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A viscous investigation on the hydrodynamic coefficients and wave loads under the interaction of side-by-side cylinders in regular waves

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ABSTRACT

The paper introduces a two-dimensional Reynolds-Averaged Navier-Stokes Computational Fluid Dynamics (RANS CFD) model to investigate the effects of adjacent floating bodies on the hydrodynamic coefficients and wave loads acting on these bodies. The analysis considers two square cylinders with a narrow gap in way of the free water surface. The physical significance of subsections of the hydrodynamic coefficient matrix becomes evident when comparing coefficients obtained from oscillating-oscillating versus oscillating-fixed models. This understanding is significant, particularly in applications where only specific portions of the hydrodynamic coefficient matrix are utilized, such as in ship-ice collision scenarios (Jiang et al., 2023a). The correlation between wave loads and wave elevations reveals that the sway force and roll moment are associated with the disparity in wave elevation at both sides of each cylinder, while heave forces correlate with the averaged wave elevation on both sides of each cylinder. It is concluded that hydrodynamic interactions are sensitive to the displacement of a floater and accordingly can dominate added mass and damping effects.

1. Introduction

Understanding the hydrodynamic interactions of multiple adjacent floating bodies is relevant for the estimation of motions and wave induced loads. The problem is especially topical in those cases where water flow around floating bodies is influenced by their proximity, thus leading to changes in the surrounding fluid and dynamic pressure fields. For ship-ice collision scenarios, the hydrodynamic coefficient matrix idealising the influences of added mass and damping is partially used to predict the ice loads. Thus, the physical significance of elements of the hydrodynamic coefficient matrix should be clearly understood (Jiang et al., 2023a). Other examples of application may relate to ship operations during replenishment at sea, the decommissioning of floating offshore installations, etc. (Mousaviraad et al., 2016; Islam et al., 2021; Zhang et al., 2021; Zhang and Maki, 2023).

In potential flow theory, the fluid velocity around a body exposed to waves is expressed via the gradient of the velocity potential, which can be decomposed into three components (Martin, 2023) namely (a) the radiation potential referring to the added mass and damping induced by motions (b) the incoming wave potential referring to hydrodynamic actions caused by the incoming waves and (c) the scattered potential describing the changes in the direction and magnitude of fluid flow, see Fig. 1.

To date, scholars that investigated the influence of hydrodynamic interactions (e.g., see Chen and Fang, 2001; Buchner et al., 2001; Huijsmans et al., 2001; Koo and Kim, 2005; Kashiwagi, 2007; Lu et al., 2011; Keijdener et al., 2017; Lagrange et al., 2018; Li, 2020) confirm that accurate prediction of hydrodynamic coefficients is critical for the prediction of motions and wave loads of side-by-side floaters that operate in close proximity. Huijsmans et al. (2001) indicated that the complete matrix of hydrodynamic coefficients is important for accurate prediction of motions. Lu et al. (2011) discovered that the sway wave force is dependent on the wave elevation difference between the two sides of a floating body.

Potential flow theory disregards the influence of viscosity, rotational vorticity, and fluid wake effects. This may lead to the overestimation of hydrodynamic forces at specific frequencies where the gap resonances between floating bodies operating at close proximity may be evident.
Consequently, wave loads may be overestimated. To overcome this problem an artificial damping term may be introduced in way of the gap free surface (e.g., Miao et al., 2006; Huijsmans et al., 2001; Newman, 2003; Chen, 2004; Pauw et al., 2007; Lu et al., 2011; Zou et al., 2023). This approach has been shown to suppress fluid motions by changing the kinematic boundary condition in way of the free surface. However, since the exact value of artificial damping is not predictable, it should be calibrated by using well established experimental data (Lu et al., 2011).

The use of CFD methods could in theory offer an improved alternative since they account for the influence of viscous flow hydrodynamics. For example, Querard et al. (2009) demonstrated that RANS CFD idealises well the viscosity and fluid wake effects and hence it may be used for the evaluation of the added mass and hydrodynamic damping coefficients. In their research, the time history curves of hydrodynamic forces and moments are used in the computation of hydrodynamic coefficients via Fourier analysis (Yeung et al., 1998). Some studies demonstrate that the viscous effects on the added mass are stronger than their effects on damping (Gu et al., 2018; Xu et al., 2019; Chen et al., 2023). Gao et al. (2019) suggested that the incident wave height and gap resonance may influence the wave loads of two side-by-side rectangular bodies. More recently, the combined effects of gap resonance and wave elevation on fluid loads for different water depths, floating drafts, incident wave headings, etc. have been confirmed by various researchers including Zhang et al. (2019), Gao et al. (2020, 2021), He et al. (2021), Zhao et al. (2022), and Zhou et al. (2023).

In potential flow theory the specific physical significance of hydrodynamic actions for bodies operating in proximity remains unclear (Li, 2020; Jiang et al., 2023b). It is therefore essential to elucidate upon their significance when utilizing a partial matrix of hydrodynamic coefficients (Jiang et al., 2023a). To date CFD has neither been concisely used to study the physical significance of each term of hydrodynamic coefficient matrices, nor has been used to study the exact relationship between the fluid loads and wave elevations. This paper introduces a 2D RANS CFD model to investigate the hydrodynamic interaction of side-by-side bodies from two perspectives namely: (a) hydrodynamic coefficients – radiation and (b) wave loads – diffraction. The aim is to better understand (1) the physical significance of each term of hydrodynamic coefficient matrix, (2) how the dynamics of one body affect the added mass and damping of an adjacent floater, (3) what is the relationship between wave loads and wave elevation around side-by-side bodies that operate in proximity.

The analysis considers two square cylinders with a narrow gap in way of the free surface of a 2D numerical wave tank. The Volume of Fluid (VOF) method is used to model the air-water fluid interface. Waves are generated with a mass source wave generator and the wave reflection is restricted by the wave damping zone where additional momentum sinks are distributed (Peric and Abdelmaksoud, 2016). At first instance, the cylinders are placed in calm water and with the aim to study the effects of added mass and damping one or two cylinders are forced to oscillate in sway, heave and roll. Then cylinders are fixed in regular linear waves to study the influence of wave loads and display their relationship with wave elevations. For verification and validation purposes results from CFD are compared against experimental data and a potential flow analysis method (Jiang et al., 2023a).

