Prediction of induced vibrations for a passenger - car ferry

L Crudu¹, ², O Neculet² and O Marcu¹

¹Naval Architecture Department, “Dunarea de Jos” University of Galati, Galati, Romania
²NASDIS Consulting SRL, Galati, Romania

E-mail: liviu.crudu@ugal.ro

Abstract. In order to evaluate the ship hull global vibrations, propeller excitation must be properly considered being mandatory to know enough accurate the magnitude of the induced hull pressure impulses. During the preliminary design stages, the pressures induced on the aft part of the ship by the operating propeller can be evaluated based on the guidelines given by the international standards or by the provisions of the Classification Societies. These approximate formulas are taking into account the wake field which, unfortunately, can be only estimated unless experimental towing tank tests are carried out. Another possibility is the numerical evaluation with different Computational Fluid Dynamics (CFD) codes. However, CFD methods are not always easy to be used requiring an accurate description of the hull forms in the aft part of the ship. The present research underlines these aspects during the preliminary prediction of propeller induced vibrations for a double-ended passenger-car ferry propelled by two azimuth fixed pitch thrusters placed at both ends of the ship. The evaluation of the global forced vibration is performed considering the 3D global Finite Element (FE) model, with NX Nastran for Windows. Based on the presented results, the paper provides reliable information to be used during the preliminary design stages.

1. Introduction
One of the major problems related to the comfort on board of ships as well as to the structural integrity of the hull is the impact of the vibrations induced by the propeller [1]. In fact, the level of vibrations due to the induced pressure fluctuations on the ship hull is strongly affected by the aft geometry of the ship which influences the flow characterized by the wake field. In the absence of experimental tests, providing the most reliable results, there are different approximate formulations to be considered for such kind of evaluations. The pressure field is the effect of the rotation of propeller blades, producing forces which could induce vibrations at natural frequency and higher order ones. The exciting frequencies are dependent on the shaft rotation (rpm) multiplied by the number of blades and higher harmonics being multiple of the number of blades [2]. Moreover, an important contribution to the pressure could be induced by the volume of cavitation developed on the propeller blade. This contribution is strongly linked to the wake field because its non-uniformity has a major impact on the generation of exciting pressures.

The evaluation of the propeller induced surface force excitations are of paramount importance and their evaluation is mandatory. Unfortunately, mainly in the early design stages, due to the lack of information, the evaluation is based on some empirical methods which have been developed during the time, using the results of the towing tank tests and full scale measurements. These formulations have to take into account the maximum value of the Taylor wake fraction and the mean effective full scale Taylor wake fraction in the propeller disk [3], [4]. This is practically introducing another
variable, the wake field, which is not accurately known in the early design stages and has to be evaluated using empirical formulations, which are specific for different types of ships, based on the statistics of previous measurements.

Consequently, it can be stated that there are four main ways which can be followed in order to evaluate the pressure fluctuations due to the operating propeller behind the ship:

- The utilization of the empirical formulas for the pressure impulses evaluations by using approximate values of the wake field based on empirical formulas and different recommendations;
- The utilization of the empirical formulas for the pressure impulses evaluations and experimental towing tank measurements;
- Calculation of pressure impulses using CFD applications based on the numerical integration of the incompressible Reynolds Averaged Navier – Stokes equations;
- Direct measurements of the pressure impulses by means of tests performed in the cavitation tunnels on the so-called dummy models fitted with pressure transducers.

It has to be noticed that the most reliable way to evaluate the fluctuating forces with high degree of accuracy is based on hydrodynamic tests on scaled models but these are costly and require access to experimental facilities like towing tanks and cavitation tunnels.

Having as main purpose the evaluation of the level of global forced vibration and the identification of the possible vibration problems of the ship, the study presents the results obtained based on a dynamic linear analysis performed by NX Nastran for Windows using FEMAP [5]. Because of the fact that the classification societies give no information related to the finesse of the mesh when dynamic analysis is required, the calculation of the vibration level is carried out for a number of three different sizes of the mesh. With the aim of highlighting the influence of the method to be followed for the evaluation of pressure fluctuations due to the operating propeller behind the ship, both empirical and numerical results are presented.

2. Structural analysis based on FE method

In order to achieve the proposed objective – the global forced vibration assessment - the present study covers two major phases: the mode shapes together with natural frequencies evaluation and the impact of the forced vibrations. Based on the results of the first stage one may find solutions to avoid the resonance phenomenon. Mention should be made that the present results have been obtained after a number of preliminary investigations when different variants of local reinforcements were considered.

For the FE analysis, the hull structure is considered based on a 3D global FE model (figure 1) made of plate and bar elements as follows: plate elements for decks, sides, bottom, double bottom, longitudinal and transverse bulkheads and primary member webs and bar elements for secondary stiffeners and face plates of primary members.

