Ship-cargo Interaction for Vessels Carrying Large Wind Turbine Monopiles

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Ship-cargo Interaction for Vessels Carrying Large Wind Turbine Monopiles

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Abstract. The increasing demand for decarbonisation to achieve the green transition leads to a higher required capacity for all types of renewable energy farms, including offshore wind. Due to the large required spacing between turbines to maximise their efficiency, their individual size is constantly increasing. A direct consequence for heavy lift and transport ships is that a decreasing number of monopiles can be transported in the cargo hold due to space restrictions. In fact, in many cases monopiles do not fit in the cargo hold at all and are attached to the main deck instead. When lashed on the deck, the monopiles span across most of the length of the ship, and their bending stiffness is significant, as they are designed to withstand harsh ocean conditions. This raises the concern that, depending on the lashing method, the monopiles can have significant effects on the dynamic behaviour of the ship's hull. In this investigation, the ship's hull and the monopiles are modelled as a coupled system with appropriate boundary conditions, and the effects of the number of monopiles and lashing method on the vertical bending responses of the vessel are quantified.

Key words: *heavy-lift ship; offshore wind; ship-cargo interaction; vertical bending*

1. Introduction

Hydroelastic investigations have evolved significantly since the pioneering work by Bishop and Price, and have included a wide range of vessel types [1]. Large bulk carriers and tankers can have natural frequencies that are relatively close to the wave excitation frequency [2].Fast vessels, such as military craft or fast ferries experience high encounter frequencies and potentially slamming [3]. And, finally, container ships, experience significant coupling of their antisymmetric vibrations, which results from their large deck openings [4]. It is clear that hydroelastic responses can become significant for different reasons depending on the case at hand.

In all of the above cases, the bending stiffness is considered to comprise almost entirely of the longitudinal stiffness of the ship hull. The stiffness distribution is sometimes considered [5], but a uniform stiffness is more commonly assumed [6]. Damping originates from both the structure itself, and the hydrodynamics. Out of the two, hydrodynamic damping is significantly easier to compute but its contribution can sometimes be only a small percentage [7]. To further complicate the situation, both types of damping can be affected by the forward speed of the vessel [8]. It is known that cargo can also induce damping [9], but these contributions are virtually never considered. Inertia is the only aspect of the system where the contributions from the ship structure, the hydrodynamics, and the cargo are regularly considered.

Considering only the inertia contributions of the cargo can be sufficient for most situations. However, the green transition has changed the types of cargo some vessels carry dramatically, forcing us to

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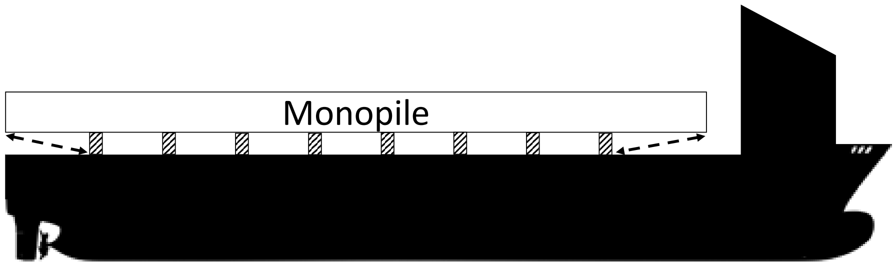


Figure 1: A schematic of the loading condition. The monopile rests on top of the main deck with an interface generated by a series of saddles. Both ends of the monopile are lashed to the ship using pre-stressed springs (dashed arrows).

reconsider this assumption. Heavy-lift vessels nowadays often carry large parts and equipment that relates to the installation of offshore wind turbines, which can include entire monopiles. The size of these turbines and, consequently, monopiles, is becoming progressively larger to accommodate the large need for clean energy. We have already reached the point where the length of the monopile is comparable to that of the heavy-lift vessel carrying it. As a result, the monopile can no longer be stored in the cargo holds and has to be lashed on the deck instead. In order to utilise as much of the cargo capacity (in terms of mass) of the ship as possible, multiple monopiles have to be carried on the deck.

