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Investigation of vibration characteristics of an energy saving device of pre-swirl stator type

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Abstract

Marine transportation is permanently oriented to the reduction of fuel consumption and nowadays this is even more pronounced due to stringent rules and regulations leading to the reduced ship environmental footprint. For this purpose, there are different technical solutions and operational strategies at disposal. Typical technical solution that reduces ship total power needs (particularly for the propulsion) is the Energy Saving Device (ESD), i.e. special hydrodynamically designed appendage installed on the ship aft body. There are different types of ESDs as for instance ducts, flow control fins (FCFs), pre swirl stators (PSSs), propeller boss cap fins, Costa bulbs, rudder fins, etc., and their design procedures differ from each one to another. If PSS is considered, from a structural point of view, beside its yielding and buckling capacities relevant dynamic properties should be ensured in order to reduce fatigue risks and probable damage accumulation. Energy saving devices could be exposed to different phenomena, as for instance Motion Induced Vibrations (MIV), Vortex Induced Vibrations (VIV) and Turbulence Induced Vibrations (TIV). Proper design from the vibration viewpoint includes natural vibration analysis of PSS fins and comparisons with relevant loadings in order to avoid frequency overlapping, i.e. resonant behavior. In this work the procedure is illustrated on the pre-swirl stator of a tanker. Several approaches to determine PSS fin natural frequencies were examined ranging from approximate procedures to sophisticated FEM approaches combined with different options to account for the effect of added mass. The obtained responses are compared with VIV frequencies and propeller blade frequencies. Special attention is paid to VIV frequency determination where approximate expressions are validated against computational fluid dynamics (CFD) approach.

Keywords: Energy Saving Device; Pre-Swirl Stator; Added mass; Dynamic response; FEM; Potential theory; CFD.

1. Introduction

Strict requirements for environmental protection and the reduction of harmful gas emissions, as well as oscillating oil prices, are some of the challenges that the shipbuilding industry and waterway transport are facing today. In accordance with the Marine Environment Protection Committee (MEPC) of the International Maritime Organization (IMO) resolution has been adopted and a new chapter on ship energy efficiency has been added and according to it, ships engaged in the international shipping should have an International Energy Efficiency Certificate (IEEC) (IMO, 2011). In order to obtain it, ships must comply with the requirements of the Energy Efficiency Design Index (EEDI) and have the Ship Energy Efficiency Management Plan (SEEMP) on board. In spite of different issues with the smooth implementation of the above regulations and its fundamental shortcomings, as discussed in (Ančić et al., 2018a,b; Vladimir et

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al. 2018) there is permanent need of ship-owners to reduce operative costs by fuel savings and consequently to reduce ship environmental footprint. In order to comply with IEEC requirements, there are different technical and operational measures at disposal. In general, technical measures are: measures related to the propulsion system, vessel design and vessel equipment, exhaust after treatment, engine internal measures, use of alternative fuel/energy (LNG, methanol, hydrogen, biofuel, electricity), while the set of operational measures is comprised of: measures related to speed reduction, smart steaming, journey planning, on board information systems, optimal maintenance, etc. Most measures for the reduction of emissions at the same time also reduce fuel consumption, and therefore have both environmental and economic benefits.

Since the ship propulsion is generally the largest energy consumer onboard, energy savings at this point significantly reduce ship fuel consumption and consequently both costs and emissions during the ship operation. For the above-mentioned purpose, the Energy Saving Devices (ESDs), i.e. special hydrodynamic appendages installed on the ship aft body and affecting the inflow or outflow of the propeller, are the promising option (Bakica et al. 2020). ESDs are classified by their relative position to the propeller plane and based on this one can distinguish their working principles. There are different types of ESDs as for instance ducts (Mewis and Guiard, 2011), flow control fins (FCFs) (Song et al. 2019; Huang and Lin 2019), pre swirl stators (PSSs) (Sakamoto et al. 2019, Shin et al. 2019), propeller boss cap fins (Mizzi et al. 2017), rudder bulb-turbine (Wang et al. 2019), Costa bulbs, rudder fins, etc., or even some hybrid solutions (like for instance WAFon-D presented by Kim et al. (2015)) and their design procedures differ from each one to another. Irrespective on the ESD type, the designer should ensure its hydrodynamic performance (shape optimization), structural integrity (both from viewpoint of extreme response and fatigue) as well as appropriate dynamic properties (to ensure that its natural frequencies are out of the excitation frequency range to avoid excessive structural vibration that might lead to damage). Literature survey indicates that most of the references in this field deal with different types of circular ducts and pre-swirl stators, where mostly their hydrodynamic performance is of interest. Regardless of the ESD type, the estimation of its gains is one of the key unknowns (Bakica et al. 2020). Although some model tests indicated promising results at model scale, sea trials showed doubtful contributions of ESDs to propulsion efficiency (Prins et al. 2016). Recognizing that only viscous flow computations can provide acceptable accuracy in ESD performance prediction, Computational Fluid Dynamics (CFD) is widely used approach (Mizzi et al. 2017; Shin et al. 2019). Bearing in mind the fact that assessing ESD performance by means of CFD requires modelling of hull-propeller-ESD interaction (Mofidi and Carrica 2014; Shen et al. 2015), leading to high computational costs, different propeller modelling principles are developed, ranging from relatively complex ones (Wu et al. 2015) to simpler idealised disk approaches (Gokce et al. 2018; Bakica et al. 2019). It should be mentioned that use of simplified propeller models with the ESDs still has to be investigated both from the viewpoint of efficiency prediction and evaluation of loads acting on ESD structure.

