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Hydroelastic analysis of underwater rotating elastic marine propellers by using a coupled BEM-FEM algorithm

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ABSTRACT

This paper focuses on the development of a numerical method for analyzing the added mass and damping of a rotating elastic marine propeller. Three-dimensional panel methods in frequency domain combined with the finite element method were employed to study the strongly coupled fluid-structure interactions of the propeller. In order to overcome the computational efficiency problem due to the asymmetric added matrices of the fluid, a mode superposition method in conjunction with Wilson-θ method was employed for calculating the structural responses of the propeller. The validity of the proposed numerical method was confirmed by comparing the present results with experimental data available in the literature and those numerical solutions computed using commercial packages ANSYS and Virtual.Lab Acoustics. The effects of the excitation frequency, inflow velocity, material parameter of propeller and the advance ratio on the added mass and damping of rotating elastic propellers were examined. The results showed that stationary flow may be sufficient for analyzing the wet modes of the propeller at a relatively high excitation frequency, and the non-penetration boundary conditions should be imposed on the deformed blade surface rather than the undeformed surface in the case of relatively lower-frequency excitations. In addition, if the inflow velocity is relatively large, the added damping due to the fluid can significantly affect the unsteady performance of the propeller.

1. Introduction

The knowledge of the vibrational properties of propellers is of great importance for successful design of modern marine propellers, especially for light weight and flexible propellers. As is well-known, the presence of the water around a vibrating propeller may change the dynamic characteristics of the propeller significantly. Generally speaking, the hydrodynamic forces of the fluid acting on a propeller can be represented by the added mass and damping due to the fluid, which are proportional to the structural responses (acceleration and velocity) of the propeller. In certain cases, the added mass and damping due to the fluid can be of the same order of magnitude,or even higher than the structural mass and damping of the solid. Thus, for many marine propellers, the knowledge of the effects of the added mass and damping on the structural behaviors of the propeller is of critical importance for predicting the vibration characteristics of the propulsion system.

In practice, a marine propeller may vibrate in various manners. However, the most important modes of the vibrations of the propeller fall into two categories, i.e., (A) vibrations of the whole propeller, such as surge, sway, heave, roll, pitch and yaw vibration modes, (B) local vibrations of the blades (as distinct from the vibration of the whole propeller). Since the wet modes of the propeller vibrations are different, there will be different effects of the hydrodynamic mass and damping due to the fluid.

Several methods have been proposed in the literature to study the effects of the added mass and damping on the vibrations of marine propellers. Lin and Tsai (2008) examed the free vibration of an underwater composite propeller blade by using the finite element method combined with the panel method. The added mass effect was considered, and the natural frequencies of the blade in wet condition were found to be much lower than those of the blade in dry condition. However, the mode shapes of the blade were found to be almost the same regardless of the blade in wet or dry conditions. Yari and Ghassemi (2016) calculated the added mass coefficients for whole propeller vibrations based on the boundary element method. The results showed that the diameter, expanded area ratio, and the thickness of the proepller have significant influence on the added mass coefficients. MacPherson et al. (2007) proposed a semi-empirical formula to predict the axial water added mass

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and the torsional water moment due to the vibration of the propeller. However, their formulas were based on experimental data of the Wageningen B-series and the KCA series, which may limit the application of their methods. Gaschler and Abdel-Maksoud (2014) obtained the added mass and damping coefficients of heaving motion in ‘A’ kind of vibration for a marine propeller by using a 3-D panel method. Both non-cavitating and cavitating conditions were investigated, and the results revealed that unsteady sheet cavitation seems to have a negligible influence on the magnitudes of the hydrodynamic mass and damping coefficients. Martio et al. (2015) calculated the added mass and damping coefficients in ‘A’ kind of vibration by prescribing sinusoidal translational and rotational motions for a propeller. For determining the effect of the viscosity on the vibrational coefficients, URANS computations were considered in their study. The viscous effects were found to be significant for some coefficients. Mao and Young (2016) applied a 3-D curved lifting line model coupled with a 2-D unsteady thin foil theory to study the added mass and damping coefficients in ‘A’ kind of vibration. It was found that the skew can affect the coupled motions with the sway, heave, pitch, and yaw components of the added mass and damping matrices, while the influence of the skew on the surge and roll components is negligible. The above review indicates that very limited research effort has been devoted to the analysis of hydrodynamic mass and damping for the 'B′ kind of vibration. This motivates the present work.