![Figure 1. 3D FE model.](image1)

![Figure 2. 3D FE model detail for the observation deck.](image2)

Three different mesh sizes have been used for the deckhouse (figure 2) in order to evaluate the vibration level, as follows:
- mesh size of one frame spacing in longitudinal and transversal direction (figure 3);
- mesh size of ½ of frame spacing in longitudinal and transversal direction (figure 4);
- mesh size of ¼ of frame spacing in longitudinal and transversal direction (figure 5);

**Figure 3.** Mesh size of one frame spacing.  
**Figure 4.** Mesh size of half frame spacing.  
**Figure 5.** Mesh size of one quarter frame spacing.

### 2.1. Mode shapes and natural frequencies

At this stage of the study, two types of masses were considered in the 3D FE model: the ship masses (steel, equipment, cargo) and the hydrodynamic added masses. The hydrodynamic added masses were calculated according to reference [6] and were introduced in the model together with the ship masses, using the density correction. The theoretical aspect of the hull girder mode shapes is presented in figure 6.

In order to take into consideration the influences of the blade frequencies, a range of frequencies between 1 Hz and 15 Hz was selected for the calculations. Figure 7 illustrates the natural frequencies of the ship for a range between 1 Hz and 15 Hz, only.

**Figure 6.** Hull girder vertical of 2-5 nodes.  
**Figure 7.** Modal frequency between 1-15 Hz.

The relevant global mode shapes of the bending modes and torsional mode of the ship are shown in figure 8 to figure 11.

**Figure 8.** The first (1st B) bending mode shape of the hull.  
**Figure 9.** The second (2nd B) bending mode shape of the hull.
2.2. Forced vibrations

The forced vibrations have as sources the pressure impulses induced by the operating propeller as well as the external forces and moments due to the main engines [7]. Being provided with special dampers that are modelled as springs elements, the main engines are not inducing relevant vibrations. Their weight is represented by mass type elements placed in the centre of gravity.

Taking into account that the Roll-on/Roll-off (Ro-Ro) double-ended vessel presents an asymmetric superstructure and is propelled by two azimuth fixed pitch thrusters placed at each end of the ship, both travelling directions and several power allocations of the propellers (50% - 50%; 60% - 40%; 70% - 30%, 85% - 15% and 100% - 0%) were considered during the research program. The present study use the optimum going ahead power allocation defined by 70% aft - 30% fore, as it was found during towing tank self-propulsion tests using a scaled model, the modelling scale being \( \lambda = 12.049 \). The corresponding revolutions are 251 rpm – for the aft propeller and 218 rpm – for the fore propeller.

The selected case is presented in figure 12. The areas where the pressure impulses are acting were identified based on a CFD analysis. The most unfavourable cases, i.e. the most exposed areas where the highest values are expected, are confirmed by the experience accumulated referring to the flow visualisation during experimental tests in cavitation tunnels. It has to be underlined that, for a more accurate evaluation, pressure impulses tests or more sophisticated CFD analysis should be carried out.

According to reference [1], the damping which is considered a combination between the structural contribution, the hydrodynamic contribution, and the lack of cargo etc., is in the range of 2% - 4%.

The forced vibrations evaluation was carried out using the 1st blade harmonics.

Based on [6], the propeller induced pressure, \( p_c \), is a combination between two main components given by the cavitating pressure \( p_c \) and the non-cavitating pressure \( p_0 \), respectively:
The non-cavitating and cavitating pressures were calculated with the empirical formula given by [6]:

\[ p_z = (p_0^2 + p_C^2)^{1/2} \]  \hfill (1)

\[ p_0 = \left( \frac{(ND)^2}{70} \right) \frac{1}{Z^{1.5}} \left( \frac{K_0}{d/R} \right) \]  \hfill (2)

\[ p_C = \left( \frac{(ND)^2}{160} \right) \frac{V_S (w_{r_{\text{max}}}-w_e)}{\sqrt{(h_a+10.4) \cdot d/R}} \left( \frac{K_C}{d/R} \right) \]  \hfill (3)

The main input data which was used in order to calculate the pressures are:

- \( N \) – propeller revolution [m],
- \( D \) – propeller diameter [m];
- \( V_S \) – ship speed [m/s];
- \( Z \) – blade number [-];
- \( R \) – propeller radius [m];
- \( d \) – distance from \( 0.9R \) to a position on the submerged hull when the blade is at the Top Dead Center position [m];
- \( w_{r_{\text{max}}} \) – maximum value of Taylor wake fraction in the propeller disc [-];
- \( w_e \) – mean effective full scale Taylor wake fraction [-];
- \( h_a \) – depth of shaft centerline [m].

### 3. Results

The comparative analysis of the propeller induced pressure using the wake fraction calculated based on above mentioned formula (*) and the experimental one is presented in table 1. The pressure impulses corresponding to the aft-fore power allocation are presented in table 2. The different Classification Societies provide information related to the acceptable vibration levels for human reaction [6] and guidelines for preventions [8], [9], [10]. The results are presented in table 3.