Although ESDs are research subject for about 50 years, to the best author's knowledge their structural aspects started to be dealt with thoroughly in the last decade. The reason for this might be in a fact that ESD structure is not reviewed by classification societies, but only its attachment to the hull, as reported by Paboeuf and Cassez (2017). Lee et al. (2016) approximated the non-linear loadings on PSS through trained neural network on a CFD results, while Ju et al. (2018) presented simplified structural safety assessment formula for fin-typed energy saving devices subjected to non-linear hydrodynamic load as well as simplified method predicting the extreme loading and fatigue damage based on that formula. The accuracy is claimed to be lower than CFD calculations but due to its simplicity it is useful at least for quick comparative assessments in the early design stage. Matsui et al. (2018) conducted numerical simulations of structural response of a duct subjected to the impact of an ice block by means of a commercial software package LS-DYNA, which is relevant for ship operation in ice-covered waters. Several years ago, an integral approach to the design of ESD considering all above aspects (hydrodynamic, structural, vibration) is investigated in collaborative project GRIP (Prins et al. 2016; Paboeuf and Cassez 2017). Within the project Paboeuf and Cassez (2017) proposed a numerical approach to evaluate the structural strength of an ESD for which the design waves producing the maximum bending of the fins are determined by potential flow based seakeeping analyses, and the corresponding loads exerted on the ESD are obtained by CFD simulations for previously determined design waves. In Paboeuf and Cassez (2017) the so-called Ship Motions Methodology (SMM) is presented, which enables to determine loads on ESDs in three steps. The first step includes potential hydrodynamic computations for a given ship speed and the outputs of this step are the RAOs (Response Amplitude Operator) calculated for all the ship motions and for all the wave headings. The second step is a spectral analysis using the wave scatter diagram and a selected return period resulting in EDW for a target RAO which is the angle of attack of the fluid relative to the ESD calculated in previous step. The final step is the use of a transfer function to transform the spectral analysis into time domain. From the lift and drag coefficients of the ESD and fluid velocities around the ESD, the lift and drag forces are computed and applied on ESD structural model. Wang et al. (2016) evaluated the fatigue life of a 5-fin PSS fitted ahead of the propeller of an 80,000 DWT bulk carrier. The fatigue loads were exerted on the fins by the stern wake and the ship motion induced velocity fields, neglecting the effect of the propeller induced inflow. A Boundary Element Method (BEM) based on the potential flow theory was employed to evaluate the loads on the fins, while as an input for this method, the viscous wake flow was produced by CFD simulations in calm water, and the motion-induced velocity was derived from potential flow based seakeeping analyses. The BEM pressure distribution is used to load the finite element (FE) model and to extract the hot spot stress at the fin connection, and finally to evaluate fatigue life. Tsou et al. (2019) analysed structural strength of PSS by means of FE commercial code LS DYNA. Also fatigue analysis has been performed for a set of arbitrarily selected sea conditions assuming the well-known Palmgren-Miner summation rule. In this paper some FE modelling aspects like mesh density influence and shell and solid element coupling are addressed. In all above approaches, loadings and responses of ESD structure are considered in a quasi-static manner. It is fair to say that validity of the above model remains unclear and additional research is needed in order to establish rational and efficient procedure for evaluation of structural integrity of ESDs. Based on the above mentioned very recent references, it is evident that ESD are very attractive research topic. This is particularly true for ESDs of PSS type, which is regularly constituted from cantilever fins exposed to significant bending deformations.

Regarding the PSS vibration analysis, that is a subject of this work, to the author's knowledge there is only reference of Paboeuf and Cassez (2017) dealing with this problem. In that work the natural vibration analysis is performed by analytical formulas and FE method (by means of commercial code NASTRAN), while the effect of added mass is calculated by approximate formulas. Comparisons of natural frequencies against propeller-blade frequency and approximately determined VIV frequency are performed.