The remaining of the article is organized as follows. Section 2 outlines theoretical concepts; Section 3 describes the computational model setup of the 2D numerical wave tank (NWT); Section 4 presents the numerical results and a case to study the influence of different drafts. Conclusions are drawn in Section 5.

2. Theory

2.1. Computational fluid dynamics (CFD) model

In this paper, it is assumed that the Navier-Stokes equations govern the two-dimensional (2D) viscous fluid flow dynamics. Accordingly, mass and momentum conservation equations are expressed as

$$\frac{d}{dt} \int_V \rho dV + \int_S \rho \mathbf{v} \cdot \mathbf{n} dS = \int_S p \mathbf{n} dV$$  \hspace{1cm} (1)

$$\frac{d}{dt} \int_V \rho \mathbf{u} dV + \int_S \rho \mathbf{u} \mathbf{v} \cdot \mathbf{n} dS = \int_S (\tau_{ij} - p \delta_{ij}) \mathbf{n} dS + \int_V \rho g \mathbf{k} dV + \int_V q dV$$  \hspace{1cm} (2)

where \( t \) is the time, \( V \) is the fluid control volume, \( S \) is the bounding surface, \( \rho \) is the fluid density, \( \mathbf{v} \) is the velocity vector of the fluid, \( \mathbf{n} \) is the unit vector normal to \( S \) and pointing outwards. The term \( u_i \) represents the Cartesian components of \( \mathbf{v} \) with \( i \) denoting \( x \) axis pointing rightwards, and \( j \) denoting \( y \) axis pointing upwards. The vector \( \mathbf{k} \) is unit value in direction \( x \), \( p \) is the pressure, \( g \) is the gravity acceleration, \( q_i \) is the additional volumetric source term for wave generation, \( q \) is the

(a) Radiation effects accounting for reflected and transmitted waves

(b) Diffraction effects accounting for radiated waves

Fig. 1. The decomposition of velocity potential in a two-dimensional model.
additional momentum source term for wave damping, $\tau_q$ are the components of viscous stress tensor.

If we assume that the fluid is incompressible, the viscous stress tensor is defined as

$$
\tau_{ij} = \mu \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}
$$

(3)

where $\mu$ is the fluid dynamic viscosity.

The additional mass source term, $q_i$, set in a specific source region for wave generation is defined as

$$
q_i = \frac{2c_i}{A} \eta(t)
$$

(4)

where $c$ is the phase velocity of wave, $A$ is the area of source region, $\eta$ is the wave elevation above the wave generator. The additional momentum source term, $\alpha \frac{\partial}{\partial x_i}$, is introduced to the damping zone to minimize the wave reflection. The source strength gradually increases from the start of the damping zone. Thus, $q_i$ can be defined as a function of the fluid velocity and location as follows.

$$
q_i = \rho(f_1 + f_2 |u|) \frac{\partial}{\partial x_i} \left( \frac{1}{\varepsilon - 1} \right)
$$

(5)

$$
\kappa = \left( \frac{x_i - x_{i,ad}}{x_{i,ad} - x_{i,ad}} \right) ^ 2
$$

(6)

In the above equations $f_1$ is the linear damping constant, $f_2$ is the quadratic damping constant, $\varepsilon$ is the character of blending functions, $x_{i,ad}$ is the start coordinate of damping zone, $x_{i,ad}$ is the end coordinate of damping zone. For the detail of wave generation and damping, please refer Peric and Abdel-Maksoud (2015, 2016).

The VOF method is employed to capture the free surface with a volume fraction factor $\alpha$. This is calculated by averaging the density $\rho$ and viscosity $\nu$ in partial elements as,

$$
\rho = \alpha \rho_w + (1 - \alpha) \rho_a = \alpha a_w + (1 - \alpha) a_a
$$

(7)

where $w$ and $a$ represent the water phase and air phase, respectively. In Eq. (7) the volume fraction factor $\alpha$ is set to be unity in the water phase and zero in the air phase. If $0 \leq \alpha \leq 1$ an interface between the two phases is formed. The conservation of $\alpha$ is realized within the context of the transport equation

$$
\frac{\partial \alpha}{\partial t} + \nabla \cdot [\alpha \mathbf{u}] = 0
$$

(8)

The mesh morphing technique which simulates the motions of a cylinder by redistributing the mesh vertices is used to deal with small amplitude oscillations (Siemens, 2021). The Navier-Stokes equations are solved by the Finite Volume Method (FVM). The integral conservation of mass and momentum is solved by a segregated flow solver and a Pressure-Implicit with Splitting of Operators (PISO algorithm) is selected to ensure pressure-velocity coupling (Issa, 1986). The convection flux is computed by a second order upwind scheme (Siemens, 2021). In CFD RANS, turbulence models are required to compute the Reynolds-averaged Navier-Stokes equations. Herein, the SST $k-\omega$ model proposed by Menter (1994) is adopted to treat the transport of turbulent shear stress. The boundary layers around no-slip wall boundaries are idealised by the all $y+$ wall treatment which uses the blended wall function to cover all three sublayers of the boundary layer (Siemens, 2021). To ensure numerical stability, the method assumes that the time step satisfies the Courant-Friedrichs-Lewy condition

$$
C_t \geq \frac{U \Delta t}{\Delta x}
$$

(9)

where $C_t$ is the Courant number, $U$ is the mesh flow speed, $\Delta t$ is the computational time step, $\Delta x$ is the cell dimension.

The pressure on the body surface and velocity gradient around the body can be obtained after solving the Navier-Stokes equations. Thus, the fluid force $\mathbf{F}$ and moment $\mathbf{M}$ on the body are computed by integrating the pressure all over the body surface

$$
\mathbf{F} = - \int_S \rho \mathbf{n} \mathbf{d} S + \int_S \mu \frac{\partial u_i}{\partial x_j} \mathbf{n} \mathbf{d} S = \int_S \mathbf{F} \mathbf{d} S + \int_S \mathbf{G} \mathbf{d} S
$$

(10)

$$
\mathbf{M} = \int_S (\mathbf{x}_i - \mathbf{x}_o) \times (\mathbf{F} + \mathbf{G}) \mathbf{d} S
$$

(11)

where $\mathbf{u}_i$ is the tangent velocity component around the body surface, $S$ is the body surface, $\mathbf{x}_f$ is the centroid of cell face, $\mathbf{x}_o$ is the specified origin vector.