The objective of the present paper is to develop a highly efficient numerical method to analyze the added mass and damping due to a rotating propeller immersed in water. Regarding the fluid-structure interactions of the propeller, the kinematic boundary conditions on the blade surface were derived by considering the non-penetration conditions. The governing equations of the propeller and the fluid were established, and the added-mass and-damping matrices due to the fluid-structure interaction were obtained in a rigorous way. The finite element method combined with a frequency-dependent panel method was used to formulate the structure model and the added mass and damping matrices due to fluid. Subsequently, a modal reduction technique combined with Wilson-θ method was employed to calculate the structural responses of the propeller, which overcomes the low numerical efficiency due to the asymmetric added matrices of the fluid. The present results were compared with experimental data available in the literature and those numerical solutions obtained from commercial software. Very good agreement was achieved. The effects of the added mass and damping on the unsteady performance of the propeller were investigated.

2. Mathematical model

2.1. Governing equations of propeller blade

For the sake of brevity, a single propeller blade with a unit force applied at the end of the blade is considered. A Cartesian o-*xyz* coordinate system rotating and advancing with the propeller blade is introduced to describe the motion of the blade. The generator line of the blade is coincident with the positive *y*-axis, and the *x*-axis is in parallel with the shaft center line, positive afterward, as shown in Fig. 1. The hub is not considered, and the blade is assumed to be fixed at its root. The blade is considered to be made of an isotropic, homogeneous and linear elastic material. The material properties of the blade are given as: Young's modulus E, Poisson's ratio ν, and density ρ. Due to the irregularity of the geometrical configuration of the blade, three -dimensional linear isoparametric elements are employed to spatially discretize the propeller blade. Each blade element has eight nodes and a total of 24 degrees of freedom.

Using the Lagrange's method, the discretized equations of motion for the elastic blade can be written as:

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + (\boldsymbol{C}_{\Omega} + \boldsymbol{C})\dot{\boldsymbol{u}} + (\boldsymbol{K}_{\Omega} + \boldsymbol{K})\boldsymbol{u} = \boldsymbol{F}_{\Omega} + \boldsymbol{F} + \boldsymbol{F}_{w} \tag{1}$$

where $\ddot{\boldsymbol{u}}$, $\dot{\boldsymbol{u}}$ and $\boldsymbol{u}$ are the vectors containing the nodal acceleration,

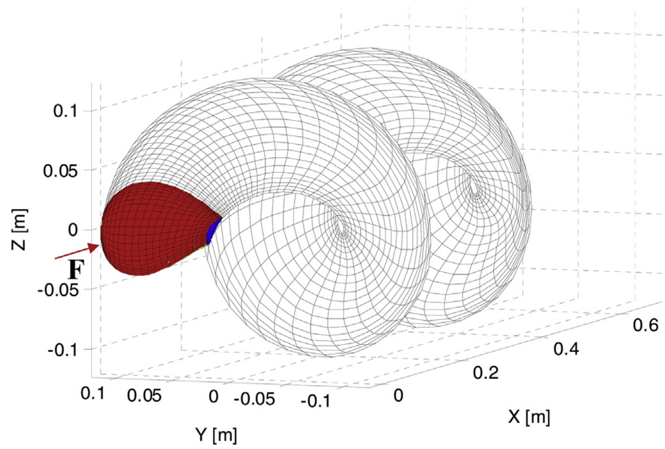

Fig. 1. The propeller-fixed coordinate system which rotates with the propeller (X, Y, Z) coordinate system is shown.