This paper aims to shed more light on vibration problems of PSS, where research gap regarding the unclear validity of currently applied procedures for wet natural vibration analysis as well as applicability of approximate formulas for VIV frequencies is addressed. In this sense, several options to determine PSS fin natural frequencies by FEM were examined like NASTRAN software (MSC Software 2010) in combination with its internal MFLUID option and Homer software (Malenica et al. 2013) combining NASTRAN and potential flow code HYDROSTAR (Bureau Veritas, 2016). The VIV frequencies are calculated by CFD simulations. Comparisons with approximate expressions are performed and relevant conclusions on their suitability are drawn.

2. Analysis procedure

2.1. General

To illustrate the dynamic analysis procedure a PSS of an oil tanker with main particulars shown in Table 1 is considered.

Table 1. Ship particulars

Length between perpendiculars (L_{PP})	320.0 m
Breadth (B)	58.0 m
Depth (D)	30.0 m
Draft (T)	20.8 m
Displacement (Δ)	312622 m^3
Design speed (U)	16.5 knots
Block coefficient (C_B)	0.81
Vertical centre of gravity (KG)	18.6 m
Moment of inertia (K_{XX}/B)	0.40
Moment of inertia (K_{YY}/L_{PP}, K_{ZZ}/L_{PP})	0.25

The considered ship is equipped with 3-fin PSS shown in Fig. 1. Vibration analysis can be performed at different levels of complexity and accuracy. Natural frequencies can be determined in a simplified manner by analytical formulas for cantilever plates or by the finite element method (Paboeuf and Cassez, 2017). In both approaches added mass effect that reduces fin natural frequency should be taken into account.

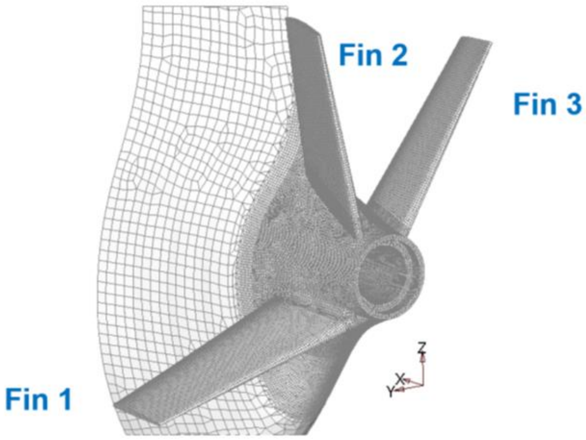


Fig. 1. 3D FEM model of a pre-swirl stator.

2.2. Simplified calculation of PSS fins natural response

According to Paboeuf and Cassez (2017) PSS fin frequencies can be approximated by rectangular plates frequencies with one (shorter) edge clamped. In this case there is analytical formula for calculation of natural frequencies (in Hz):

$$f_i = \frac{\lambda_{ij}^2}{2\pi a^2}\left(\frac{Eh^3}{12\gamma(1-\nu^2)}\right)^{1/2} \quad : \quad i = 1,2,3\ldots; \qquad j = 1,2,3\ldots \tag{1}$$

where λ_{ij} represents dimensionless parameter which is a function of boundary conditions and ratio of plate edges (a/b), E is Young modulus, h is plate thickness while γ and v represent mass per unit area and Poisson´s ratio, respectively. In this analysis, values of boundary parameters are directly adopted from Paboeuf and Cassez (2017) and yield $\lambda_{11}^2 = 3.46$ and $\lambda_{12}^2 = 17.99$. Also, a and b denote plate length and width, respectively.

Effect of added mass is approximately expressed as (Paboeuf and Cassez 2017):

$$\frac{(f_i)_{wet}}{(f_i)_{dry}} = \frac{1}{\sqrt{\left(1+\frac{A_p}{M_p}\right)}} \tag{2}$$

where A_p is added mass and M_p is plate (fin) mass.

According to Paboeuf and Cassez (2017) and Blevins (1979) added mass is predicted using eq. (2) as $2/\pi$ times the volume enclosed by rotating the plate about the longer symmetric axis:

$$2A_p = \left(\frac{2}{\pi}\right)\frac{\pi}{4}b^2 a\rho \tag{3}$$

where ρ is surrounding fluid density.