Due to the large size and stiffness of the monopiles, their stiffness contributions would need to be incorporated in the model to achieve accurate dynamic responses. The monopiles typically rest parallel to the ship on a series of saddles that are welded on the deck and are additionally constrained with flexible lashings on either end (Figure 1). As motion in the longitudinal direction of the ship is not fully constrained, the friction between the monopiles and the saddles can potentially induce damping to some of the the mode shapes of the system. Consequently, both stiffness and damping contributions should be considered.

In this investigation, a heavy-lift vessel is modelled as a beam and the monopiles are modelled as additional beams on top. The rigid saddles are fixed on the ship deck and contact is generated between them and the monopiles. The lashings on either end of the monopiles are modelled as linear springs. The contact between the monopiles and the saddles is modelled both as frictionless and frictional. The effects of different levels of lashing stiffness and levels of saddle friction on the dynamic responses are quantified.

2. FEA model setup

A parametric study was performed through 19 cases, which are summarised in Table 1. Three different loading conditions were modelled, namely a) ship without monopiles, b) ship with one monopile on deck, and c) ship with three monopiles on deck. In the case of multiple monopiles, they were modelled as a single beam with the combined mass and stiffness. Since this study is focusing on symmetric responses, this assumption should not affect the accuracy of the results.

Both the ship and the monopiles were modelled in ANSYS as Timoshenko beams (BEAM188), and warping responses were not considered. The main particulars of the ship and the monopiles can be found in Table 2. The stiffness and mass distributions for the vessel were incorporated, as shown in Figure 2. The mass of the ship before the monopiles are loaded is clearly concentrated on the forward end, because that is where both the superstructure and a lot of the machinery are located. Both the ship and the monopiles were considered to be manufactured out of structural steel, with an elastic modulus of 200 GPa and a Poisson's ratio of 0.3. For the monopiles a density of 7850 kg/m^3 was used, whereas for the ship the mass per unit length per section was used instead. The beam representing the ship was modelled at the height of the main deck, with the neutral axis offset at the appropriate distance towards

Table 1: Overview of the examined configurations.

Property	Case number																		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
No monopiles	•																		
1 monopile		•	•	•	•	•	•	•	•	•									
3 monopiles											•	•	•	•	•	•	•	•	•
100% lashing		•	•	•							•	•	•						
66% lashing					•	•	•							•	•	•			
33% lashing								•	•	•							•	•	•
No friction		•			•			•			•			•			•		
Finite friction			•			•			•			•			•			•	
Infinite friction				•			•			•			•			•			•

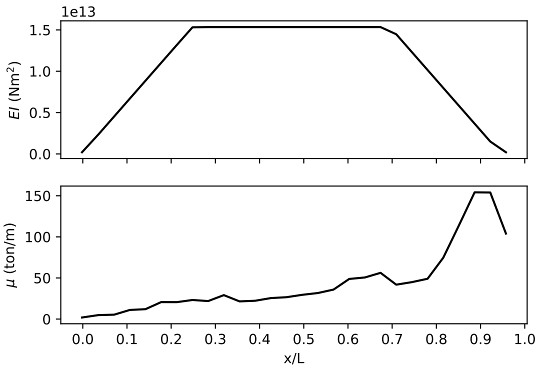


Figure 2: The distribution of the bending stiffness (top) and mass/unit length (bottom) along the length of the ship, before the monopiles are introduced.

the keel. This allowed the saddles to be directly fixed at main deck level. A frequency-independent structural damping coefficient of 0.01 was applied, based on existing literature [10].