### 2.2. Potential flow (PF) analysis model

Potential flow theory assumes that the fluid is inviscid and irrotational. Therefore, the fluid velocity is expressed by the gradient of a velocity potential namely, $\phi$. As the fluid is considered incompressible, the Laplace equation can be used to express the conservation of mass,

$$
\nabla^2 \phi = \frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} = 0
$$

(12)

The velocity potential $\phi$ can be further sub-divided into the radiation potential $\phi_j$, $j = 1, 2, \ldots, 6$, the potential of incident waves $\phi_{01}$, and the scattered potential $\phi_{12}$. Assuming linear superposition, Li (2020) proposed the following expression for the potential of a multimodal system

$$
\phi = \sum_{m=1}^{M} \sum_{n=1}^{N} \left( -i \omega m \phi_{m} \right) + \phi_0 + \sum_{m=1}^{M} \phi_{m}
$$

(13)

In Equation (13) the upper script $m$ presents the serial number of bodies and $\mathbf{M}$ is the total number of bodies; $\omega$ is the wave frequency; $\eta$ is the amplitude of j-mode motion, $\phi_{j}$ is the radiation potential of j-mode motion; $\phi_{0}$ is the amplitude of incident waves.

According to Jiang et al. (2023b), the added mass and damping of a two-body system can be expressed with the linear superposition principle,

$$
\alpha_{ij} + i \beta_{ij} = - \rho \int_{S} \left( \phi_{m} + \phi_{n} \right) \frac{\partial (\phi_{m} + \phi_{n})}{\partial n} dS
$$

$$
= -\rho \int_{S} \frac{\partial \phi_{m}}{\partial n} \phi_{m} dS - \rho \int_{S} \frac{\partial \phi_{n}}{\partial n} \phi_{n} dS - \rho \int_{S} \frac{\partial \phi_{m}}{\partial n} \phi_{n} dS - \rho \int_{S} \frac{\partial \phi_{n}}{\partial n} \phi_{m} dS
$$

$$
= a_{ij}^{11} + a_{ij}^{12} + a_{ij}^{22} + i a_{ij}^{21} + i a_{ij}^{12} + i a_{ij}^{21}
$$

(14)

where $k,j$ indicate the hydrodynamic coefficients in the $k$th mode because of the $j$th mode motion, $a_{ij}$ is the added mass, $b_{ij}$ is the damping, $\rho$ is the water density, $S$ is the wetted surface, $n$ is the normal of wetted surface. Theoretically, the added mass $a_{ij}^{11}$, $a_{ij}^{22}$ and damping $b_{ij}^{11}$, $b_{ij}^{22}$ terms represent the individual (self-imposed) radiation effect on a self-oscillating floating body. The added mass and damping terms namely $a_{ij}^{12}$, $a_{ij}^{21}$ and $b_{ij}^{12}$, $b_{ij}^{21}$ represent the radiation effects due to body-body hydrodynamic interactions. The wave exciting loads on a body of mass $m$ can be expressed as

$$
F_{m} = -i \omega m \int_{S} \left( \phi_0 + \phi_{*} \right) dS
$$

(15)

### 2.3. Evaluation of hydrodynamic coefficients using CFD

When a body is forced to oscillate at the free surface with an amplitude of $x_i(t) = \eta \sin(\omega t)$, the fluid load can be expressed as
where \(a_{kk}\) and \(b_{kk}\) are the added mass in the \(k\)th mode of motions and \(\omega\) represents the frequency of oscillations. Using Fourier analysis, the instantaneous hydrodynamic coefficients can be obtained from the time histories of hydrodynamics. Yeung et al. (1998) proposed that the hydrodynamic coefficients can be calculated by a single oscillating period \(T\) in the time domain as

\[
a_{kk} = \frac{1}{\pi \eta_k} \int_{-T/2}^{+T/2} F_i(t) \sin(\omega t) dt
\]

(17)

\[
b_{kk} = \frac{1}{\pi \eta_k} \int_{-T/2}^{+T/2} F_i(t) \cos(\omega t) dt
\]

(18)

The above method, as depicted in Fig. 2, allows for the calculation of the hydrodynamic coefficients by using the fluid loads obtained from the CFD simulation. The added mass and damping coefficients per unit length can be dimensionless and defined as

\[
A_{kk} = \frac{a_{kk}}{\rho V} \quad k = 2, 3
\]

(19)

\[
B_{kk} = \frac{b_{kk}}{\rho V \sqrt{g}} \quad k = 2, 3
\]

(20)

\[
A_{kk} = \frac{a_{kk}}{\rho \sqrt{V}} \quad k = 4
\]

(21)

\[
B_{kk} = \frac{b_{kk}}{\rho \sqrt{V}} \quad k = 4
\]

(22)

where \(\nabla\) is the displacement, \(B\) is the width at undisturbed free surface.

### 3. Computational Modelling

The 2D NWT idealization depicted in Fig. 3 features no-slip wall boundary conditions on port, starboard, and bottom, and a pressure outlet boundary condition at the top. The Cartesian coordinate system origin is set at the free surface. Accordingly, in way of the horizontal midpoint, waves propagate along the x-axis and elevate along the y-axis.

The wave generator situated near the left hand side damping zone is practically remote from the cylinders topology. Two damping zones are set at both ends of the NWT to restrict waves reflected by the port and starboard boundaries. According to Perić and Abdel-Maksoud (2016), the length of each damping zone is \(2.5 \times \lambda\) (\(\lambda\) = wavelength). The length of the NWT is approximately \(20 \times \lambda\) and the water depth \(h = \lambda_{\text{max}}\) (i.e., the longest wavelength). Newman (2018) recommends that waves can be idealised in deep waters for \(kh \geq \pi\), where \(k\) is the wave number.

In this paper \(kh \geq 2\pi\) for all cases, hence the influence of water depth is ignored.