2.3. FE analysis of PSS fins natural response

Dynamic analysis procedures of PSS fins are illustrated in Fig. 2, where above mentioned options to use Homer or NASTRAN are distinguished. Dry natural vibration is actually very simple, and there are two basic approaches. The first includes modelling of the aft structure with all fins of PSS and constraining the model at some frame of the ship aft body. Another way is to model only one fin with appropriate boundary conditions (mostly clamped at fin connection at the root). It is reasonable to assume that single-fin approach should give slightly higher natural frequencies, as consequence of boundary conditions. Homer requires special finite element model of PSS structure connected to ship outer shell, as shown in the next section. When it comes to the added mass calculation, it can be done directly within NASTRAN, or by means of HYDROSTAR, which is used within the Homer framework for a set of dry natural modes precalculated by NASTRAN utilizing BEM. So, potential discrepancies in the results should be attributed to the added mass calculation procedure because structural model and solver are the same on both options.

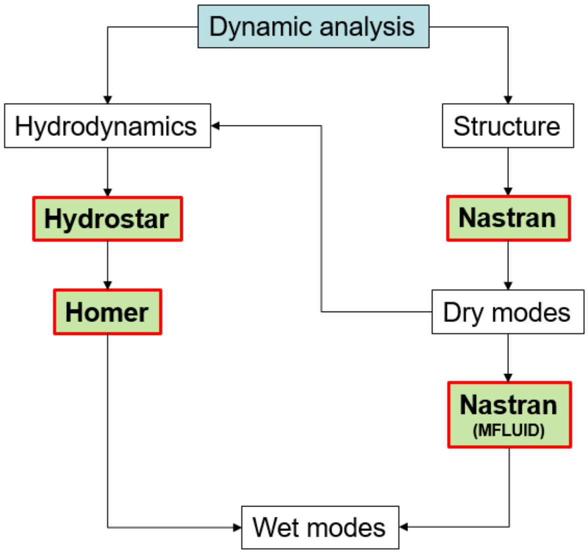


Fig. 2. Dynamic analysis procedures by FEM.

2.4. Propeller blade frequency

Propeller blade frequency is dependent on the blade number and propeller rotation speed. It is calculated according to following equation:

$$f_{propeller} = \frac{RPM \cdot n}{60} H \tag{4}$$

where *RPM* denotes number of revolutions per minute, *n* is number of propeller blades, while *H* denotes number of harmonic.

2.5. Vortex induced vibrations

VIV is an unsteady oscillating flow that takes place at certain conditions when a fluid flows around a body. It can cause severe vibrations that can be reduced by increasing the structural damping or by breaking down the wake pattern by addition of spoilers. Generally, this could be achieved by modifying either the structure or the flow (Paboeuf and Cassez 2017). VIV frequencies can be determined by the empirical formula or by complex CFD computations.

2.5.1. Empirical formula

The frequency f_{VIV}, (in Hz), can be expressed by the following formula:

$$f_{VIV} = \frac{S_t \cdot V}{d} \tag{5}$$

where S_t is Strouhal number, V is fluid velocity and d is length of the profile´s chord.

The Strouhal number depends on Reynolds number defined by:

$$Re = \frac{\rho \cdot V \cdot L}{\mu} \tag{6}$$

where ρ is fluid density, μ is dynamic viscosity and L is profile length.

Relationship between Strouhal number and Reynolds number is available in the literature for different profiles, and here the approach suggested by Paboeuf and Cassez (2017) to consider the fin cross-section as the circular one is followed (Leinhard 1966; Achenbach and Heinecke 1981).

2.5.2. VIV analysis by CFD

In order to simulate the flow separation and to estimate the vortex shedding frequency near the PSS, a full ship model is generated. Given a highly non-uniform flow at the ship stern with a strong pressure gradient, only the full ship model can accurately represent the flow separation occurring at the PSS surface. Also, to consider the angle of attack correctly on the fins for an actual ship in service, the ship is allowed two degrees of freedom for an estimation of the dynamic sinkage and trim which could affect the overall separation frequency for each fin (mainly due to pitch motion). To account for the ship motion, free surface is modelled as shown in Fig. 3. with the snapshot of the mesh near the PSS. Frequency is estimated by analyzing the force signal on the PSS surface when the overall domain (free surface, ship motions) solution converges. The simulation is run only for the calm water case where the inlet speed is set equal as the ship service speed.

