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LOCAL SHIP VIBRATIONS – A METHODOLOGY FOR THEIR EVALUATION AND CONTROL FROM THE EARLY DESIGN PHASE

Ottavio Pinto - Cesare Tarditi

FINCANTIERI ,Naval Vessel Business Unit
Hydrodynamic and Acoustic Department
Via Cipro 11 , 16129 Genoa , Italy , fax +39-10-5995379,
e-mail: Ottavio.Pinto@fincantieri.it
e-mail: Cesare.Tarditi@fincantieri.it

Abstract

Local Vibrations represent one of the most critical parameters for the comfort evaluation on board of the ship. Normally a limit value of vibration is defined in the ship specification and the shipyard has the responsibility to assure that the specified values are not exceeded.
Depending mainly on the extension, vibration phenomena could be divided in Global Hull vibrations and local vibrations. Hull vibrations interest all the ship as a beam in its natural modes; local vibrations interest only parts of the ship.
Local and Global vibrations could be studied and controlled modelling the ship with FEM 3D model in an advanced design phase, but Global and Local Vibrations could also be investigated in an early design phase when the ship is not yet modelled, with simplified models.
The paper presents a methodology, developed by Fincantieri and experimentally tested on previous constructions, for the risk evaluation of local vibration exceeding the limit curve due to the main machinery excitations.
Starting on the assumed machinery excitation spectra and depending on the their distance from main ship spaces, on the characteristics of the elastic connections, and on the type of structures, an evaluation of the deck vibrations amplitude is obtained.
On the base of these data, when a risk of excess is noted, a deeper analysis on the evaluation of the deck natural frequency could be made in order to investigate more in detail the risk of resonance and in case of coincidence between the main machinery excitation and the natural frequency, structural countermeasures shall be taken.

1. INTRODUCTION

Vibration in ship structures is a major concern for those who design and operate vessels.

Excessive vibrations aboard ship can result in fatigue failure of structural members or machinery components, can adversely affect the efficiency of operating crews, and can result in discomfort or annoyance to passengers and crew.
The following design trends contributed to this:
• Light-weight construction and, therefore, low values of structural stiffness and mass;
• Arrangement of living and working spaces in the vicinity of the propeller and main engines;
• Increases in propulsion machinery power, to achieve high service speed, resulted in higher levels of vibration excitation.
Another factor, not related to ship design, but one that has brought the study of vibration to the forefront, has been a general increase in the awareness of and concern for the environment in which people live and work.
A ship is subject to self-generated dynamic forces of a periodic nature from propulsion and auxiliary machines, as well as to serious transient forces generated by random seas. Each force can excite different vibration phenomena, with different extension and frequency.
The definition “ship vibrations” normally identify elastic oscillations of all the ship’s hull and/or of its parts.
The frequency range of these oscillation overlap from 0 to about 0.5÷1 Hz the ship motions due to sea actions, from 20 Hz up to 20000 Hz the airborne noise, and from about 1 Hz to about 80 Hz the range that mainly affect human comfort and efficiency.
The transitions between ship motions, ship vibrations and ship acoustics are smooth. In the field of vibration, depending on the extension and on the frequency, vibration phenomena, could be divided in Global Hull vibrations and Local vibrations.

Global vibrations interest all the hull structures as a free-free beam (both ends free) when subjected to dynamic loads. Global vibrations mainly are in the range from 1 Hz up to 10 Hz. They could be studied and controlled modelling the ship with FEM 3D model and avoiding coincidences between the natural hull modes and the exciting forces. In order to have a preliminary estimation of these natural frequencies some experimental simplified laws or in alternative simplified FEM model could be used with the few data at disposal also in the preliminary design. In this case due to the low frequency investigated, the model could be not so detailed and accurate.
Typical large substructures, such as the aft part of the ship, the deckhouse and the double bottom, are coupled in a way that they cannot be considered isolated.
The ship structure, the propeller and the main propulsion machinery primarily establish the global vibration characteristics of the ship. After the ship is built, modifications to correct excessive global vibrations resulting from improper design are generally most expensive and unpractical. In addition, vibration of the hull girder will excite major substructures, local structural elements, and shipboard equipment.

