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Ship vibration and noise: Some topical aspects

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In 1975 he joined Lloyd's Register, first in the Technical Investigation Department where he served for nine years in general engineering analysis and troubleshooting roles and, subsequently, in the Advanced Engineering and Performance Technology Departments. In 1984 he became Deputy Head of the Advanced Engineering Department and in 1987 was appointed in the same role in the Performance Technology Department, In 1992 he became Head of the Technical Investigation Department. Then in 2003 he became the Senior Principal Surveyor for Marine Technology and Investigation and now has global responsibility for these matters within Lloyd's Register. In his present role he also sits on several international technical committees and is author of the book Marine Propellers and Propulsion.

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Synopsis

The paper considers some topical aspects of noise and vibration in the shipboard environment. In particular, hydrodynamic and other sources of excitation are discussed in relation to their quantification, analysis and the assessment of the ship's response. Within the discussion comfort class rules are considered in relation to their ISO basis and application in the marine industry. Some comment is made concerning noise in relation to the external shipboard environment.

1 Introduction

As ship design advances, particularly with regard to structural optimisation and high speeds to meet market demands, there is a tendency for noise and vibration problems to become more pronounced. In some instances this manifests itself in the difficulty of achieving a satisfactory solution within the operational and design constraints of the ship, while in other cases progress has brought other excitation sources to the fore.

Design practice should include elements of model testing, calculation and heuristic deduction from previous experience. While in many instances all three elements are included in the design process in some cases the model testing phase is omitted, frequently on the grounds of cost. Where significant power in the context of a particular hull form or a restriction on draught has to be accommodated, then cavitation testing of the propeller in a properly scaled wake field and measurement of the associated propeller induced hull surface pressures is particularly valuable in terms of stabilising cavitation and minimising excitation. Notwithstanding the desirability of undertaking the three basic elements of design, it must be appreciated that error bounds are associated with each of these elements and, at best, they are a guide for achieving a good design together with sound engineering judgement. In the case of cavitation testing it is the scale effects between model and full scale which may be complex; for computation it is the accuracy of the mathematical model and for heuristic deduction it is the closeness of previous designs and operating conditions to the subject ship under consideration.

The attainment of an optimised shipboard environment is not simply about the setting of criteria. Rather it is the placing of the criteria in the context of the whole design process such that they are realistic and achievable within the bounds of current knowledge and engineering practice. Furthermore, these criteria should not restrict innovation and development, but should embrace flexibility for those who wish to market an enhanced product, albeit at additional cost, or to include emerging technologies. Nevertheless, in all cases the assessment criteria should seek to embrace the total effects of noise and vibration rather than address partial aspects.

2 Vibration and Noise

There are a number of sources of vibration and noise present in a ship or marine vehicle. Typically these may include:

- The prime movers - typically diesel engines.
- Shaft-line dynamics
- Propeller radiated pressures and bearing forces.
- Air conditioning systems.
- Manoeuvring devices such as transverse propulsion units
- Cargo handling and mooring machinery.
- Vortex shedding mechanisms
- Intakes and exhausts.
- Slamming phenomena.

Much has been written about many of these topics, for example [1, 2, 3, 4, 5], and there is little reason to rehearse those discussions further. However, recently a number of relatively new problems associated with vortex shedding and after-body slamming have been experienced and merit consideration as does the more recent understanding of propeller induced vibration signatures.

2.1 Propeller Radiated Signatures

The excitation from machinery is frequently, but not invariably, harmonic in content. As such, signatures from these items tend to more closely follow the mathematical basis for conventionally used analysis techniques: typically Fourier`analysis. Propeller generated signatures however, most commonly produce time series signatures with significant cyclic perturbations.

The basis of the development of the propeller induced hull pressure signature is the acceleration of the cavity volumes with respect to time on the propeller blades, modified by the self induced component of pressure generation arising from the vibration of the ship structure at the point of interest. As such, the hydrodynamic excitation process is a time domain event whose physical processes can better understood through the pressure time series.

In experimental studies the pressure time signature is most commonly analysed using a Fourier based technique, due largely to the need to relate excitation sources to ship hull and structural response characteristics. Fourier techniques, which were originally developed as a curve fitting process, have as their underlying tenet the requirement of piecewise continuity of the function that is being analysed; whether this is over a long or short time frame. Given that this condition is satisfied then, assuming a sufficient number of terms are taken in the series and the numerical stability of the algorithm is acceptable then the method will satisfactorily curve fit the function as a sum of transcendental functions whose coefficients may then be input into finite element or other computational processes.