The analysis assumes that each cylinder is equidistant to the origin of the coordinate system. Each cylinder has beam, \(B = 0.4\) m and rounded corners with a radius of 0.625% of the beam \(B\) i.e., 2.5 mm. The draught (T) of each cylinder is equivalent to \(B/2\) and two distances (\(B/2\), \(B/4\)) between the two adjacent cylinders are set to investigate the influence of gap width. To measure the wave elevation, three gauges (G1: upwind gauge; G2: gap gauge; G3: downwind gauge) are set in way of the free surface under two distinct scenarios namely (a) cylinder 1 is forced to oscillate while cylinder 2 is fixed at the free surface and (b) both cylinders are forced to oscillate synchronously i.e., at the same frequencies, amplitude, and phase. The oscillation amplitudes of sway, heave, and roll are identical at both settings, measuring 0.01 m for sway and heave, and 0.05 rad for roll. The frequency series are also identical for both settings (see Table 1). The circular frequency is dimensionless and defined as,

\[
\omega' = \frac{\omega}{\sqrt{g}}
\]

(23)

To assess the influence of body-body interactions on the hydrodynamic coefficients the cylinders are forced to oscillate in way of the free surface under two distinct scenarios namely (a) cylinder 1 is forced to oscillate while cylinder 2 is fixed at the free surface and (b) both cylinders are forced to oscillate synchronously i.e., at the same frequencies, amplitude, and phase. The oscillation amplitudes of sway, heave, and roll are identical at both settings, measuring 0.01 m for sway and heave, and 0.05 rad for roll. The frequency series are also identical for both settings (see Table 1). The circular frequency is dimensionless and defined as,

\[
\omega' = \frac{\omega}{\sqrt{g}}
\]

(23)

To assess the influence of body-body interactions on the wave loads, cylinders 1 and 2 are fixed in incident regular waves with and amplitude of 0.01 m. The wave condition is generally consistent in terms of frequency domain discretization (\(\omega' = 0.75\) is added to capture the peak wave loads) as well as the amplitude of oscillations used in the radiation study. Since the investigation is done in 2D three degrees of freedom are considered (sway, heave, and roll).

A display of the hydrodynamic mesh is displayed in Fig. 4. A structured grid is employed to discretize the fluid domain. The mesh in the near field (width = \(10 \times B\) and height = \(7 \times B\)), is refined so that the cell size is less than \(2\% \times B\). The cell height in the vicinity of the free surface is set at 10% of the wave amplitude and the cell aspect ratio is less than 7 as recommended by Querrard et al. (2009). A boundary layer with 0.03 m thickness is established, and the distance between the first node of cell and the wall of the cylinder is assumed small enough to adjust to averaging a wall function \(y^+ < 2\). The latter ascertains that the wall function can appropriately calculate the dimensionless velocity \(u^+\) in the viscous sublayer, where the fluid flow is almost laminar and dominated by viscous effects.

To investigate the influence of mesh refinement a convergence test was carried out. The study considered three mesh configurations (M1, M2, M3) summarised in Table 2. The cell sizes in the near field are set to approximately 2.5%, 1.25%, and 0.625% of the cylinder beam for mesh configurations M1, M2, and M3, respectively. Similarly, cell sizes near...

---

**Fig. 2.** The evaluation of added mass and damping coefficients (pure radiation assumptions).
the free surface are refined according to the same ratios. At first instance the added mass of a cylinder was evaluated while it undergoes sway oscillations at \( \omega^* = 0.7 \). A comparison on the dimensionless sway added masses is illustrated in Fig. 5. As shown in Fig. 5, the difference between M2 and M3 is 0.03% while the difference between M1 and M3 0.34%. The significant increase in cell number from M2 to M3 results in only a minor improvement in precision, indicating that M2 is capable of producing results with sufficient accuracy.

The PISO algorithm (Issa, 1986) was used to idealise the pressure-velocity coupling. PISO is sensitive to time stepping and hence it was ensured that it conforms to the Courant-Friedrichs-Lewy condition (Lu et al., 2020). A constant time step of \( 10^{-3} \) s has been used for the investigation of hydrodynamic coefficients. For the investigation of wave loads, the simulation precision is strongly affected by the wave frequency. This is because the fluid velocity is proportional to the constant amplitude wave frequency. To ensure numerical precision, the time step varied from \( 5 \times 10^{-4} \) s to \( 2 \times 10^{-4} \) as the wave frequency increases, hence the Courant Number \( C_r \leq 0.1 \). Table 3 summarises the time steps and their corresponding \( C_r \) at different wave frequencies.

### Table 1
Circular frequencies used in the radiation and diffraction research.

<table>
<thead>
<tr>
<th>( \omega^* )</th>
<th>Radiation</th>
<th>Diffraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>0.6</td>
<td>0.7</td>
</tr>
<tr>
<td></td>
<td>( \backslash )</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.8</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>1.2</td>
<td>1.2</td>
</tr>
</tbody>
</table>

### Table 2
Configurations of M1, M2, and M3 used in the mesh convergence test.

<table>
<thead>
<tr>
<th>Cases</th>
<th>M1</th>
<th>M2</th>
<th>M3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total cell no.</td>
<td>143882</td>
<td>280060</td>
<td>434614</td>
</tr>
<tr>
<td>Cell no. per wavelength</td>
<td>212</td>
<td>426</td>
<td>512</td>
</tr>
<tr>
<td>Cell no. per wave height</td>
<td>40</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Cell no. per box breadth</td>
<td>40</td>
<td>80</td>
<td>160</td>
</tr>
</tbody>
</table>

### Table 3
Time steps and corresponding Courant number.

<table>
<thead>
<tr>
<th>( \omega^* )</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.75</th>
<th>0.8</th>
<th>1.0</th>
<th>1.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time step [s]</td>
<td>5e-4</td>
<td>5e-4</td>
<td>4e-4</td>
<td>3e-4</td>
<td>3e-4</td>
<td>2e-4</td>
<td>2e-4</td>
</tr>
<tr>
<td>Courant number</td>
<td>0.07</td>
<td>0.09</td>
<td>0.1</td>
<td>0.1</td>
<td>0.1</td>
<td>0.1</td>
<td>0.1</td>
</tr>
</tbody>
</table>
4. Results and discussion

4.1. Prediction of hydrodynamic coefficients

In literature there is no experimental data that may be used to demonstrate the hydrodynamic coefficients under the interaction of side-by-side floaters. In this paper the experimental work of Vugts (1968) has been employed to validate numerical results for the case of a single rectangular cylinder that is forced to oscillate in calm water and at the free surface. The cross section of a cylinder, her floating condition, oscillating frequency and amplitude were identical to those used in existing numerical simulations. The comparison presented in Fig. 6 generally confirms the validity of simulation parameters (e.g., mesh refinement, time step, boundary conditions, turbulence model, velocity pressure algorithm, etc.). Notwithstanding this, roll added mass is approximately overestimated by 30%–50%. The same discrepancy was observed by Querard et al. (2009) and may be attributed to the lack of ability of concurrent RANS CFD turbulence models to idealise flow separation in the vicinity of sharp cylinder corners. It is noted that such uncertainties cannot be removed by reducing the time step (Tam et al., 1996; Bouris and Bergeles, 1999; Cheng et al., 2003).