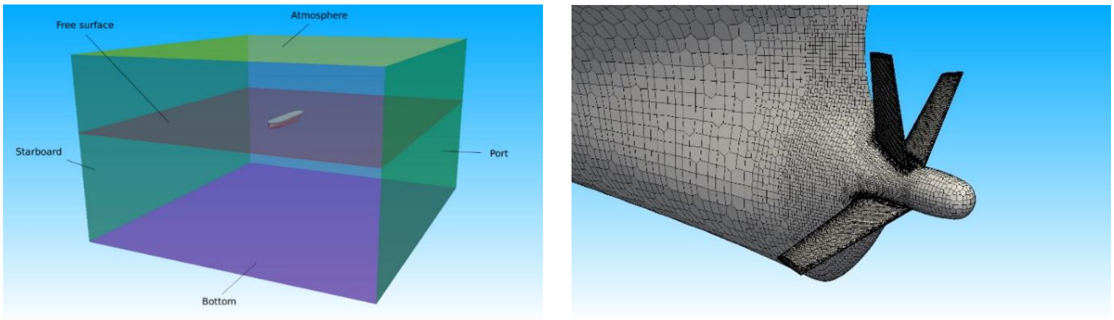


Fig. 3. Computational domain (left) and the computational mesh near stator.

Simulations are performed in the Finite Volume (FV) framework of OpenFOAM which is an actively developed open-source CFD library using a NavalHydro Pack developed in the foam-extend environment (Vukčević 2016). The flow solution is fully non-linear to second-order accuracy. Since the vortex separation is a highly transient phenomena, PISO loop is employed inside a SIMPLE loop to couple the pressure and the velocity solutions. Computational mesh is shown in Fig. 4. Simulation is run on mesh of 1.9M size. Regarding the turbulence modeling, high frequencies are not important in this study since they do not transfer any energy to the structure and goal here is to avoid resonance with the propeller rotation and structural flexible modes. For stated reason, $k-\omega$ SST turbulence model is used in all the simulations for its proven quality when capturing the strong pressure gradient effects from flow separation. It is expected from the simpler turbulence models to properly capture the major features of the flow such as the dominant vortex separation frequency which is relevant for resonance comparison.

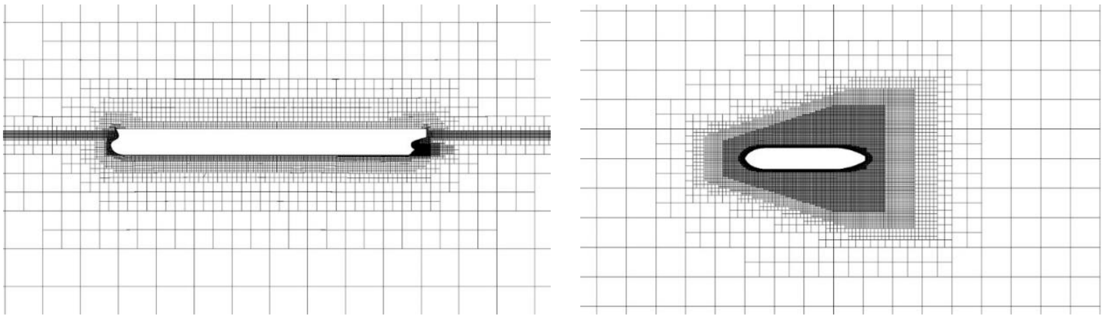


Fig. 4. CFD Mesh.

3. Numerical results and discussion

Based on the previously described procedure, vibration analysis is performed including comparisons of natural frequencies with the excitation ones. For this assessment two fluid models are applied, i.e. MFLUID option in NASTRAN, Fig. 5, and application of HYDROSTAR software within Homer software framework, Fig. 6. In the latter case an artificial FE model is developed, where the ship hull form has been kept in its original shape and ship mass properties are modelled by a concentrated mass element and rigidly connected (RBE2 element) to all finite element nodes except those belonging to the PSS. In this way one obtains wet eigenvalues and eigenvectors for 6 rigid body modes and number of flexible modes related to PSS fins. Calculated values are presented in Table 2.

