters (full scale values)

Model positions	x=0m , y=0m, x=0m , y=8.1m, x=8.1m, y=0m, x=8.1m, y=8.1m
Frequency of fictitious motion	f=0.011Hz, f=0.008Hz
Amplitude of fictitious motion	A=4.8m, A=7.6m
Damping level (% of critical damping of 18337.6kNs/m)	5%, 10%

The amount of actively added damping is determined by considering the energy taken from the semi-submersible per oscillation cycle or any time period selected. The change of work is calculated as the in-product of the transverse force measured at the fairleads and the semi-submersible cross-flow motion, both in basin fixed coordinates:

$$\Delta W = \sum_{ts}^{te} F_y \cdot \Delta y ,$$

where *ts* is a start time and *te* refers to the end time. Assuming an equivalent linear damping the change in work can be rewritten as

$$\Delta W = \sum_{ts}^{te} (-b\dot{y}) \cdot \Delta y .$$

From these relations the equivalent linear damping b and damping ratio ζ can be determined:

$$b = \frac{-\Delta W}{\sum_{ts}^{te} \dot{y}^2 \Delta t}, \zeta = \frac{b}{2\sqrt{K(m_a + m_s)}} .$$

The results of all tests can be found in Figure 9. As can be observed the set damping levels are measured back at the model fairleads, independent of the assessed frequencies, amplitude and model offsets. Furthermore, for each damping level, the measured values are constant for the various test, indicating that the repeatability is also good. This proves that the system can be used with confidence as a means to actively add an equivalent linear damping level.

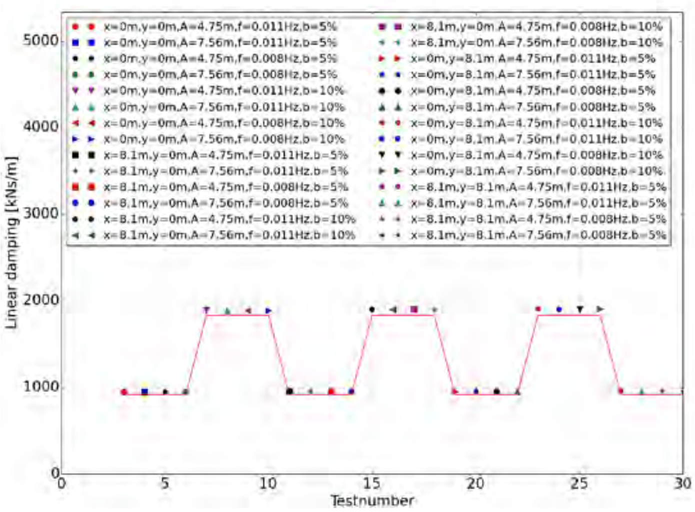

Figure 9:Overview results damping system integrity tests

ACTIVE DAMPING SYSTEM ADVANTAGES AND DISAVANTAGES

The novel active damping system has advantages and disadvantages, which are listed below:

Advantages:

- Easy change of damping levels/added mass levels.
- Non-intrusive, meaning that changing from tests without damping/added mass to tests with damping/added mass can be done quickly.
- Not only linear damping can be modeled, but also quadratic damping.
- Any other force that can be expressed as a function of the horizontal spring stiffness and the top anchor point displacement can theoretically be generated.
- Non-linearity can be handled using analytical relations
- No additional setup time, since the damping device is embedded in the standard setup.

Disadvantages

- Depending on the model setup and responses, the system must be tailored in terms of filtering.
- Real time measurement data needed as input, meaning that possible measurement errors deteriorate the system performance.
- Errors in the input of the system parameters (horizontal stiffness, spring length and anchor positions) result a deterioration of the active damping system performance.
- The damping device only works in the cross-flow direction.

VIM TESTS WITH AND WITHOUT DAMPING

To assess the influence of damping on the VIM response, for both the paired-column semi-submersible and the semi-submersible with four columns VIM tests were conducted with and without active damping. Multiple headings were assessed, but in this paper only the results for the heading with maximum VIM responses are treated: 0 degrees. A more detailed presentation and discussion of the results can be found in Antony et al. [9].

Focus is on the maximum cross-flow motion amplitude $(A/D)_{max}$ and the nominal motion amplitude $(A/D)_{nom}$ that are calculated as follows:

$$\left(A/D\right)_{\max} = \frac{y_{\max} - y_{\min}}{2D},$$

$$\left(A/D\right)_{nom} = \frac{\sqrt{2}\sigma_y}{D},$$

with D the diagonal of the column and σ_y the standard deviation of the cross-flow motion.

For the paired-column semi and 0 degrees heading, damping levels ranging from 1.5% up to 20% of the critical damping of 18337.6kNs/m were tested for reduced velocities from Ur = 3.5 up to Ur=10. The reduced velocity is thereby defined as:

$$U_r = \frac{U_{tow} t_{ref}}{D},$$

with D the diagonal of the column and t_{ref} the sway period as determined from decay tests. In Figure 10 the maximum cross-flow motion amplitudes are plotted for all tested damping levels. The nominal motions are plotted in Figure 11.