According to Equation (14), the added mass and damping of a two-body system can be divided into four segments, respectively. As depicted in Section 3, two distinct settings are designed to calculate the added mass and damping of Cylinder 1. In Setting (1), the added mass and damping is induced by the oscillation of Cylinder 1 and is affected by the existence of fixed Cylinder 2. In Setting (2), the added mass and damping of Cylinder 1 is induced by the oscillation of Cylinders 1 and 2. For comparison purposes the results obtained using potential flow theory assumptions for the two-cylinder model are also provided. By adding an artificial damping term $\mu_a$, the boundary condition in way of the free surface in the gap is defined as

$$\frac{\partial^2 \phi}{\partial t^2} + \frac{\partial \phi}{\partial y} + \mu_a \frac{\partial \phi}{\partial t} = 0$$

(24)

The value of the above-mentioned artificial damping term is adjusted to ensure a match between the potential flow and CFD models. Herein, $\mu_a = 0.8$ is used for the calculation of added mass and damping.

Fig. 7 illustrates the sway added mass and damping of cylinder 1 in the single- and the side-by-side two-cylinder configurations. Generally, there is a good agreement between CFD and potential flow results. However, scenario (1) is subject to an intense fluctuation in the added mass curve and a big peak in the damping curve. Both these effects can be attributed to the gap resonance. Fig. 8 illustrates the wave elevation at the center of gap induced by the sway motion in both settings. Scenario (1) exhibits a pronounced peak, which suggests that the gap resonance manifests when only one cylinder oscillates. The correspondence between the frequency of fluctuations in the hydrodynamic coefficients curves and the gap wave curve indicate the influence of gap resonance on the hydrodynamic coefficients. The result is that the added mass increases in the first half of the resonance region and asymptotically decreases for the remaining part of the response. On the other hand, damping increases throughout resonance region. When both cylinders oscillate at the same frequency and phase, the hydrodynamic interaction reduces the added mass of cylinder 1 at low frequencies but increases the added mass at high frequencies. Dissimilarly, the hydrodynamic interaction reduces the damping at all frequencies in scenario (2).

The heave added mass and damping coefficients of cylinder 1 in the single- and side-by-side two-cylinder models are shown in Fig. 9. CFD and potential flow analysis results are in good agreement for the heave added mass. In the single-cylinder model, the corresponding agreement of damping is also satisfactory. However, the damping predicted by the CFD method is larger than the potential flow prediction at all frequencies. In Fig. 10 the large peaks in the curve of scenarios (1) and (2) demonstrate the wave elevation at the center of the gap’s free surface. This suggests that the occurrence of gap resonance is independent of the number of heave oscillating body and contradicts the phenomenon observed during sway oscillations. The higher peak observed in scenario (2) reflects the effects of stronger gap resonance which causes a larger reduction in the added mass and larger increase in the damping of Cylinder 1 (see Fig. 9). The impact of the gap resonance is proportional to its intensity. At the left-hand side of the resonance region, the added mass curves of settings (1) and (2) of the side-by-side two-cylinder
Fig. 7. Non-dimensional added mass and damping coefficients of Cylinder 1 swaying at the free surface, single denotes the single-cylinder model, 1 denotes the scenario (1) of side-by-side cylinders model (cylinder 1 is oscillating and cylinder 2 is fixed), 1 + 2 denotes the scenario (2) of side-by-side cylinders model (cylinder 1 and cylinder 2 are oscillating at same frequency), Potential 1 denotes $a_{11}^{11}$ or $b_{11}^{11}$, Potential 1 + 2 denotes $a_{11}^{11} + a_{12}^{12}$ or $b_{11}^{11} + b_{12}^{12}$.

Fig. 8. Non-dimensional wave elevation at the center of gap in between the two cylinders, $\eta_2$ denotes the wave elevation in the gap, $\eta_3$ denotes the oscillation amplitude of sway, 1 denotes the scenario (1) of side-by-side cylinders model (cylinder 1 is oscillating and cylinder 2 is fixed), 1 + 2 denotes the scenario (2) of side-by-side cylinders model (cylinder 1 and cylinder 2 are oscillating at same frequency).

Fig. 9. The non-dimensional added mass and damping coefficients of Cylinder 1 heaving at the free surface, single denotes the single-cylinder model, 1 denotes scenario (1), 1 + 2 denotes scenario (2), Potential 1 denotes $a_{13}^{13}$ or $b_{13}^{13}$, Potential 1 + 2 denotes $a_{13}^{13} + a_{12}^{12}$ or $b_{13}^{13} + b_{12}^{12}$.

Fig. 10. The non-dimensional wave elevation at the center of the gap in between the two cylinders, $\eta_3$ denotes the wave elevation in the gap, $\eta_3$ denotes the oscillation amplitude of heave, 1 denotes scenario (1), 1 + 2 denotes scenario (2).
model are higher. This indicates that hydrodynamic interactions can increase the heave added mass at low frequencies. A similar phenomenon is also observed in the damping curves. At the right-hand side of the resonance region, the distance between curves is smaller. This suggests that hydrodynamic interactions have smaller impact on the hydrodynamic coefficients at high frequencies.

Fig. 11 displays the roll added mass and damping of cylinder 1 in the single-cylinder model and both settings of side-by-side two-cylinder model. The RANS CFD method predicts greater added mass and damping than potential flow theory as it considers of the flow separation and vortices in way of the cylinder corners (Querard et al., 2009). Both potential flow and CFD methods produce similar variations in terms of hydrodynamic coefficient curves at different frequencies. For example, they all predict that the fluctuation of added mass only happens in scenario (1), since the gap resonance only happens in this scenario (see Fig. 12). The gap resonance amplifies the added mass in the first half and decreases it in the second half of the resonance region. On the other hand, damping always increases. Hydrodynamic interactions lead to decrease of the added mass at low frequencies and increase at high frequencies. Notably hydrodynamic damping decreases throughout the frequency range if no gap resonance occurs (see scenario (2)). Consequently, the roll hydrodynamic coefficients exhibit a response pattern like the one observed for the sway hydrodynamic coefficients.