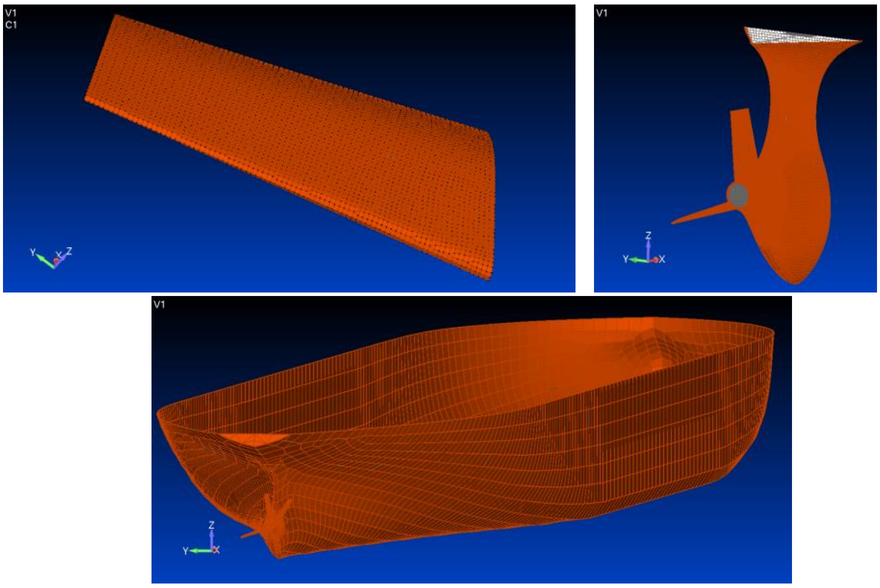


Fig. 5. Different models to predict fin wet natural modes in NASTRAN with MFLUID option (single-fin model, complete aft structure model, complete ship model).

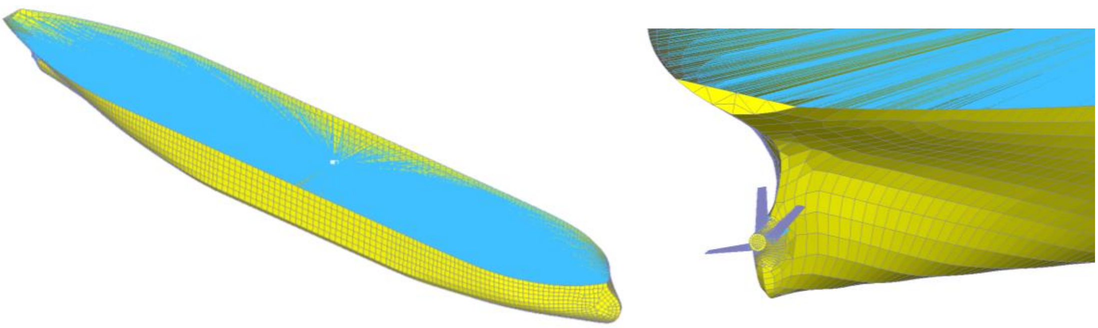
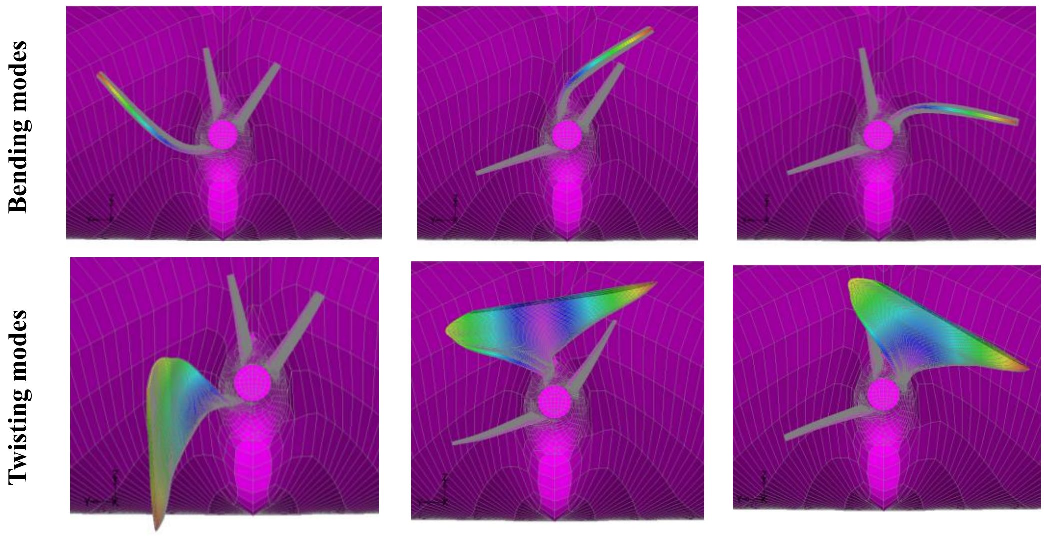


Fig. 6. Artificial FE model for computation of wet natural frequencies by means of Homer.

Table 2. Comparisons of PSS fin frequencies with excitation frequencies (Hz).

Fin no.	Mode	Dry		Wet		Propeller blade frequency	VIV frequency	
		Anal.	FEM	NASTRAN	Homer		CFD	Empirical
1	1st	26.01	29.2	17.96	18.55	4.27	1.41	~1.2
	2nd	98.77	96.73	76.19	77.92	8.53		
2	1st	34.88	39.21	24.66	25.12	4.27	0.71	
	2nd	134.43	116.10	90.63	92.97	8.53		
3	1st	29.22	32.60	19.79	20.76	4.27	0.69	
	2nd	120.66	105.79	81.80	83.20	8.53		

As can be seen from the above results, there is quite low risk of resonance. Bending and twisting modes of all three PSS fins are illustrated in Fig. 7. For the illustration, mode shapes obtained by single fin approach and complete (artificial) FE model for Fin 1 are shown in Table 3.