Local Vibrations interest local structure that may be identified as panels, plates, girders, bulkheads, minor equipment foundations, etc. and components of larger structures (major substructures) or of the hull girder. The natural vibration characteristics of local structures can be regarded, for the sake of simplicity, as being independent of the vibration behaviour of the structure surrounding them. Local vibrations are mainly in the range from 10 up to 100 Hz, which also include the range where human sensitivity is higher.
It is also for this reason that most problems encountered aboard ship occur in local structural elements; they are the result of either strong inputs received from the parent structure, amplified by resonance effects in the local structure, or are the response to vibratory forces generated by mechanical equipments attached to the local structure.
Respect to the global vibrations, in this case, if a FEM model is applied for predictions, it must be very accurate and therefore not applicable in a preliminary design. Related to local

vibrations an other more simply and at the same time enough accurate method could be used since the preliminary design in order to verify the correct dimensioning of stiffeners and plate thickness in correspondence of the evaluated machinery excitation force.
Excluding the relations that there are between vibration and structural/machinery reliability, ship motions, and noise, this paper is addressed to present a methodology for the evaluation, since the preliminary design, of the potential local vibration exceeding the acceptance criteria due to the machinery excitation

Criteria related to the allowed local vibrations, are mainly connected to the exposure to the vibration respect to the comfort, efficiency and health on board.
Internationally, the standard ISO 6954 (edition 1984) gained general acceptance for the evaluation of ship vibrations, especially in relation to human exposure. An important feature of this standard was that, for the purposes of assessment, peak values of amplitudes had to be considered individually for each excitation frequency.
Even if the periodic excitation forces are subject to a certain degree of variation, leading to the well-known problem with the “crest factor”, the standard allows quick and easy experimental assessment, and calculation predictions.
New ISO 6954 standard was released in December 2000, harmonised with the principles of ISO 2631-1, and for the assessment, a single value over the frequency range from 1 to 80 Hz is formed, basing on the frequency-weighting curve of ISO 2631-2.
Even if it better reflects the human sensitivity, the new standard looses the easy possibility of utilization either for measurements or for prediction purposes.
In recent years, it has become standard practice to regulate vibration aspects for a new shipbuilding in the contractual ship documents. Basing on international standards, or on different criteria, limit values of vibration for new ship constructions are agreed with ship-owners and imposed in the ship technical specification. It is shipyard responsibility to assure that the limit values are not exceeded in the spaces of interest.
In case of excess, actions, if any, must be taken.
It must be considered that all the actions when the ship is completed or in advanced phase of constructions could be very difficult to apply and time-cost expensive.

2. PROCEDURE FOR LOCAL VIBRATION CALCULATION

It is, therefore, interest of the shipyard to perform, from the preliminary design phase, a proper analysis in order to investigate the possible risk of high local vibration and, after that, to take all the necessary actions possibly in advance to reduce the risk of excesses at acceptable levels.
Theoretically, in order to avoid high vibrations it could be necessary to prevent resonance conditions. For a complex structure like a ship, in all the normal operational conditions, these requirements could not be guaranteed in all the conditions.
On the other hand, when resonance conditions are verified the amplification due to the resonance depends not only on the safety margins between excitation frequency and natural frequency but also on the damping coefficient of the system, that is usually very difficult to estimate, and shall be assumed (as other parameters, at least in a preliminary phase) with proper safety factors based on experience.
The following procedure experimentally tested was defined in order to find the possible local vibration problems in selected positions related to the vibration originated from the main machinery.
The proposed method could be considered an application of a wider method successfully tested, used for airborne noise evaluations in all the ships spaces. This methodology is

originated from theoretical considerations with coefficient that are obtained experimentally on previous constructions. The formulation considers, from a logical sequence point of view: the vibration acceleration levels measured, or estimated, on the source, the behaviour of mounts and of the machinery foundations, the losses due to structural intersection, the decay of the vibration along the propagation path between source and the considered position where the local vibration shall be estimated.
The mathematical description of the above steps allows to calculate the vibration level, on the position of interest, with the following formula:

$$Lv = 20 \cdot log\left[\left({M \cdot a_f}/{n} \right) \cdot {T_m}/{K_f} \cdot \omega \right] - \sum_{i=1}^{n} m_i \cdot \eta_i - \sum_{s=1}^{n} n_s \cdot I_s + \Delta_{Lv} + K_1 + 10 \cdot log\, n \qquad [1]$$