To gain a phenomenological understanding of cavitation behaviour sufficient to affect a proper cure to a technical problem rather more than a Fourier-based curve fitting algorithm is necessary. This is for two reasons: first, a set of coefficients of transcendental functions tell little about the structure of the underlying cavitation causing the problem and secondly, and perhaps more importantly, cavitation based signatures are rarely uniform with respect to time. There are blade surface pressure changes which vary from blade to blade in a single revolution and changes from one revolution to the next. These changes are random in nature and result from the interaction of the temporal changes in the flow; the flow field, this being the sum of the steady inflow field and the seaway induced velocities; and the blade to blade geometric variations due to the manufacturing tolerances of the propeller blades. These changes influence both the general form of the cavity volume variation and the higher frequencies and noise generated from the random perturbations of the topological form of the underlying cavity structure.

If a phenomenological approach is adopted for the analysis of propeller induced hull pressure signatures so as to develop a solution to a practical problem and minimise sea trial or dry docking down time, then other analytical approaches are required [6]. A number of candidate approaches offer themselves and among these are Short Form Fourier Transforms, Joint Time-frequency analysis, wavelet techniques and a double integral analysis of the underlying pressure signature. Experience has shown that each of these methods has shortcomings due for the most part to the near adiabatic collapse of the cavity volumes in adverse wake gradients. Nevertheless, wavelet methods and the double integral technique have been shown to have some advantages when considering different aspects of the problem. In the case of the wavelets most of Lloyd's Register's present work has focussed on standard applications of Daubechies formulations which have allowed some progress to be made. Further discrimination is believed to be possible if purpose designed wavelet forms are used to describe different cavity phenomena.

Notwithstanding the wavelet class of methods, the double integral approach has been shown to be the most successful at phenomenological discrimination. The pressure integration approach is essentially a time domain process, which together with visual observations of cavitation can link the dynamics of visual events with the dynamics of pressure pulses. It is clear from both ship and model scale analysis of such data that the more severe excitation events are generated by cavitation which grows, collapses and rebounds in a small cylindrical sector of the propeller disc and slipstream which spans the wake peak. It is the passage of the propeller blades through this slow speed region which causes the flare-up and collapse of cavity volumes on the blade and in the tip vortex shed by the advancing blade.

2.2 Vortex Shedding Mechanisms

Vibration induced from the flow over sea chest openings has been a troublesome feature in some ships and has prevented the meeting of localised comfort criteria. Such vibrations, which commonly manifest themselves in local structural resonant behaviour, are clearly not directly related to machinery rotational speeds. Rather, they are related to vortex shedding over the sea chest hull opening grills and, therefore, are Strouhal and Froude number dependent based on ship speed.

Other examples of vortex shedding induced vibrations have recently been encountered. These have included A-brackets, extended centre-line skegs and fin appendages fitted to ships to improve course keeping stability. The characteristics of these problems were high vibration levels in the ship structure or failure of the structural elements.

Vortex shedding occurs when the fluid flow around the after part of an appendage is separated from the structure at a given Reynolds number and the oscillating pressures cause the elastic structure to vibrate. The shedding frequency is given in terms of Strouhal number and for bodies with rough surfaces at ship scale it is frequently acceptable for estimation purposes to use a value for the Strouhal number of 0.2. When structures vibrate in the transverse direction with a frequency at or near the vortex shedding frequency they tend to increase the strength of the shed vorticity which, in turn, may increase the structural excitation. Furthermore, if the vortex shedding frequency is close to the natural frequency of the structure it will move to the frequency of the structure. Then once the vortex shedding frequency is synchronised with the frequency of the structure it will often tend to remain at that frequency even when the flow speed changes over a limited range.

The following two examples illustrate this type of problem. In the first case two fins were installed aft of the rudder to improve the ship's course keeping and the ship was propelled by a four blade controllable pitch propeller. Installation of the fins resulted in increased vibration levels and subsequent failure of the fins. It was concluded that the fin's first response to vortex excitation started when the vortex shedding became coincident with the fin's natural frequency of 22 Hz in a bending-torsion mode. As a result of the resonance, the fin's vibration increased. The vortex frequency did not change with the ship speed but remained locked to the fin natural frequency. The fins' next vibration mode of 25.4 Hz, a torsional mode, was excited when the vortex shedding frequency became coincident with that frequency. At the ship's maximum operating speed the only remaining frequency component was that of 25.4 Hz. The high stresses resulted in the fin failures.