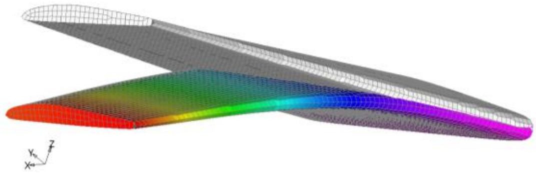
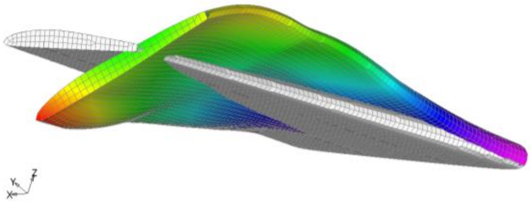
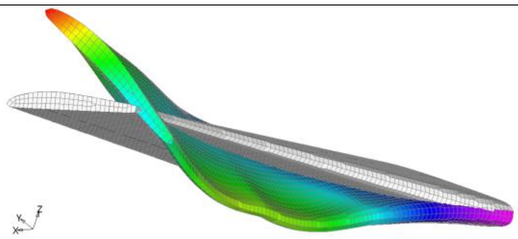
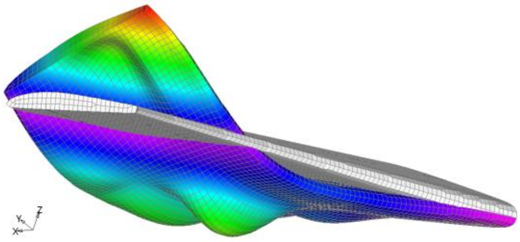


Fin 1 **Fin 2** **Fin3**

Fig. 7. Bending and twisting modes of PSS fins.

VIV frequencies shown in Table 2 are calculated by empirical formula and CFD software OpenFOAM, where significant discrepancies occur. Frequencies from the CFD are calculated from the last 100 seconds of simulation time, after the overall simulation domain has converged. Time signal is processed using Fast Fourier Transformation (FFT) and the dominant frequency is reported. Higher frequency on the Fin 1 is assumed to occur due to different angle of attack compared to other two fins which are located at similar positions with respect to the mean flow at the ship stern. Overall, differences between the analytical formulas and CFD calculations are attributed to 3D effects such as tip vortices, non-uniform flow on the PSS wings due to adverse pressure gradient effects in the ship wake and different angle of attack. Snapshots of performed CFD simulations are illustrated in Fig. 8. Force oscillations for Fin 1 and its pressure distribution for selected time step are shown in Fig. 9. It is fair to say that in spite of relatively large discrepancies, both approaches indicate that there is no resonance risk for the considered structure. Streamlines in the vicinity of PSS fins are illustrated in Fig. 10.

Table 3. Mode shapes and natural frequencies (Hz) of PSS Fin1 obtained by single fin approach, FE analysis.

Mode no.	Mode shape	Dry natural frequency	Wet natural frequency
1		29.20	17.96
2		96.73	76.19
3		103.52	80.11
4		142.77	93.12

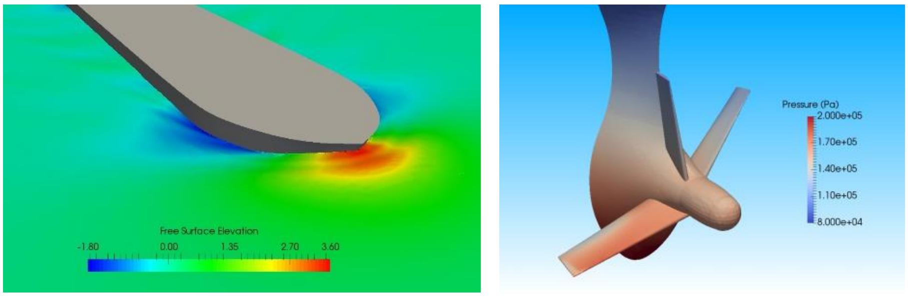


Fig. 8. Snapshots from the CFD simulation.