In which:

- Lv = vibration levels in the point of interest referred to 10^{-5} m/s^2
- M = weight machinery in Kg
- a_f = acceleration spectra of the source m/s^2
- n = number of connections between machinery and foundation
- T_m = Mounts force Transmissibility between machinery and foundation
- K_f = foundation stiffness (N/m)
- $\omega = 2\pi f$
- f= frequency (Hz)
- m_i = partial vibration propagation path between source and receiver (m)
- η_i = damping of the different partial vibration propagation path (dB/m)
- n_s = numbers of structural intersection of the same type along the vibration path
- I_s = losses due to each kind of structural intersection
- ΔLv = additional vibration decay (ex. acoustic treatment)
- K_1 = constant that depends on the reference considered

The formula can be applied several times for all the main machinery, and the vibration levels can be automatically computed in all the investigated area considering all the machinery contributions.
In this first step of calculation it can be assumed that at each frequency of the excitation range, in the selected position (as example a deck), corresponds a natural frequency with a damping coefficient equal to 0.05. That corresponds to an increase of the vibration level of about 20 dB in all frequency range of machinery excitation respect to the real value in absence of resonance. This is a conservative, but realistic, law obtained from experimental measurement taken on board in correspondence of resonance.
The procedure could be utilised in a preliminary phase considering the acceptable structure-borne levels expected for the different sources. In case the first calculation run shows risks of high vibrations, deeper calculations are necessary; after factory tests on main equipments, when the data measured on the different sources are known, the calculation could be tuned more precisely. Possible problems due to the considered machinery are in this way highlighted and countermeasures could be properly taken.
In case the above approach puts in evidence a risk of excess in local vibration levels, a deeper analysis on the evaluation of the local structure natural frequency could be made in order to investigate more in detail the risk of resonance.

3. LOCAL STRUCTURE NATURAL FREQUENCIES CALCULATION

The evaluation of the natural frequency of the local structure of interest represents the second step of the proposed procedure.
This evaluation could be made with analytical approximated formulas. Considering, as example, the local structure of a deck, it is necessary to identify all possible rectangular portion of deck where the stiffeners are regularly distributed in that area, with the hypothesis of simply supported edge conditions, and uniformly distributed mass; the area is limited where pillars or vertical bulkheads cannot allow displacement in the normal direction to the deck. Typically this approach is used only for the determination of the first mode of panel vibration. Appropriate effective added mass must be considered in this calculation; indicatively values can vary from 50 to 100 kg/m^2 on decks, depending on the kind of areas.
There are a number of other parameters that can influence natural frequencies of local structures, such as curvature of the structures, residual stresses of welds or distortions, real added mass and edge conditions; it is therefore necessary to take care on the assumptions in order to be on the safe side for the design, but in the meantime it is necessary to apply all the experience to avoid unrealistic, useless, too much conservative predictions.
Once calculated the natural frequency in this way, it must be verified that this frequency is at least 20% more than the frequency of the excitation forces. In case contrary corrective actions must be studied and, if considered necessary to reduce the risk, also applied.

4. CALCULATION EXAMPLE

From previous experience on fast ferries, it was noted a critical area for vibration in the aft part of the main passenger deck, above the engines and (in some cases, when dedicated to passenger area) above the water jets.
It was therefore decided to apply this procedure for the local vibration calculation, since the beginning of the design of latest fast ferry of recent construction.
Fig. 1 shows the results of this procedure applied to the aft passenger area of the main deck, just above the two aft propulsion diesels. For this construction each Propulsion Engine has at maximum 1000 rpm corresponding to an exciting frequency of 16.6 Hz. In the figure, the contribution of the propulsion engines (that were demonstrated from calculation to be the most critical vibration source in that area) to the local vibration levels in different ship areas is shown.