The saddles were also modelled as Timoshenko beams, but with a sufficiently large stiffness so that they can be considered rigid. The saddles longitudinal locations are listed in Table 3, and their height (distance between main deck and monopile) was 1.05 m. The longitudinal location of the aft end of the monopiles coincided with the aft perpendicular of the ship. An element size of 2 m was used for the ship and the monopiles. This was slightly modified only to account for the precise locations of the saddles along the length of the ship.

Contact was generated between the top end of the saddles and the monopiles (Figure 3). As the analysis performed within this investigation was fully linear, a no-separation condition was automatically applied. For each of the cases described in the previous paragraph, three levels of friction were used, namely a) frictionless contact, b) frictional contact, and c)"infinite" friction. Due to the linear nature of the analysis, Coulomb friction cannot be applied for frictional contact, but the modelling was included to be used later in nonlinear analysis (not part of this publication). Within the linear solvers, ANSYS treats the frictional contact (with finite or infinite friction), as a full constraint in the x-direction, instead. This, combined with the no-separation condition, practically resulted in a pinned condition at the saddle points.

The monopiles were lashed on the deck using linear springs (COMBIN14) with a stiffness of 45.8 kN/m for the aft lashing and 118.1 kN/m for the fore lashing, respectively. These values correspond

Table 2: Dimensions of the ship and the monopile(s).

Property	Symbol	Value	Units
Ship Length	L	141.0	m
Ship Breadth	B	26.6	m
Ship Depth (amidships)	D	16.0	m
Draft Light Ship	T_0	5.5	m
Draft One Monopile	T_1	6.5	m
Draft Three Monopiles	T_3	7.5	m
Displacement, Light Ship	Δ_0	13,932	ton
Displacement, One Monopile	Δ_1	17,009	ton
Displacement, Three Monopiles	Δ_3	20,223	ton
Amidships 2^{nd} moment of area	I_{yy}	76.7	m^4
Amidships centroid vertical location	z_{CG}	8.0	m
Monopile length	L_m	112.0	m
Monopile diameter	D_m	8.0	m
Monopile thickness	t_m	0.104	m

Table 3: Locations of the saddles.

#	Frame	x (m)	Station
1	23.3	14.64	2.08
2	45.5	32.4	4.60
3	66.2	48.96	6.94
4	86	64.8	9.19
5	94	71.2	10.10
6	110	84	11.91
7	122	93.6	13.28
8	138	106.4	15.09

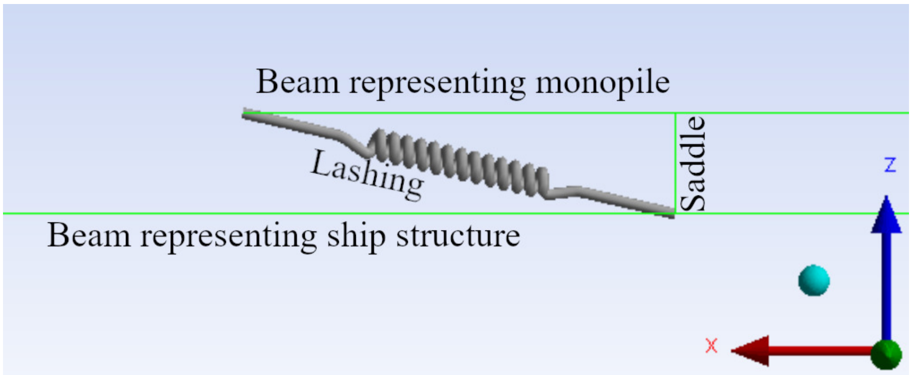


Figure 3: A closeup of the model within ANSYS. The saddles were fixed on the ship side and contact was generated between their other end and the monopile(s).

to the 100% stiffness, and cases with lower lashing stiffness were also investigated (Table 1). A pre-tension of 50 kN was used for each lashing but it was found that its presence made no difference in the responses. The springs connected the ends of the monopiles to the bottom of the nearest saddle (Figure 1). In cases with multiple monopiles, and since the group of monopiles was modelled as a single, stiffer and heavier beam, the lashing stiffness and pre-tension force were tripled.