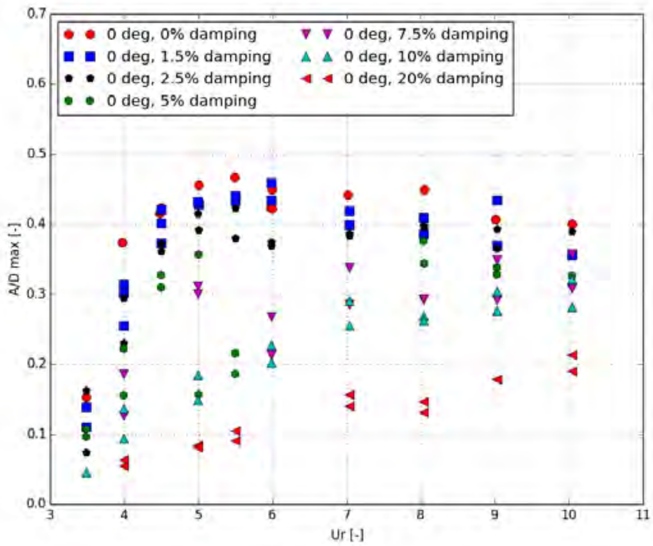

Figure 10: Paired-column semi maximum cross-flow amplitudes for 0 degrees heading

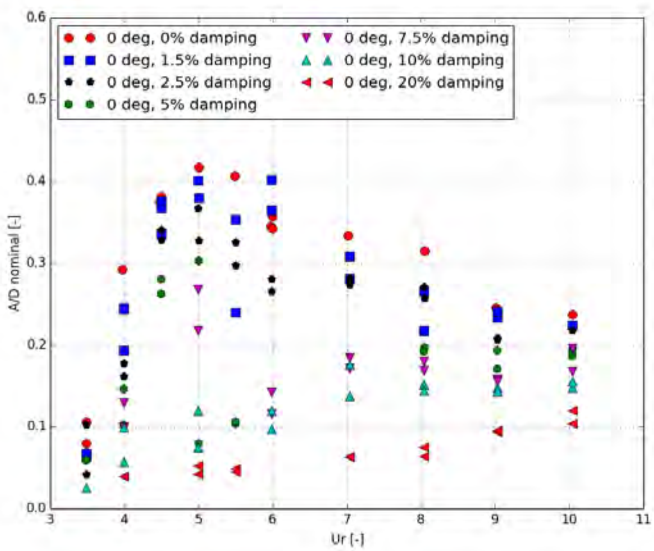

Figure 11: Paired-column semi nominal cross-flow amplitudes for 0 degrees heading

Almost all tests are repeated once and the results show that repeatability proved to be good. Furthermore, it can be clearly seen that damping in the cross-flow direction has a huge impact on the maximum and nominal A/D. Starting with small reductions from 1.5% up to 5%, from 7.5% damping on significant reductions were observed. For 20% damping the peak in the VIM response was completely vanished and the data implies lock-in was not present. A reduction of more than 75% in the nominal A/D was found for 20% critical damping.

For the semi-submersible with four columns, damping levels up to 25% of the critical damping of 13082kNs/m were assessed for reduced velocities from Ur = 4 up to Ur=10. The maximum and nominal A/D are presented in respectively Figure 12 and Figure 13. Again focus is on the heading for which maximum VIM responses are occurring, which is 45 degrees for the four column semi.