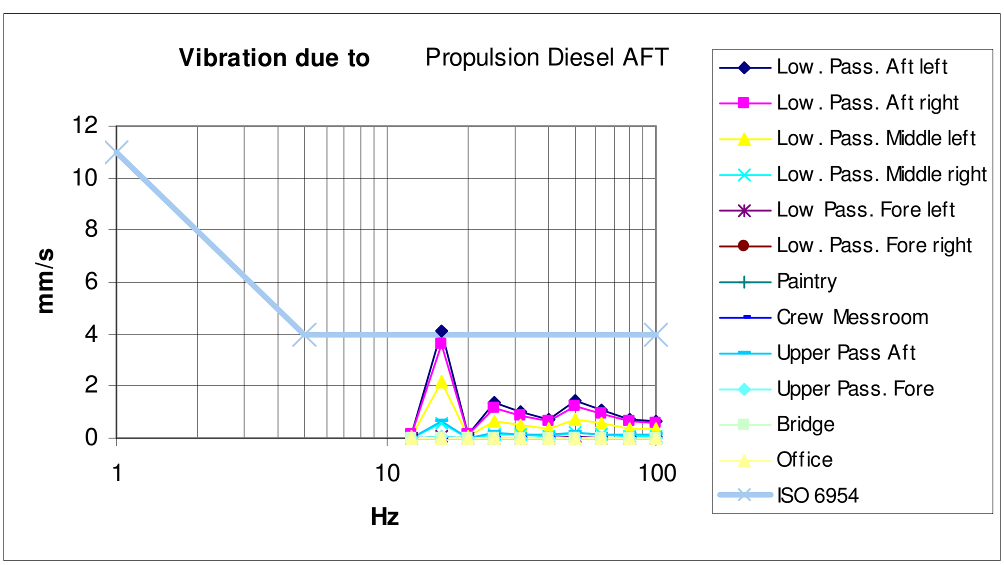


Fig. 1

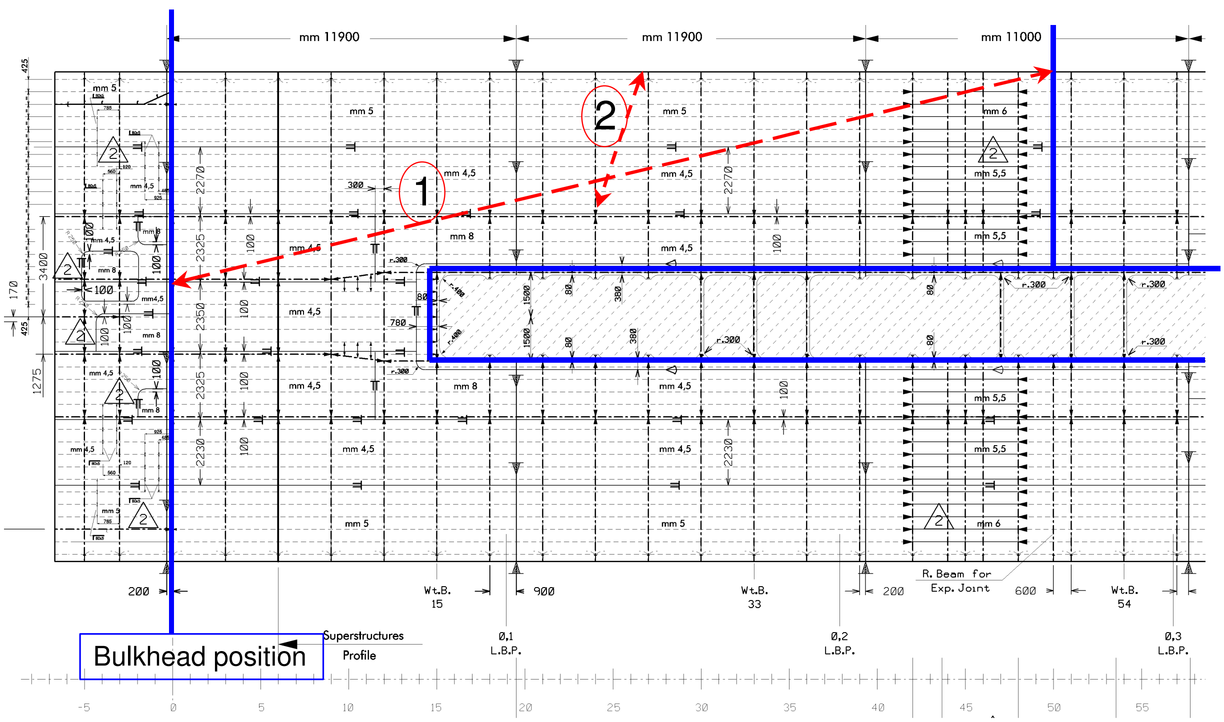
As it can be seen, the estimated vibrations levels (in resonance condition if it happens) on the deck in aft passenger areas are close to the maximum level admissible due to the induced vibration of the propulsion engines at the maximum speed.