The structural model was subjected to modal analysis to identify the natural frequencies and mode shapes and then harmonic analysis to calculate the frequency response functions. For the harmonic analysis, a vertical point load was applied to the fore end of the ship and a frequency sweep was performed, in the range of the first four natural frequencies. The hydrodynamic contributions were excluded, as this investigation focused entirely on the interaction between the ship and the cargo and the contributions of the latter to stiffness and damping.

3. Natural frequencies and mode shapes

The mode shapes of the ship without any monopiles on deck are depicted in Figure 4. The modes were scaled to have an amplitude of 1 at the aft end. It is important to emphasise that mode 4 included a significant degree of longitudinal displacement as well as vertical bending, but the longitudinal displacement is not included in the figure. The mode shapes for the monopiles are not depicted, but their first three modes were bending modes and mode 4 was a longitudinal mode. The corresponding natural frequencies are listed in Table 4 (left). As for uniform beams the longitudinal natural frequency only depends on the material properties and the length, the single monopile and three monopiles share that frequency, which in this case is equal to 22.53 Hz.

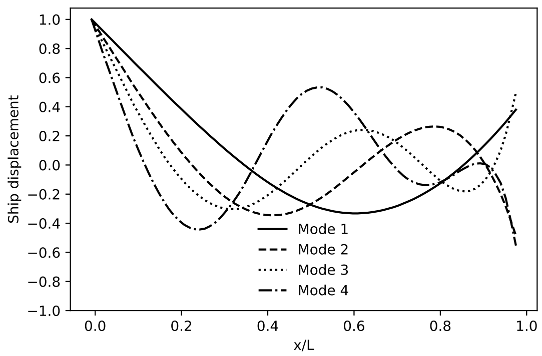


Figure 4: The mode shapes of the ship without any monopiles on deck - displacement in the z-direction is depicted. Mode 4 also includes significant displacement in the longitudinal direction, which is not depicted in this graph.

Table 4: Natural frequencies for the ship and the monopiles before linking them to each other (left) and as a linked system(right). VB stands for vertical bending and L stands for longitudinal mode, with the number preceding indicating the number of nodes. The fourth mode is purely longitudinal for monopile(s) before becoming attached to the ship.

Mode type	Natural frequency (Hz)		
	Ship	1 Monopile	3 Monopiles
2VB	3.07	3.87	3.88
3VB	7.49	10.02	10.27
4VB	12.06	18.16	19.11
5VB/1L*	14.74	22.53	22.53

Mode type		Natural frequency (Hz)	
Ship	Monopile(s)	1 monopile	3 monopiles
2VB	1VB	2.84	2.77
3VB(-1L)	2VB	6.68	6.35
4VB	3VB	11.02	10.95
5VB/1L	4VB/1L	14.40	15.17

The natural frequencies of the coupled system are listed in Table 4 (right). The natural frequencies were found to be reduced compared to their counterparts for the ship without cargo, which indicated that the mass contributions of the monopiles were more significant than the stiffness contributions. The only notable exception to this was the fourth mode, of which the natural frequency is higher for three monopiles than for one. As will be discussed later, this is a special mode as the longitudinal vibrations are about as pronounced as the vertical bending, and it appears that this causes the increased stiffness of the system to dominate over the increased inertia.

The natural frequencies of the ship with one monopile on deck were between 89% and 98% of the corresponding frequencies without any cargo, and this range became 84-102% for the ship with 3 monopiles on board. The higher percentages, included the only recorded increase in natural frequency, corresponded to the fourth mode for both one and three monopiles on board. The corresponding mode shapes are depicted in Figures 5a and 5b, with the displacements plotted separately for the x and z directions, and for the ship and the monopiles. The mode shapes were scaled so that the maximum absolute displacement is equal to 1. In most cases, this corresponded to the displacement of the aft end of the ship, but in other cases it was the fore end or the longitudinal displacement. The natural frequencies and mode shapes in the aforementioned table and figures correspond to cases 2 and 11, that is 100% lashing stiffness and no friction between the monopiles and the saddles. It was found, however, that neither the natural frequencies nor the mode shapes are affected by those parameters. In fact, it was evident that the level of lashing stiffness had insignificant influence on the responses altogether, which was attributed to the no-separation condition applied through the linear contact.