In the second case the centreline skeg was extended to improve the ship's directional stability. This ship's propulsion system comprised two thruster units employing two contra-rotating fixed pitch propellers on each unit: the propellers blade numbers being four and five. The skeg extension resulted in excessive vibrations in the ship structure and the investigation showed that the skeg's natural frequency at 18.3 Hz was excited by the five blade propeller at around seven knots. As a result the skeg's vibration amplitudes increased initiating synchronisation of vortex shedding and skeg natural frequencies. Higher up the speed range when the four bladed propeller excitation become coincident with the skeg natural frequency, the vibration levels reached their first peak amplitude at 9.5 knots. A second vibration peak was then measured between ten and eleven knots when the vortex shedding frequency became coincident with the skeg natural frequency.

The dynamic behaviour of structures subjected to vortex shedding excitation depends upon the ship speed, the structural profile and its trailing edge shape, the structural natural frequencies and damping and the interaction between the fluid flow and structural vibrations. Reduction of the vibration amplitudes of the structure caused by vortex shedding may be achieved by:

- Avoidance of resonance between the vortex-induced excitations and the structural natural frequency.
- Lowering the vortex excitation levels.
- Reducing response of the structure.

Resonance can be avoided by modifying either the vortex excitation frequency or the structural natural frequency. Ordinarily the structural natural frequency should be increased sufficiently to avoid resonances with vortex shedding mechanisms. That may be achieved by increasing the structure's stiffness or changing the aspect ratio. Other solutions can be to increase the vortex shedding excitation frequency by changing the structure's trailing edge shape. In all cases it is necessary to evaluate the structural natural frequencies and ensure that they are not coincident with the vortex shedding and propeller excitations.

2.3 After-body Slamming

Shock impacts such as slamming also need consideration since as well as generating structural tertiary stresses in the ship structure, these events can be disturbing to passengers. In particular after-body slamming can excite resonant conditions in the ship structure [7]; most typically the 2-node vertical mode. The incidence of after-body slamming, in contrast to fore-body slamming, frequently reduces with increasing ship speed. This is because the ship's entrained wave system increases at higher speed and gives a measure of protection to the hull after-body from the otherwise uninterrupted incidence of the environmental wave system. In addition to being a function of reducing ship speed, the slamming threshold speed is also dependent on the sea state, recognising that the resultant sea state comprises both underlying swell and wind induced wave components which strongly influence the directional slamming threshold. Furthermore, a common characteristic possessed by ships that suffer from after-body slamming is a relatively flat after-body design coupled with relatively small immersion. In this latter context after-body slamming has been known to occur in sea conditions with wave heights less than 1m. Consequently, the exploration at an early design stage of hull forms that avoid this problem in association with the predicted sea and ship motions is of particular importance.

3 Habitability Criteria

Following the introduction, in 1984, of the ISO 6954 criteria of assessment for shipboard vibration, Lloyd's Register and others have accumulated a considerable body of knowledge and background information on the interpretation of that standard for all ship types. Clearly, that experience should not be discarded since vibration level assessment is a subjective measure which varies from person to person for a variety of reasons; typically due to age, state of health, sex, personal susceptibility and so on. The 1984 standard assessed, on an individual basis, the discrete harmonic responses in a measured vibration spectrum between 5 Hz and 100 Hz, such that none should rise above 4 mm/s in order to remain within the *No Complaints Expected* zone. There was then a grey area over the same frequency range until a level of 9 mm/s was reached when the standard signalled *Complaints Expected*. However, within this framework any number of harmonic components could rise to the limiting lower level and little guidance was given on the combination of vibration components in different orthogonal directions. Again, the way in which the human body responds to and experiences these vibration components varies considerably, both in terms of forced responses or the resonances of bodily organs.

There are significant differences between the current 2000 version of the ISO 6954 Code and its predecessor dating from 1984. Furthermore, there has been a slow take-up of the ISO 6954 2000 in the marine industry, in part, due to the uncertainties in its correlation with the accumulated data built up by the industry in the application of the earlier standard. ISO 6954 1984 applied to crew areas of merchant ships only but was adapted to passenger areas for various classification society comfort rules. Vibration was measured as maximum repetitive peak value by applying a conversion factor Cf to the peak values which was then compared to the ranges defining acceptability. Although measuring vibration data to this standard was relatively straightforward, the major difficulty and cause of disagreement between the parties was in the application of the conversion factor Cf. Cf is equivalent to a crest factor and if not determined by measurement, by finding the ratio of the true peak to the r.m.s value, should be assumed to have a value of 1.8. However, due to the influence of different vibration sources the vibration signature varies around the ship and it could be difficult to measure the crest factor at every measurement location.