It can be concluded that in a fixed-oscillating two-cylinder model, the coefficients $a^{11}$ and $b^{11}$ represent the added mass and damping of the oscillating cylinder for a fixed cylinder. In an oscillating-oscillating two-cylinder model, $a^{11} + a^{12}$ and $b^{11} + b^{12}$ respectively represent the added mass and damping of one cylinder under the influence of the other cylinder.

4.2. Wave load assessment

The results depicted in Fig. 13 indicate that the sway force is primarily governed by first order components. This is expected in relatively small amplitude waves. A good agreement between the sway fluid force from CFD and potential flow models is also evident. The sway force curve for each cylinder exhibits a sharp peak around $\omega^* \approx 0.75$. This could be attributed to the gap resonance depicted in the form of a large spike, see Fig. 14. The same figure displays the wave elevations at the downwind side of Cylinder 1, the center of gap, and the upwind side of Cylinder 2. The wave elevation reflects the intensity of fluid velocity variation around the cylinders. According to Bernoulli’s law, the flow rate is closely related to the pressure. Thus, the gap resonance causes fluctuations in pressure on the cylinder surface due to rapid changes in the velocity of fluid flow. Fig. 15(a) demonstrates the maximum difference in wave elevation at the two sides of the cylinder in a wave period, which exhibits similar trends to the curves of sway force in Fig. 13. At low frequencies, the sway forces on both cylinders and the maximum difference in wave elevation at both sides of cylinders are similar. At high frequencies, the sway forces on upwind cylinder are greater than those on downwind cylinder. The features are displayed in the curves of maximum difference of wave elevation at both sides of both cylinders, see Fig. 15. The wave amplitude and the difference of wave phase at different locations should be considered. For instance, the absolute difference (without the consideration of wave phase) in wave elevations between the upwind side of cylinder 1 and the gap can be directly calculated with the values given in Fig. 14. By using this calculation, the difference at $\omega^* = 0.75$ is smaller than the difference at $\omega^* = 1.0$. Notwithstanding this, the sway force is bigger at $\omega^* = 0.75$ (see Fig. 15(a)). The decomposition of sway force reveals different orders of components have distinct correspondence to the difference of wave elevations. The first order component exhibits a response pattern like the wave elevation and the total sway force. The relationship between the second order component of the sway force and the wave elevation might remain unclear because of the small amplitude of the second order component, a consequence of limited wave height.

In Fig. 16, the dimensionless heave force is compared against potential flow analysis results. In a similar fashion to the sway forces, heave forces are dominated by first order response. There is good agreement between CFD and potential flow. The heave force on both

Fig. 11. The non-dimensional added mass and damping coefficients of Cylinder 1 rolling at the free surface, single denotes the single-cylinder model, 1 denotes scenario (1), 1 + 2 denotes scenario (2), Potential 1 denotes $a_{44}^{11}$ or $b_{44}^{11}$, Potential 1 + 2 denotes $a_{44}^{11} + a_{44}^{12}$ or $b_{44}^{11} + b_{44}^{12}$.

Fig. 12. The non-dimensional wave elevation at the center of gap in between the two cylinders, $\eta_g$ denotes the wave elevation in the gap, $\eta_g$ denotes the oscillation amplitude of roll, $B$ denotes the cylinder beam, 1 denotes scenario (1), 1 + 2 denotes scenario (2).
The heave force acting on the upwind cylinder is generally larger. For $\omega^* > 1$, the downwind cylinder experiences nearly zero heave possibly due to the decline of wave diffraction. That is, the upwind cylinder blocks the propagation of waves, which results in smaller wave loads on the downwind cylinder. The latter is confirmed by the wave elevation shown in Fig. 14. The heave force is significantly enlarged in the gap resonance region where the augmentation on the downwind cylinder is larger. As seen in Fig. 16(a), the patterns of the heave force curves closely resemble those of the averaged wave elevation at both sides of each cylinder. Notwithstanding this, the relationship is evident only in the first-order components of heave force.

Fig. 17 shows good agreement between CFD and potential flow results for the dimensionless roll moments. This suggests the strong coupling between the sway forces and roll moments, even under the influence of gap resonance. At low frequencies, the waves diffract through the upwind and downwind cylinders, leading to small differences in wave elevation at both sides of each cylinder (see Fig. 15(a)). Consequently, the two cylinders experience similar roll moments. In way of the gap resonance region, the difference of wave elevations at two sides of each cylinder is enlarged by the resonance, corresponding with enlarged roll moments. At high frequencies, the waves are blocked by the upwind cylinder. Thus, the wave elevation is close to zero in the gap and at the leeward side of the downwind cylinder. However, the wave frequency influences at the windward side of the upwind cylinder is not as strong as the influence at the leeward side (see Fig. 14). Hence, the roll moments of the downwind cylinder are close to zero at high frequencies, and the roll moments on the upwind cylinder at high frequencies are even larger than those at low frequencies. The harmonic analysis indicates that the roll moment is primarily governed by first-order components. The wave elevation predominantly impacts the first-order roll moments.

Fig. 15. The non-dimensional maximum difference of wave elevation at two sides of Cylinder 1 and 2 (a) and non-dimensional mean of wave elevation at two sides of Cylinder 1 and 2 (b), $\Delta \eta = |\eta_u - \eta_g|$ or $|\eta_g - \eta_d|$, $\bar{\eta} = (\eta_u + \eta_g)/2$ or $(\eta_g + \eta_d)/2$, $\eta_u$ denotes the wave elevation at upwind side, $\eta_g$ denotes the wave elevation in the gap, $\eta_d$ denotes the wave elevation at downwind side.
4.3. The influence of gap width

Previous analysis in Section 4 has established a strong link between wave elevation in a gap and hydrodynamic coefficients and fluid loads. It has been observed that gap wave elevation is heightened when the gap fluid resonates at its natural frequency, as reported in studies by Molin (2001), Faltinsen et al. (2007), and Zhao et al. (2018). Molin (2001) specifically notes that the natural frequency of fluid in a gap is influenced by the geometry of the gap, including its width. Reducing the gap width between two cylinders leads to an increase in the natural frequency of gap oscillation, thus shifting the peak of the gap wave elevation curve relative to the wave frequency. Furthermore, Jiang et al. (2023c) suggest that gap resonance is a result of rapid water exchange in and out of the gap, implying that the width of the gap could impact the speed of this water exchange and, consequently, the intensity of the gap resonance. This rationale forms the basis for further investigation into the effect of gap width on hydrodynamic responses. To explore this hypothesis, the gap between two cylinders (as depicted in Fig. 3) was narrowed to a quarter of cylinder beam (B/4) by symmetrically moving the cylinders towards the origin of the coordinate system.