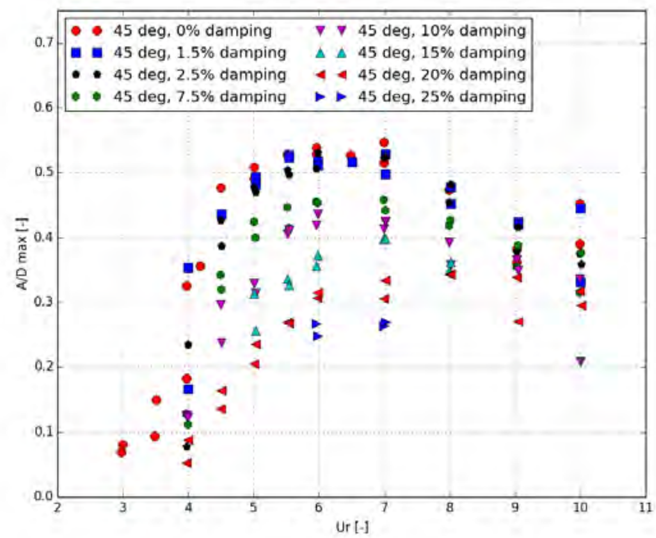

Figure 12: Four column semi maximum cross-flow amplitudes for 45 degrees heading

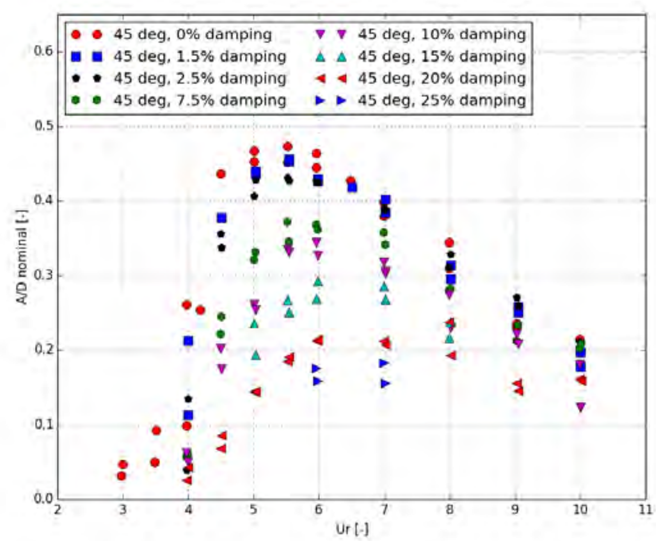

Figure 13: Four column semi nominal cross-flow amplitudes for 45 degrees heading

Just like for the paired-column semi, strong reductions of the VIM response were observed due to cross-flow damping. Reductions started to become significant starting from 7.5% critical damping and reductions of more than 60% in the nominal A/D are observed for 25% critical damping. It can be observed that the nominal A/D peak response seems to shift towards higher Ur's for increased damping levels.

Comparing the impact of damping on the VIM response for the paired-column semi and the four column semi, it is clear that the cross-flow damping had a larger impact for the paired column semi than for the four column semi-submersible.

COLUMN FORCE MEASUREMENTS

Column forces were measured on each columns of the four column semi-submersible. The measured loads include inertia forces, which are subtracted to arrive at the hydrodynamic loads. Using the force measurements on the fairleads, also the hydrodynamic loads on the pontoon can be evaluated. For the load in the cross-flow direction the following relation holds:

$$F_{h,y,pontoon} = m_s \cdot a_y - \sum F_{h,y,columns} - \sum F_{y,fairleads}$$

A straightforward manner to visualize the contribution of each column and the pontoon is to determine for each of them the total work done during the VIM test runs. The work done by the columns and the pontoon is shown for the conventional semi with 45 degree heading and 10% critical damping in Figure 14, which is representative for all 45 degree heading cases. Since forward and backward towing runs are used, solid lines are used to show the trend based on the relative position of the columns with respect to the flow: for a forward run with 45 degree heading, the portside forward column is at 270 degree to the flow direction side (portside), whereas for a backward run this column is on the 90 degree side (starboard).

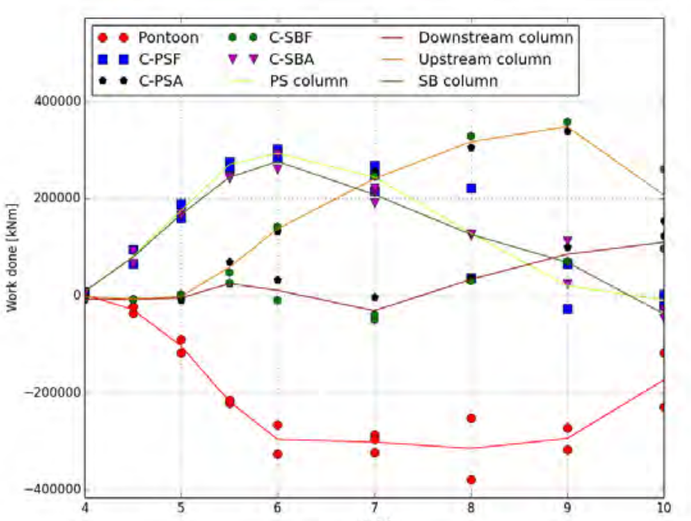

Figure 14: Four column semi work done by columns and pontoon for 45 degrees heading and 10% damping

From Figure 14 it can be concluded that: 1) the pontoon damps for all Ur's the VIM response, 2) the upstream and portside and starboard side columns excite the VIM response and, 3) the downstream column has little impact on the VIM response. For other headings the contributions of the columns can be different and the pontoon always has a damping effect. More results can be found in Antony et al. [9].

CONCLUDING REMARKS

Within the RPSEA project the influence of damping in cross-flow direction on the VIM response was investigated. Hereto, MARIN developed a novel active damping system that allows for a quick change of damping levels. Furthermore, the active damping system is non-intrusive, meaning that when no damping is set no influence of the damping system is present. Integrity tests demonstrated a correct working of the active damping system and VIM tests with damping showed the significance of damping on VIM responses. For the paired-column semi reductions of more than 75% in nominal A/D were found for 20% critical damping. For the four column semi reductions of more than 60% in the nominal A/D were observed for 25% critical damping. The results imply that for future VIM model test damping should be considered to obtain more realistic predictions of VIM response and thus the fatigue of risers.

Column force measurements on the four column semi-submersible showed that for 45 degrees heading for which maximum VIM response occur, the upstream, portside and starboard side columns excite VIM motions. The pontoon always damps VIM responses.

ACKNOWLEDGMENTS

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