If the stiffness contributions of the monopiles are ignored, and they are treated only as additional masses, the first three natural frequencies increase by 3% and 6% for a single monopile and between 6% and 13% for three monopiles on board. On the other hand, the fourth natural frequency is reduced by 3% and 7% for one monopile and three monopiles, respectively. This further demonstrates the distinctive behaviour of this mode due to its highly coupled nature.

The introduction of the monopiles on the deck of the ship was found to create mode coupling. When carrying one monopile, the first three symmetric modes, namely 2-node bending, 3-node bending and 4-node bending, include longitudinal components for the ship, which are practically absent in the responses of the monopile. The fourth mode, which is 5-node bending for the ship and 4-node bending for the monopile, contains significant longitudinal components for both. When three monopiles are carried, longitudinal components are present for all modes in the responses of both the ship and the monopile, although they are mostly significant for the former. The fourth mode has the highest longitudinal components in all cases.

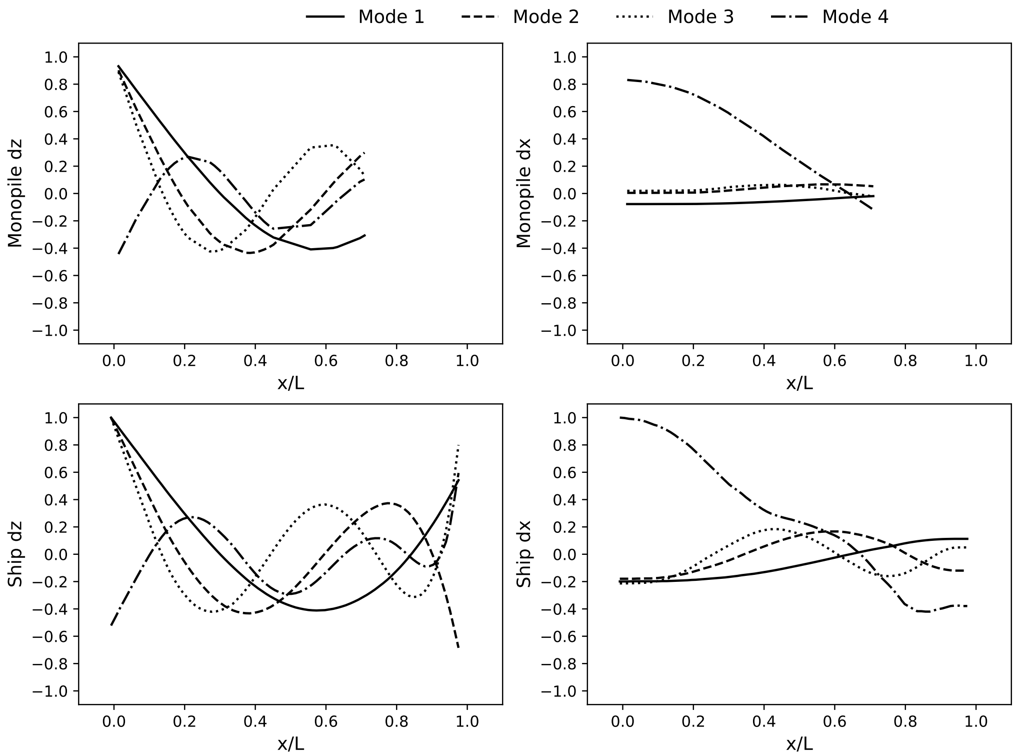
In all cases, the monopiles feature one node less than the ship in vertical bending. So, for example, for the first symmetric mode is the 2-node bending mode for the ship and only 1-node bending for the monopile(s). This is a direct result of the fact that the monopiles tend to follow the ship's shape due to the no-separation condition but span approximately 80% of the length of the ship. Due to the mass and stiffness distribution of the ship, its modes are not symmetric and the nodal positions are shifted towards the forward half. Consequently, the monopiles end before the last nodal point is reached.