The first run of calculation was made assuming the structure borne noise levels guaranteed from the engine supplier as maximum levels in the different operating conditions. After that, the levels directly measured on the engine during the Factory acceptance test were assumed (Fig. 1 refers to this condition).

The spectrum levels obtained in this way were compared with the reference levels coming from the ISO 6954 issue 1984 in all the frequency range.

Having noted (from Fig.1) the risk of vibration excesses in case of resonance, as second step of the procedure it was necessary to make an evaluation of the possible local natural frequencies on above identified critical aft passenger areas.

Basing on the structure configurations, two typical panels were identified and calculated; panels and results of natural frequency calculations are in Fig.2.



1 CALCULATED RESONANT FREQUENCY = 16.2 Hz

2 CALCULATED RESONANT FREQUENCY = 42.4 Hz

Fig. 2

It is clear that in this case there is quite a coincidence between the Propulsion Diesel excitation (at the maximum power, that is about 16.6 Hz) and the natural frequency of the deck, and therefore there is a strong possibility of resonance condition; considering all the possible uncertainty reasons, it was considered relevant the risk of high vibrations in all the high speed (up to the maximum) operating range.

It was therefore decided to identify possible corrective actions.

5. ACTIONS TO BE CONSIDERED

Both the expected vibration levels in resonance condition, and the expected quite coincidence between the natural frequency of the deck and the excitation frequency, induced to consider critical the risk of high vibration. However, considering the strong weight constraints that there are on a fast ferry, and also the fact that the expected levels are just at the limit value but not significantly above, it was decided to identify possible actions to reduce the risk of high vibrations, but it was also decided to apply, if necessary, the solution on board only after the results of measurements during the ship sea trials.
These corrective actions could mainly involve the stiffness of the deck, the edge conditions, and the mounting conditions of the exciting machinery. All the considered actions require an increase of weight.
In particular, the following countermeasures were estimated:
Increase of stiffeners dimension below the passenger deck in order to move the natural frequency from about 16 Hz to at least 19.5 Hz. The possible solution considered was an increase of the existing 350*8*220*14 mm T main ribs with new ones of 450mm height with consequently more stiffness and more weight.
Variation of the mounts stiffness. Theoretically a reduction of mount's stiffness reduces the value of the vibration on the deck. In practice this way could not be followed due to the fact that mounts were already selected as soft as possible in order to reduce the noise on board.
Increase of damping along the propagation path of vibration. In general this solution can be followed to reduce noise/vibration levels, even if it could be extremely cost and weight consuming to apply efficient treatments on all propagation paths. In the case studied, due to the low frequency involved, this action was evaluated no effective.
Increase of foundation stiffness. An increase of foundation stiffness can reduce the vibrations transmitted on the deck but this solution is not easy to apply on board with engine room plan completed.
The results taken on board of the vessel, in the points A and B (aft passenger area of main deck) as in Fig. 3 during the at sea acceptance test, confirmed the expected presence of a significant vibration level at 16 Hz (fig. 4) but with a value lower than the limit.