In contrast the ISO 6954 2000 version of the code applies to both passenger and crew areas. The vibration signature is measured as an overall frequency weighted r.m.s. acceleration or velocity over the frequency range 1 to 80 Hz. In this way the currently applicable version of the code attempts to consider a single vibration value which characterises the total vibration signature in the part of the ship under consideration, rather than addressing a collection of single frequency components whose individual values may well meet an acceptability criterion but whose combined effect grossly exceed the acceptable value. The difficulty in applying the new code frequently lies in the definition of the acceptable value to be placed in a new building contract between a ship owner and a builder. In order to provide some guidance in this respect an analysis of measured time series vibration signatures for five ships has been carried out using both the 1984 and 2000 versions of the ISO Code. The ship types included four passenger ships, two ferries and two cruse ships, together with an LNG ship: the sizes of these ships varied between 104 m to 320 m length between perpendiculars. For this analysis a constant value of Cf of 1.8 was applied to all of the data. Figure 1 shows the relationship between the two assessment criteria for various locations within the five ships.

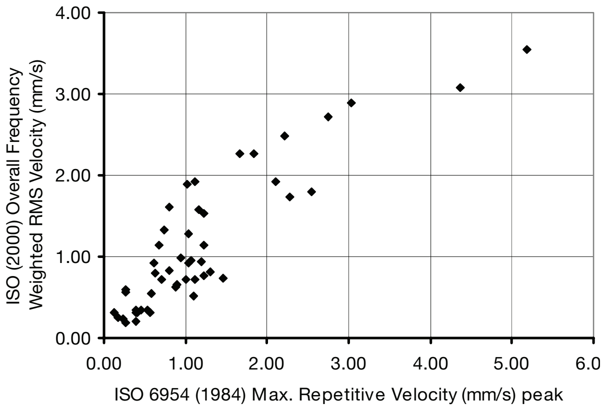

Figure 1: Comparison of Shipboard Measurements Analysed to the ISO 1984 and 2000 Codes.

In order to explore the relationship between the two standards further a number of theoretical waveforms were analysed using both the ISO 6954 1984 and 2000 versions. The results for the ISO 6954 1984 analysis are presented as peak values, however, according to the standard these values should be multiplied by 1.8 in order to represent the maximum repetitive value which further accentuates the differences. Table 1 shows the results of this analysis.

Table 1 Signal Comparison According to ISO 6954 (1984) and (2000)

Signal Form	Amplitude (peak) - Frequency	ISO 6954 1984 Peak	ISO 6954 2000 r.m.s
Sine Wave	2.2 mm/s - 31.5 Hz	2.20	1.57
Sine Wave	2.0 mm/s - 7 Hz + 4.0 mm/s - 10 Hz	4.00	2.72
Square Wave	2.0 mm/s - 6 Hz	2.41	1.77
Square Wave	2.0 mm/s - 10 Hz	2.19	1.72
White Noise	Std.Dev. 100 mm/s	5.70	59.4

In the case of the noise signature the amplitude of noise signals does not vary in a periodic and, therefore, predictable, manner. Therefore, in order to describe the amplitudes adequately it is necessary to introduce the concept of amplitude density. As such, instead of being able to predict an instantaneous amplitude it is only possible to assign a probability that the random signal will fall within a certain amplitude range. A Normal (Gaussian) amplitude density curve was used to model the random processes encountered in practice. Consequently, in Table 1 the standard deviation is quoted about a zero mean amplitude.

Noise, like vibration, is a subjective human response which varies significantly between individuals. In the case of passenger accommodation, those who join ships for vacation purposes are to a large extent conditioned in their expectation by experiences ashore. As such, these land environments largely form the basis of their expectation of a standard for cruising and ferry transits. Clearly, this expectation is difficult to achieve and in some cases is unrealistic, not least because the ship is a floating and moving platform whereas, for the most part, in land environments building foundations are perceptibly rigid.

In the case of crew accommodation, where it is becoming accepted that in order to attract seafarers to the profession an increasing level of shipboard comfort needs to be achieved. This is true for both noise, vibration and internal climate and although maximum noise levels are prescribed for the crew working and rest spaces by IMO such an equivalent prescription is not generally the case for vibration. With regard to the passenger environment IMO does not prescribe limits. This is left to the owners to define based on their perceptions of the market they serve, their passengers' expectation and the associated costs of implementing vibration and noise attenuation. Within this context owners and builders frequently find it convenient to use classification society standards as an objective set of criteria upon which to base a contract. Moreover, if so desired by the owner, these standards may then act as through-life criteria of assessment for the ship so as to maintain the passenger and crew environment for as long as the particular market need exists.