4. Frequency response functions

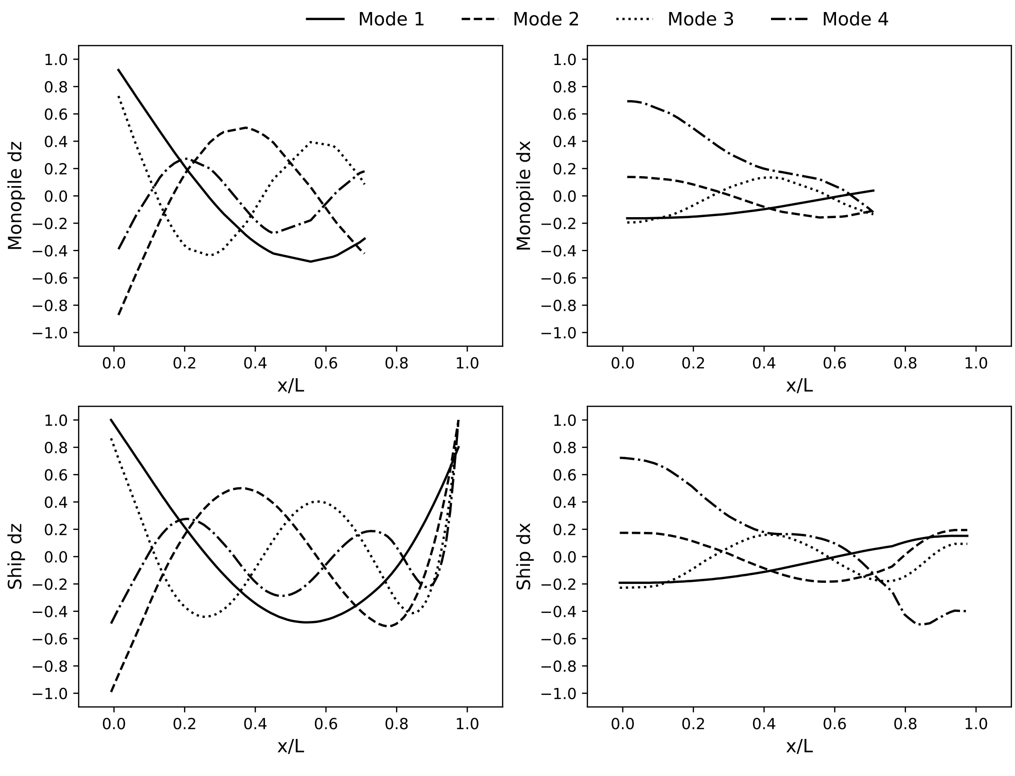
Figure 6 depicts the frequency response functions for the vertical displacement of various points along the length of the ship and the monopile, under a point load excitation. Once more, only cases with zero friction and 100% lashing stiffness are depicted in these plots. The lashing stiffness was found to have no impact on the frequency response functions. The effect of loading, however, is clearly demonstrated: loading the ship with an increasing number of monopiles shifts the peaks to lower frequencies (with the exception of the fourth mode, which shifts to higher frequencies). On the other hand, the trends regarding the magnitude of the response vary per peak as, in some cases, loading with three monopiles generated higher responses than loading with one monopile.