Fig. 18 illustrates the sway added mass and damping of cylinder 1 in the side-by-side two-cylinder configuration with a gap width of B/4. A comparison with the hydrodynamic coefficient curves shown in Fig. 7 reveals similar curve shapes, indicating that the hydrodynamic interactions have comparable effects on these coefficients in the model with narrower gap. The CFD results align well with the potential flow.
outcomes, supporting the notion that the added mass and damping coefficients correspond to $a^{12}$ and $b^{11}$, respectively, in the fixed-oscillating two-cylinder model. In the oscillating-oscillating model, these coefficients correspond to $a^{12} + a^{11} + b^{12}$. A notable difference is the rightward shift in the location of fluctuations, attributed to the gap resonance occurring at a higher frequency in the narrowed gap model, as depicted in Fig. 19(a). Fig. 19(a) indicates that gap resonance is exclusive to the fixed-oscillating model, consistent with the narrowed B/2 gap model.

Figs. 20 and 21 illustrate the heave and roll added mass and damping coefficients of cylinder 1 in the side-by-side two-cylinder models with a gap width of B/4, respectively. As with sway, these figures demonstrate a correlation between the hydrodynamic coefficients in specific scenarios and the respective matrix elements in potential flow theory. They also confirm that large amplitude fluctuations or peaks are closely associated with gap resonance, as evidenced by the matching occurrence frequencies shown in Fig. 19(b) and (c). An intriguing observation is that while narrowing the gap effectively increases the wave elevation at the resonant frequency, its impact on the intensity of fluctuations in the hydrodynamic coefficients is less pronounced. This indicates a more subtle or limited role of gap width in determining the hydrodynamic coefficients in the studied configuration.

Narrowing the gap between two cylinders increases the natural frequency of gap oscillation, thereby altering the peak position of the gap wave elevation curve in relation to wave frequency. This shift impacts the maximum difference and mean of wave elevation at two sides of Cylinder 1 and 2. Observations in section 4.2 suggest that wave loads could be influenced by the change in wave conditions around the cylinders due to their tight connections. In order to test this hypothesis, the model with B/4 gap width is subjected to waves identical to those of fixed cylinders model with B/2 gap width.

Fig. 22 illustrates the non-dimensional wave elevation at the downwind side of Cylinder 1, gap center, and upwind side of Cylinder 2. It reveals that the gap resonance occurs at around $\omega^{\ast} = 0.8$, which is higher than the natural frequency of the wider gap oscillation, as shown in Fig. 14. The curves of wave elevation at the downwind and upwind sides are flatter with less fluctuation than in the wider gap model, reflecting that the behavior of fluid in narrower gap exerts a weaker influence on the fluid at the downwind and upwind sides. The smaller influence could be attributed to the smaller volume of fluid in the narrower gap.

Fig. 23(a) presents the maximum difference in wave elevation at the two sides of each cylinder, where the peaks exhibit at $\omega^{\ast} = 0.8$, coinciding with the natural frequency of gap oscillation. Similarly, the peaks of mean wave elevation, see Fig. 23(b), are also located at the natural frequency, supporting the hypothesis that the maximum difference and mean of wave elevation are strongly associated with the large amplitude wave resonance in the gap. Figs. 24 and 26 show that the trends in sway and roll wave loads mirror the maximum wave elevation difference, while Fig. 25 indicates that heave wave loads follow the trend of mean wave elevation. These findings suggest that gap width minimally affects the relationship between the wave loads and wave conditions around the cylinders. As discussed previously, the relationship is primarily observed in the first order wave loads. Since the wave amplitude (0.01 m) is small as compared to the wavelength (the wavelength range spreads from 0.87 m to 5.03 m) the wave steepness 2A/L is quite small (A is the wave amplitude and L is the wavelength). Thus, the second order components of wave loads are considerably smaller than the first order components because the first and second components are of order of O(2A/L) and O((2A/L)^2), respectively (Rahman et al., 1999). In conclusion the second order components are too insignificant to affect the relationship between the wave loads and the surrounding wave conditions.

4.4. The influence of different drafts

Drafts of adjacent bodies can significantly differ for the case of ship – ice floe interactions (Jiang et al., 2023a & 2023b). This discrepancy may have a significant impact on the hydrodynamic coefficients and wave loads and could be attributed to lack of modelling viscous effects. Fig. 27 depicts a 2D numerical idealization of relevance. The hydrodynamic coefficients of two cylinders are calculated by subjecting them to forced oscillations at the same frequency and phase in way of the free surface of calm water. The coefficients are dimensionless and they are derived according to the method described in Section 2.

Fig. 28 displays the sway added mass and damping coefficients of cylinders 1 and 2. For the downwind Cylinder 1, the added mass coefficients are initially positive at low frequencies but rapidly fluctuate to negative values around $\omega^{\ast} = 0.9$. This behavior is similar to the one obtained for the added mass curve observed in scenario (1) of the equal draft model, see Fig. 7. It suggests that the influence of the upwind cylinder 2 on downwind cylinder 1 is minor. The same result is demonstrated in the damping coefficients, where the curve of downwind cylinder 1 closely resembles the damping curve of scenario (1) rather than scenario (2). This small influence can be attributed to the shallow draft and small volume of upwind of cylinder 2. In contrast, downwind cylinder 2 has a profound effect on upwind cylinder 1. This mirroring trend in the hydrodynamic coefficient curves of upwind cylinder 1 is induced by the influence of downwind cylinder 2. As discussed in Section 4, the added mass and damping coefficients of upwind cylinders may be defined as $a^{22} + a^{12}$ and $b^{22} + b^{12}$. This is because both cylinders are assumed to oscillate. The decomposition of the hydrodynamic coefficients for Cylinder 2, as shown in Fig. 29, reveals that these coefficients are significantly influenced by the hydrodynamic interactions with Cylinder 1. This observation is consistent with draft variations and hence their substantial difference in terms of displacement volume. Therefore, it is worthy to emphasize that hydrodynamic interactions have a transformative effect on the properties of the hydrodynamic
Fig. 30 illustrates the variation of heave added mass and damping coefficients of Cylinders 1 and 2 at drafts of B/2 and B/10, respectively. In a similar fashion to the equal draft idealisations, the troughs in the added mass curves and the peaks in the damping curves are induced by gap resonance effects. The beams of Cylinders 1 and 2 are identical. Thus heave added mass and damping are close in absolute value terms. However, the hydrodynamic coefficients are non-dimensionalised with respect to the mass of each cylinder. Consequently, the non-dimensional hydrodynamic coefficients of upwind Cylinder 2 are approximately five times greater than those of downwind Cylinder 1.