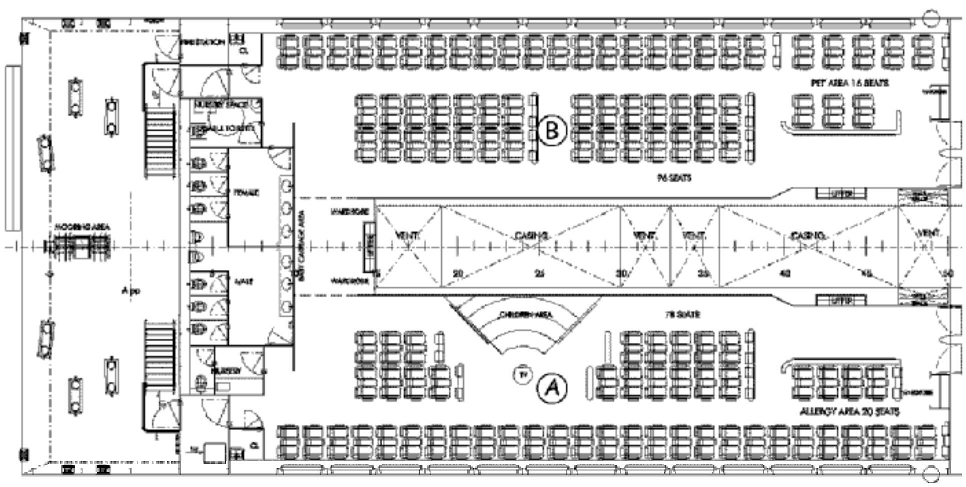


Fig. 3

The vessel was then delivered without any remarks or adverse comment, and without necessity of actions to reduce the vibration levels.

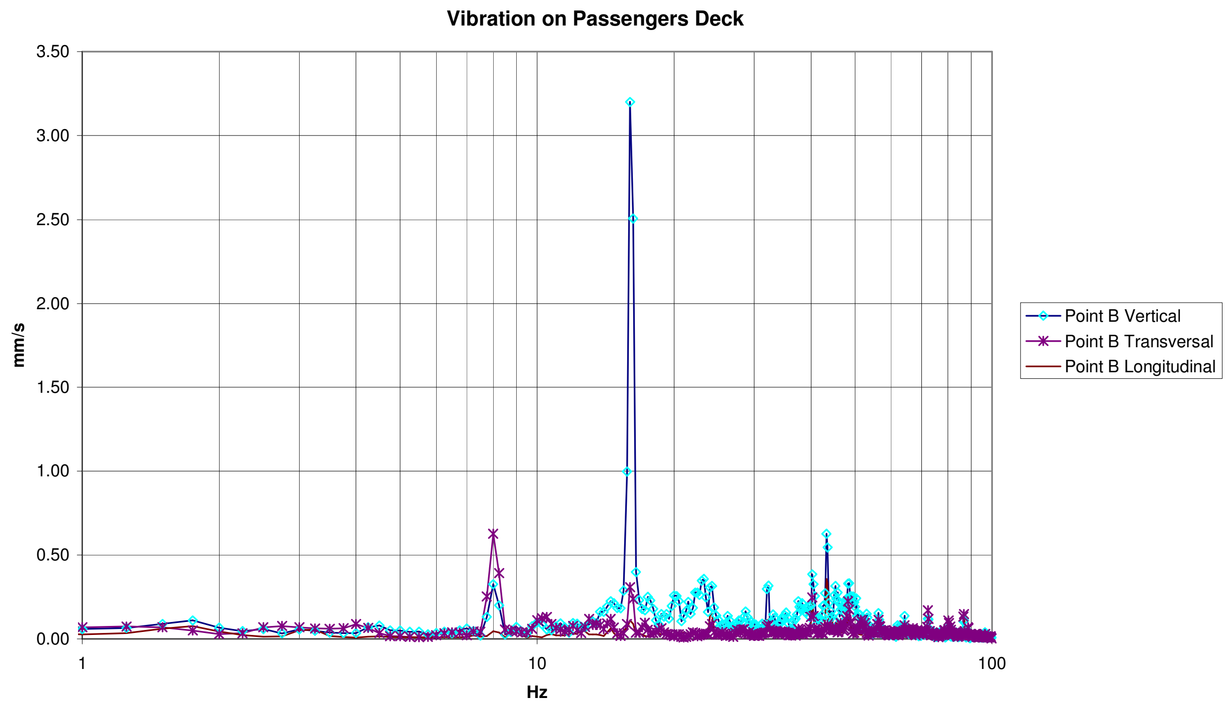


Fig. 4

6. CONCLUSIONS

A methodology experimentally tested for the risk evaluation of local vibrations exceeding the limit curve due to the main machinery excitations has been presented. The results of a test case confirm the necessity and the useful of the local vibrations estimation model in order to evaluate, from the early design phase, the risk of high vibrations and the impact of all the possible actions in terms of effectiveness, increase of cost and weight.

7. REFERENCE

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