The first peak (2-node bending for the ship), seems to be affected in a limited way for most locations of the ship and the monopile(s), with the exception of 0.375L, where an increasing number of monopiles clearly increases the level of response. Similarly, the second peak (3-node bending for the ship) mostly features a shift in the natural frequency, and dramatic changes in magnitude are only seen on the monopile, at 0.375L and 0.75L. The third peak features an inverse behaviour: the monopile response is largely unaffected by the number of monopiles, whereas the are large differences in in the ship response, particularly near the forward end. The fourth mode is excited significantly less than any of the others, so changes in its magnitude are less important.

The reason why the frequency response functions for other levels of friction are not depicted is that the differences were found difficult to visualise in such a plot. Differences in the amplitude of resonance peaks of the order of a few percentiles were observed as a result of "friction" or, to be precise, the



(a) 1 monopile on deck



(b) 3 monopiles on deck

Figure 5: The mode shapes for the ship with one monopile on deck (top) and three monopiles on deck (bottom). Displacements are provided separately for the ship and the monopile, in the z and x directions.

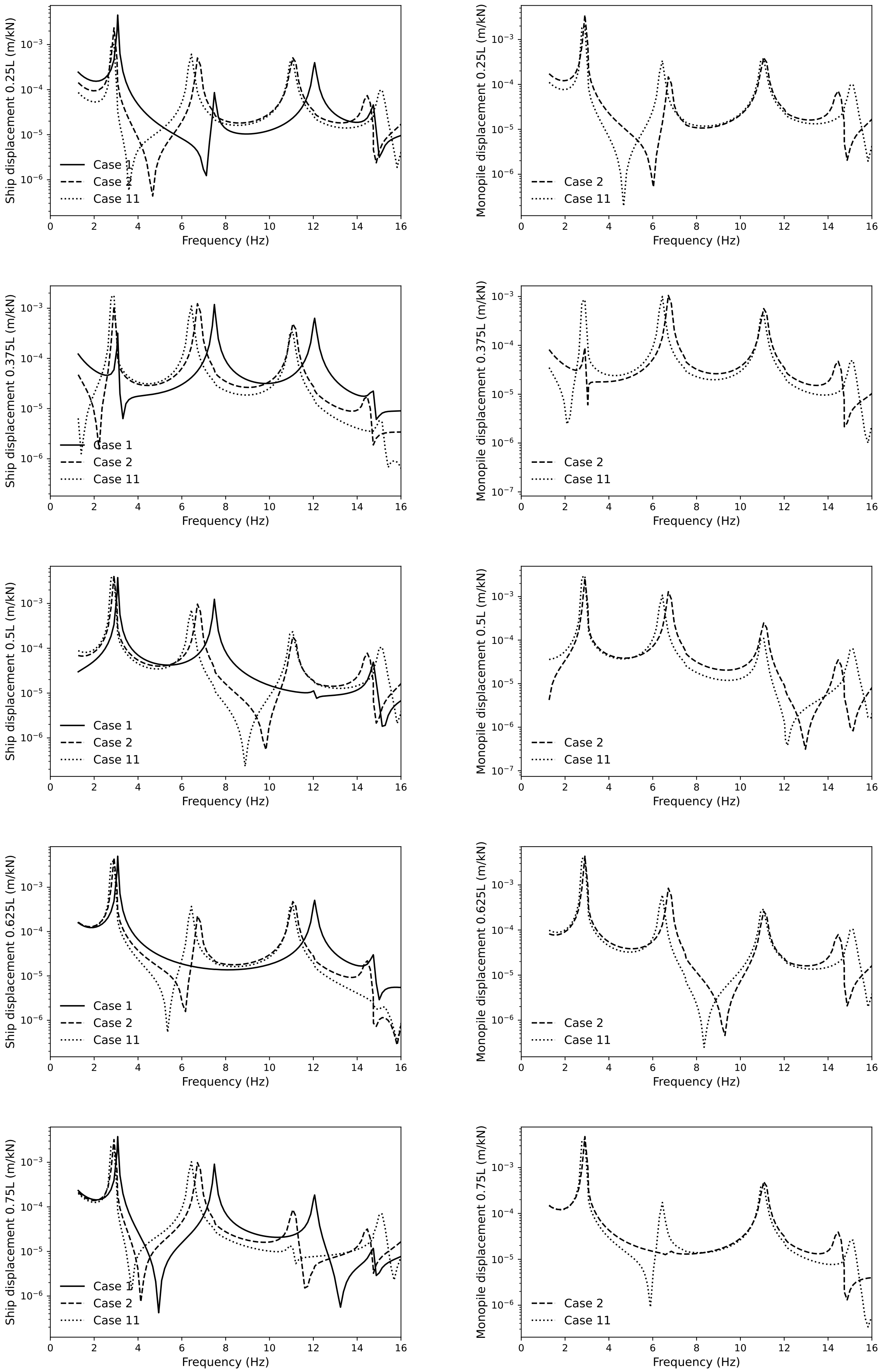


Figure 6: The vertical displacement frequency response functions for various points along the length of the ship (left) and the monopile(right). All cases are for no friction and 100% lashing stiffness.

constraint in the x-direction. As soon as the structure became inherently damped, these small differences in the responses completely disappeared. It was concluded that, for the linear case, the constraint in the x-direction makes no substantial difference. If the fully free case (frictionless) and fully constrained cases (finite or infinite friction) are considered the extremes, and a frictional contact is considered an intermediate case, it is indicated that friction would not cause any difference in the responses. Nonlinear analysis will demonstrate whether this persists when the no-separation condition is absent.

In terms of bending moment responses, the results were also mixed. The maximum bending moment on the ship is reduced by 18% for the first mode when one or more monopiles are introduced. The situation is similar for mode 3 (4-node bending), whereas for mode 2 (3-node bending), the introduction of the monopiles increases the maximum bending moment by 1% and 18%, for one and three monopiles, respectively. The bending loads on the monopiles appear to be reduced between 12% and 