Fig. 31 presents the roll added mass and damping coefficients of Cylinders 1 (B/2 draft) and 2 (B/10 draft) in the side-by-side two-cylinder model configuration. Notably, the added mass coefficients predicted by the CFD method are considerably larger than those evaluated by potential flow theory. Additionally, unlike the results from scenario (2) of the model with identical draft, the gap resonance leads to significant fluctuations in the added mass curves and peaks in the damping curves, even when during synchronous oscillations. This difference could be attributed to the influence of varying drafts that result in different free surface disturbances.

Fig. 32 presents the non-dimensional sway, heave, and roll wave loads. The results follow similar patterns to the sway force variations observed in the identical draft model (see Section 4.2). Hence, the gap resonance appears to induce significant peaks in the curves of wave loads. The disparity between the two cylinders can be attributed to the difference in drafts and in fact suggests direct proportionality to draft variations. As the wave frequency increases, diffraction effects weaken, leading to reduced wave passage through the upwind cylinder. Consequently, at high frequencies the sway force on the upwind cylinder coefficients.

Fig. 30 illustrates the variation of heave added mass and damping coefficients of Cylinders 1 and 2 at drafts of B/2 and B/10, respectively.
becomes higher than that observed on way of the downwind cylinder. A similar trend is depicted in the roll moment curves. For example, the upwind Cylinder 2 experiences smaller roll loads when its draft varies from B/2 to B/10, while the downwind Cylinder 1 shows similar roll loads because its draft does remain intact. Draft changes for Cylinder 2 do not have a significant influence on the heave forces. This observation could be attributed to the consistency of the waterplane area. The heave load acts vertically, and as a result, the influence of draft may be less significant as compared to the effect of waterplane area. As a result, irrespective to the frequency range the heave motion induced wave load on the upwind Cylinder 2 is higher than that on the downwind Cylinder 1. The roll moment is closely coupled with the sway force. This results in higher roll moments on the downwind Cylinder 1 at low frequencies. This observation is also evident in the shapes of the roll moment and sway force curves.

5. Conclusions

The paper discussed the influence of the hydrodynamic interactions between two side-by-side cylinders. Special emphasis has been
attributed on the variation of hydrodynamic coefficients and wave loads. The analysis assumed regular wave patterns generated by the mass source method in a 2D NWT. Wave reflections were prevented by introducing two damping zones according to the momentum source method at both ends of the NWT. The results from potential flow analysis and CFD were assessed and compared with the aim to understand the influence of hydrodynamic interactions on the definition of hydrodynamic coefficients and fluid actions.

The oscillating-oscillating and oscillating-fixed settings of the side-by-side cylinder models lead to the conclusion that the matrix of hydrodynamic coefficients can be partitioned into two groups, resulting from (a) the body’s own motion and (b) hydrodynamic interactions. The
added mass and damping of a cylinder affected by an oscillating cylindrical body may differ from those affected by a fixed cylindrical body. In the presence of a fixed cylinder, added mass and damping are represented as \( a_{1j}^{11} \) and \( b_{1j}^{11} \), respectively. Conversely, under the influence of an oscillating cylinder, the added mass and damping are represented by \( a_{1j}^{11} + a_{1j}^{12} \) and \( b_{1j}^{11} + b_{1j}^{12} \). The degree of hydrodynamic interactions is also proportional to the displacement of the cylinders. The interactions stemming from a cylinder with larger displacement volume can dominate added mass and damping effects of the smaller cylinder. These findings may be significant in applications where partial utilization of the hydrodynamic coefficient matrix is required (e.g., ship-ice collision scenarios). The wave elevation around the side-by-side cylinders is closely linked to linear wave loads. The effects on sway forces are akin to roll moments but distinct from heave forces. Specifically, while sway forces and roll moments correlate well with disparity in wave elevation at the opposite sides of cylinder, heave forces remain directly associated with the mean of wave elevation at both sides. The computation of wave elevation differences and averages must consider the phase of wave elevation at different locations. The gap width plays a role in influencing
hydrodynamic coefficients since it affects the occurrence frequency in way of the gap resonance. Its impact on the intensity of fluctuations within the resonant region appears to be limited. The effect of gap width does not alter the components' physical significance of hydrodynamic coefficient matrix and the relationship between wave loads and the surrounding wave elevation.

The model presented only considers sway, heave, roll and cannot capture the complexities of flow field in way of end gaps. Future work could focus on modelling in three dimensions to capture these complexities. Such investigation may shed light on the influence of surge, pitch, and yaw on hydrodynamic coefficients, the manifestation of flow patterns and the prediction of more realistic fluid actions. This research was conducted in regular waves of small steepness. Thus, the presented studies do not necessarily reveal the exact relationship between the surrounding wave conditions and higher order wave loads. In-depth studies will be conducted in the future to unveil nonlinear effects arising from large wave steepness.

CRediT authorship contribution statement

Zongyu Jiang: Writing – original draft, Visualization, Validation, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. Fang Li: Writing – review & editing, Software, Methodology, Conceptualization. Sasan Tavakoli: Writing – review & editing, Methodology, Formal analysis, Data curation. Pentti Kujala: Writing – review & editing, Supervision, Funding acquisition, Formal analysis, Conceptualization. Mikko Suominen: Writing – review & editing, Supervision, Software, Investigation. Spyros Hirdaris: Writing – review & editing, Supervision, Project administration, Methodology, Investigation, Funding acquisition, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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