velocity, and displacement components of the blade, respectively; $\boldsymbol{M}$ and $\boldsymbol{K}$ are the global mass and stiffness matrices assembled by the element matrices $\boldsymbol{M}^e$ and $\boldsymbol{K}^e$, defined as: $\boldsymbol{M}^e = \iiint_{V_e} \rho_s \boldsymbol{N}^{\mathrm{T}}\boldsymbol{N}\mathrm{d}V$ and $\boldsymbol{K}^e = \iiint_{V_e} \boldsymbol{B}^{\mathrm{T}}\boldsymbol{D}_s\boldsymbol{B}\mathrm{d}V$, where $\boldsymbol{N}$ is the shape function matrix, $\boldsymbol{B}$ is the strain-displacement matrix and $\boldsymbol{D}_s$ is the material constitutive matrix of each element. The Rayleigh damping model is considered here for taking into account the structural damping of the propeller. In doing so, the damping of the propeller can be represented by a global damping matrix $\boldsymbol{C}$, given as $\boldsymbol{C} = \alpha\boldsymbol{M} + \beta\boldsymbol{K}$, where α and β are two pre-assigned constants. In Eq. (1), $\boldsymbol{C}_{\Omega}\dot{\boldsymbol{u}}$, $\boldsymbol{K}_{\Omega}\boldsymbol{u}$ and $\boldsymbol{F}_{\Omega}$ are the equivalent force vectors due to the centrifugal and Coriolis forces, which are assumed to be negligible when the rotating speed of the blade is small. $\boldsymbol{F}$ is the unit force vector due to the structural forces, and $\boldsymbol{F}_w$ is the hydrodynamic force vector due to the fluid-structural interaction.

2.2. Governing equations of fluid

The flowing fluid surrounding the propeller is assumed to be incompressible, inviscid and irrotational. A panel method is employed to determine the hydrodynamic force vector $\boldsymbol{F}_w$. The total velocity $\boldsymbol{V}_{total}$ of the fluid can be expressed in the sum of a uniform inflow velocity $\boldsymbol{V}_0$ and a disturbed velocity due to a perturbation potential ϕ, given as:

$$\boldsymbol{V}_{total} = \boldsymbol{V}_0 + \nabla\phi \tag{2}$$

where ϕ is the perturbation velocity potential corresponding to the propeller-induced flow field. ∇ is the gradient operator.

According to the potential flow theory and the kinematic boundary conditions of the fluid on the blade surface, the propeller-induced perturbation potential ϕ can be written as:

$$\nabla^2\phi = 0, \text{ in fluid domain } \Omega \tag{3}$$

$$-\frac{\partial\phi}{\partial\boldsymbol{n}} = \boldsymbol{V}_0\cdot\boldsymbol{n} - \frac{\partial\boldsymbol{\delta}}{\partial t}\cdot\boldsymbol{n} + ((\boldsymbol{\delta}\cdot\nabla)\boldsymbol{V}_0)\cdot\boldsymbol{n}, \text{ on blade surface } \Gamma \tag{4}$$

$$\Delta\phi_w(\boldsymbol{R}_{wake}, t) = \Delta\phi(\boldsymbol{R}_{re}, t - t\prime), \text{ Morino Kutta condition} \tag{5}$$

where $\boldsymbol{n}$ is the outward unit normal vector. $\boldsymbol{\delta}$ is the displacement vector for the nodal points on the blade surface. $\boldsymbol{V}_0$ is the velocity of the incoming flow. $\Delta\phi_w(\boldsymbol{R}_{wake}, t)$ represents the potential jump across the wake sheets. $\Delta\phi(\boldsymbol{R}_{re}, t - t')$ is the potential jump across the blade surface at the trailing edge, which is equal to the potential at the upper side (suction side) subtracting that at the lower side (pressure side). t' is the time required for the fluid traveling from the blade trailing edge $\boldsymbol{R}_{re}$ to the wake point $\boldsymbol{R}_{wake}$ along the wake surface. The kinematic boundary condition defined in Eq.

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