18% when moving from one monopile on board to three monopiles on board. Of course, this corresponds to the loads per monopile, whereas the three monopiles together take a larger bending moment than the single monopile. From a practical perspective, this means that carrying more monopiles can reduce the bending loads on them, but this, of course, does not consider potential multi-body interactions between monopiles, particularly if they are stacked on top of each other on deck.

5. Conclusions

The stiffness and damping contributions of cargo are not typically considered when calculating the dynamic responses of ships. In this investigation, a ship carrying one or multiple monopiles for offshore wind turbines was modelled, including the ship-cargo interaction. The monopiles were located on the main deck of the vessel, on top of saddles which were fixed on the deck, and kept in place using flexible lashings. The combined responses were modelled while varying the number of monopiles carried, the lashing stiffness, and the level of friction between the monopiles and the saddles. Fully linear solvers were used, meaning that no separation was allowed during contact, and also friction was linearised.

It was demonstrated that the level of lashing stiffness, at least for the levels investigated, made no difference in the natural frequencies, mode shapes, or frequency response functions. The number of monopiles carried, on the other hand, influenced significantly all of the above. The symmetric modes of the systems were found to include, in many cases, both vertical bending and longitudinal components. The effects of friction on the dynamic responses disappeared as soon as structural damping was introduced in the model. In summary, the effects of the monopiles worked as a competition between their mass and stiffness contributions, with the natural frequencies only being affected in a limited manner because of that.

This pilot study clearly demonstrated that the influence of ship cargo in terms of contributions to stiffness and damping of the system should not be neglected for ships carrying large and stiff cargo, similar to the monopiles in this case. If the wet responses of the vessel were to be considered, it is expected that the natural frequencies would reduce further, and perhaps the longitudinal vibration components would be less pronounced, as their natural frequencies should not be affected by added mass. It is difficult to predict what the effects of potential loss of contact in some of the saddles would be. More accurate prediction in that respect would necessitate the use of nonlinear solvers.

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