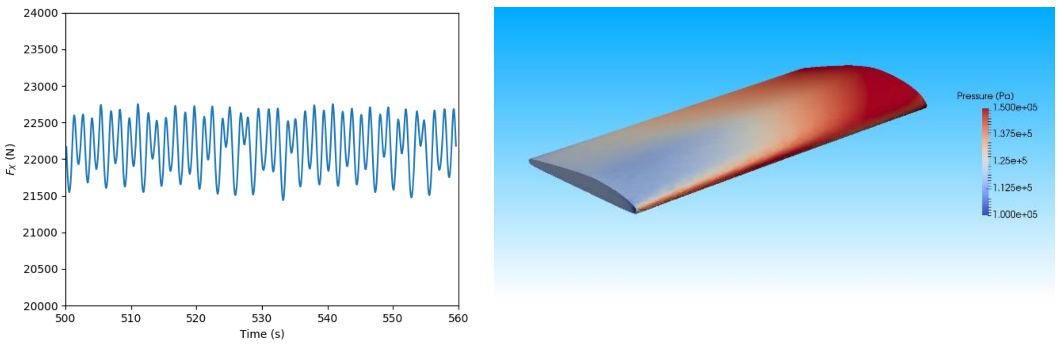


Fig. 9. Axial force fluctuation on Fin 1 (left) with the pressure distribution on the Fin 1 (right).

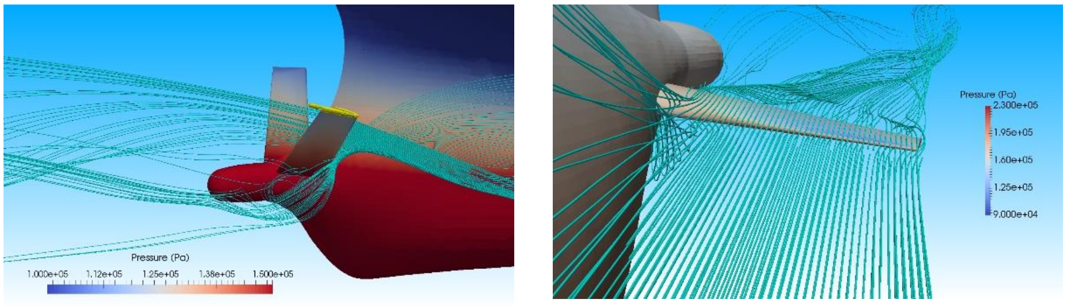


Fig. 10. Streamlines in the vicinity of PSS fins.

4. Conclusion

Efforts to reduce ship fuel consumption and consequently to lower its environmental footprint motivate shipbuilders to develop different types of energy saving devices. For such devices regularly empirical design procedures with unclear levels of accuracy and validity are applied. In order to increase ESDs operational safety, beside hydrodynamic performance assessment, their design should include accurate strength and vibration analyses. In this paper a dynamic analysis of pre-swirl stator type energy saving device is considered. Natural vibration analysis of PSS fins is performed by analytical formulas and by means of commercial software NASTRAN and Homer, utilizing FE method. Natural frequencies are compared against propeller blade excitation frequencies and VIV frequencies. The later ones are determined by empirical expressions and by the sophisticated CFD computations by means of open source code OpenFOAM. So, the paper original contribution lies in the evaluation of different options to calculate fin natural frequencies and validation of empirical formula to predict VIV frequencies against complex CFD computations. The main findings of this investigation can be summarized as follows:

- Dry natural vibration analysis performed by analytical formulas leads to the same conclusion as FEM results, i.e. there is no resonance risk for PSS fins. However, the discrepancies between results are rather high indicating that this approach might not be fully reliable. This is particularly case if the fin shape and internal framing is complex.
- Effect of added is significant and should be taken into account.
- There are different structural modelling options of PSS fins, and single-fin approach with appropriate boundary conditions (clamped) seems to be reasonable option.
- Wet natural vibrations can be analysed both by NASTRAN with MFLUID option or by combining NASTRAN with HYDROSTAR within the Homer framework. For this specific application, the former approach seems to be more user friendly since generation of separate hydrodynamic model is not needed.
- CFD results show that empirical formula for VIV frequency results in relatively high scattering of the results. Although it might be good for some specific applications, CFD simulations are highly recommended if accurate investigation of VIV phenomenon is needed.
- If evaluation of structural integrity of the considered PSS structure is needed, quasi-static approach can be reliably used, because overlapping of natural and excitation frequencies both for the lowest bending and twisting modes is avoided.

Future investigation of PSS design procedure should be focused on development of reliable methods for the structural integrity evaluation, where special attention should be paid to definition of relevant sea conditions, performing of CFD computations for those conditions, reliable transfer of calculated loadings and finally the structural analysis and comparisons with relevant criteria. It should be also mentioned that design criteria for ship appendices like PSS are not fully mastered today and require more attention.

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