### Table 1. Propeller induced pressure.

| \( w_{r_{\text{max}}} \) =0.55* | \( w_{r_{\text{max}}} \) =0.12 | \( w_{r_{\text{max}}} \) =0.12 | CFD |
|---|---|---|---|
| Pressure aft [Pa] | 6162 | 3149 | 3149 | 2940 |
| Pressure Difference [%] | 49 % | 7% | | |

### Table 2. Pressure impulses for 70% aft - 30% for power allocation.

| Location | \( N \) [rpm] | \( p_0 \) [kPa] | \( p_C \) [kPa] | \( p_z \) [kPa] | propeller frequency [Hz] | 1-st blade frequency [Hz] |
|---|---|---|---|---|---|---|
| Fore | 218 | 2.08 | 4.15 | 4.64 | 3.63 | 14.5 |
| Aft | 251 | 2.76 | 5.50 | 6.16 | 4.18 | 16.7 |
Table 3. Maximum vibration level for passengers and crew spaces (human reaction).

|                        | ISO 6954: 1984 | ISO 6954:2000 |
|------------------------|----------------|----------------|
|                        | Peak velocity  | Frequency weighted |
| (5–100 Hz), mm/s       | (1–80 Hz) velocity mm/s |
| Passengers’ spaces     |                |                |
| Passenger cabin Luxury  | 2.5            | 2.1            |
| Passenger cabin Standard| 4.0            | 2.4            |
| Public spaces          | 4.0            | 3.0            |
| Open recreation decks  | 5.0            | 3.5            |
| Crew spaces            |                |                |
| Accommodation and navigation spaces | 5.0 | 3.5        |
| Work spaces            | 6.0            | 5.0            |

Comparative results based on the analyses of the velocity amplitude in z direction for the three selected meshes and for the both empirical and experimental wake fraction are graphically presented in figure 13 to figure 18. The same power allocation, 70% aft - 30% fore, was considered.

**Figure 13.** The first bending mode. Velocity amplitude in z direction for $w_{r_{max}} = 0.55$.

**Figure 14.** The first bending mode. Velocity amplitude in z direction for $w_{r_{max}} = 0.12$.

**Figure 15.** Fore (rpm 218). The first blade frequency. Velocity amplitude in z direction for $w_{r_{max}} = 0.55$.

**Figure 16.** Fore (rpm 218). The first bending mode. Velocity amplitude in z direction for $w_{r_{max}} = 0.12$. 
4. Conclusions

The present study highlights the fact that, for a dynamic FE analysis, a very fine mesh does not bring a significant improvement in terms of the accuracy. The values presented in table 4 refer to the influences of the fineness of the mesh on the amplitudes of the velocities, expressed in percentages. It was observed that the differences between the results for the three types of mesh is about 5% while the difference between the course mesh and the fine mesh is up to 10%. It can be concluded that the differences are within reasonable limits and for large models the utilization of a too fine mesh could lead to an exaggerate computer time consumption. Consequently, a mesh size of ½ of frame spacing in longitudinal direction (figure 4), will lead to an enough accurate evaluation and reasonable computational time allocation. Mention should be made that, during the preliminary design stages, when a multitude of variants involving significant modifications have to be considered in a very limited period of time, the computer time consumption become an important issue.

Using the graphical representation, see figures 13 to figure 18, it can be observed that a peak value closed to natural frequency of 1.89 Hz can be identified. It has to be underlined that only the velocity amplitudes corresponding to a wake Taylor fraction \( w_{\tau_{\text{max}}} = 0.12 \) do not lead to values which are over the limits provided by the rules of the classification societies. Thus, being not a critical value will not affect the comfort on board and will not create problems from the global and local structural point of view as compared to the case referring to the observation deck.

Another important aspect is related to the accuracy of the wake values to be considered for the evaluations of the pressures impulses forces induced by the propeller. The paper clearly underlines that the evaluation using empirical formulations could lead to large differences as compared to those provided by the experimental measurements (see table 1). Consequently, the results will be significantly influenced by the exaggerate pressure impulses forces, inducing unrealistic results which could strongly influence the design accuracy. Taking into account that experimental tests on scaled models are not always easy to be carried out due to costs and time, a deeper investigation of the CFD availability and performances have to be considered.

Table 4. Mesh size influences.

| Mesh size  | \( w_{\tau_{\text{max}}} = 0.55 \) | \( w_{\tau_{\text{max}}} = 0.12 \) |
|------------|---------------------------------|---------------------------------|
| 1/1        | 7.6                             | 3.9                             |
| 1/2        | 7.2                             | 3.7                             |
| 1/4        | 7.1                             | 3.6                             |
| Value amplitude [mm/s] | 5%                              | 5%                              |
| Difference [%]   | 1%                              | 3%                              |
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