The expectation and human tolerance of noise in different transport media is very different between aeroplanes; land-based transport such as cars, buses and trains; and ships. Indeed, the acceptable noise level is a function of variables like exposure time and the individuals' perception of the transport media. For passenger ships people's expectations perhaps focus on hotels as being the nearest equivalent. From the analysis of full scale trials [8] it is seen that the lower end of the cruise ship luxury cabin noise band meets the higher end of the noise range for the better hotel room market. With regard to passenger sleeping areas, acceptable limits are reasonably easy to define and these can be grouped for convenience into superior and standard classes for a number of ship categories recognising the logarithmic nature of the dB scale. Indeed, the broad range of noise level bands for passenger accommodation is seen in the Lloyd's Register's Passenger and Crew Accommodation Comfort (PCAC) Provisional Rules and experience has shown these levels to be reasonably effective. The situation is similar for noise levels in the crew accommodation and working spaces where the IMO Code [9] forms the underlying basis for the criteria upon which the PCAC Notation is built. Within this context when relatively low levels of sound are to be met in ships, the mutual interference between cabins then becomes an important consideration and the minimum sound insulation index [8, 10] between adjacent cabins needs careful attention.

However, the question of public spaces is not so easy to address; either with respect to enclosed or open deck areas. In the case of enclosed spaces it is reasonable to expect that shopping areas and libraries, for example, may need a different treatment from say general open internal spaces and restaurants. In the former case of the shopping area a more lenient noise limitation may be appropriate, depending upon the purchasing atmosphere to be created, whereas, in the case of a library a more contemplative atmosphere may be sought. For discotheque, theatre and dance floor areas the specification of operational limits is considered inappropriate since the internal noise levels are determined by artistic and market needs with, hopefully, some influence of physiological criteria and, therefore, effective control by PCAC-type criteria in service is not possible. Notwithstanding this, a theatre, in contrast to a discotheque, may require periods of extreme quiet in order to emphasise some dramatic moment. This, however, is seen as a matter for an owner to specify since the need for lectures or dramatic theatrical moments, in contrast to variety shows, is ship specific depending on the sector of the market that is being addressed. What is of importance in relation to comfort rules is the ability to contain whatever noise is being created in one of these public arenas so that it does not interfere with adjacent areas, or indeed other areas remote from the source. The correct prescription of minimum air-borne insulation indices for materials used in bulkheads and decks is therefore seen as essential. In addition, where dancing or gymnastic activity is destined to take place in a public space or, alternatively, walking on decks covered in hard materials such as wood or marble, then the weighted normalised impact sound pressure level (Ln,w) [11] requires special consideration.

4 Environmental Influences of Noise

Noise may have important external consequences for both airborne and waterborne transmissions. In the case of problems associated with airborne noise, for example, residents living near an island cruise ship terminal complained about the noise from one particular ship. Noise measurements were taken at a number of locations in the residential area, both when the ship was and was not present at its berth. The overall A-weighted sound pressure levels were considered acceptable when the ship was in-port but analysis of the noise signals showed a distinct peak at 37.5 Hz. This resulted in a tonal characteristic to the noise signature and was the feature which the residents found annoying. In that particular case the source of the tonal component was identified as gas pulses from the four diesel generator engines exciting the funnel and the frequency of the pulses corresponded to the firing frequency of the engines. Such a case, together with many others involving noise emissions from cargo handling or working operations, serves to underline the importance minimising the noise emissions from ships.

In the context of seaborne noise, leaving to one side the stealth considerations associated with warships and submarines, a growing international interest is that of its possible effects on marine life. These concerns embrace interference with marine mammal navigation as well as the general marine life communications, both of mammals and other sea life. In the case of marine mammals such as whales, dolphins and porpoises they have internal ear structures very similar to those of humans and a reasonable amount is known about the their oral communication frequency ranges in comparison to other marine forms of life. This is because mammals by comparison to fish and other life forms are relatively easy to train and work with. The extent, however, of the frequency range of the mammals hearing requires resolution and, research on an international basis, is progressing in this area.

5 Conclusions

This paper has considered a number of aspects relating to the noise and vibration of ships. In particular the analysis of propeller radiated pressure signatures acting on the hull surface has been explored as has aspects of after body slamming and excitation from vortex shedding sources.

A correlation exercise between the ISO 6954 1984 and 2000 vibration assessment criteria and based on a number of vibration measurements derived from five ship sea trials and analysed wave forms has been presented.

Some comment has also been made on the problems associated with noise and its acceptability.

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