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Maritime Sustainability and Maritime Labour Convention - Reducing Vibration and Noise Levels On board Ships for Health and Safety of Seafarers

Authors: Raymond Chia*, Dr Ivan Tam** and Dr Arun Kr. Dev**
*Vroon Offshore Services Pte Ltd, Singapore
**School of Marine Science and Technology (MaST), Newcastle University (Singapore)

ABSTRACT

The Maritime Labour Convention (MLC) addresses the employment standards of seafarers with the aim of improving social sustainability in the areas of fair wages, contractual terms, working and living conditions as well as their health and safety on board ships. This paper presents the requirements of MLC regarding noise and vibration mainly in the accommodation spaces, and the possible solutions to minimise them to meet the new demands. By reducing vibration and noise levels on board ships, seafarers would be able to enjoy better rest and thereby improve their health and safety. It in turns also reduces human factors related accidents and improves productivity.

1. INTRODUCTION

On 27 July 2012, the General Assembly of the United Nations (UN) adopted a resolution on the renewal of their commitment to sustainable development and to ensuring the promotion of an economically, socially and environmentally sustainable future for present and future generations (UN, 2012).

It was followed up by the International Maritime Organization (IMO) with a document titled “A Concept of a Sustainable Maritime Transportation System” for the 2013 World Maritime Day. In this document, the IMO argued that A Sustainable Maritime Transportation System will need the collaboration of shore-side actors, from both industry and Governments, in the due implementation of the Maritime Labour Convention (MLC) for the protection and provision of care for seafarers, in order to ensure that social integrity does not become eroded and that qualified, professional sailors have an attractive work environment (IMO, 2013).

The International Chamber of Shipping (ICS), which is an international trade association for the shipping industry, with representation in the IMO, also came up with a paper titled “Sustainable Development” for the 2013 World Maritime Day. According to ICS, Shipping is the only industrial
sector to have a comprehensive international framework in place addressing employment standards following the entry into force of the Maritime Labour Convention on 20 August 2013 (ICS, 2013).

The Maritime Labour Convention (MLC) is an instrument adopted on 23rd February 2006 by the International Labour Organization (ILO) in Geneva, Switzerland. It addresses the employment standards of seafarers with the aim of improving social sustainability in the areas of fair wages, contractual terms, working and living conditions as well as their health and safety on board ships (ILO, 2006).

According to ILO, the purpose of MLC Regulation 3.1 is to ensure that seafarers have decent accommodation and recreational facilities on board that are consistent with promoting the seafarers’ health and well-being (ILO, 2006). It can be argued that MLC Regulation 3.1 could be indirectly linked to the safety of a ship as the environmentally catastrophic grounding of the Exxon Valdez in Alaska in 1989 might be due to the fatigue of the deck watch keeping the officer on duty (Hetherington et al., 2006). A study of Australian seafarers done by Parker et al. as cited by Hetherington et al. has also revealed that 70% of seafarers reported poor to very poor sleep (Parker et al., 2002, Hetherington et al., 2006).

The aim of this paper is to examine MLC Regulation 3.1 specifically on the design and performance of new (Post-MLC) ships in particular relation to noise and vibration. This paper is organised into five main sections: Introduction, Review of Post-MLC Noise and Vibration Requirements in Accommodation Spaces, Impact on the Design and Performance of Post-MLC Ships, Possible Solutions, and Conclusion.

2. REVIEW OF POST-MLC NOISE AND VIBRATION REQUIREMENTS IN ACCOMMODATION SPACES

2.1 MLC Regulation 3.1

The Regulations and the provisions of Part A of the MLC are mandatory whereas the provisions of Part B are not mandatory. As per Standard A3.1.6 (h), the “accommodation and recreational and catering facilities shall meet the requirements…on health and safety protection and accident prevention, with respect to preventing the risk of exposure to hazardous levels of noise and vibration…” (ILO, 2006).

An example of the non-mandatory guideline in MLC 2006 for the prevention of noise (Guideline B3.1.12.4) is that the limits for noise levels for working and living spaces should be in conformity
with the ILO international guidelines on exposure levels, including those in the ILO code of practice entitled Ambient factors in the workplace, 2001, and, where applicable, the specific protection recommended by the International Maritime Organization, and with any subsequent amending and supplementary instruments for acceptable noise levels on board ships. An example of the non-mandatory guideline in MLC for the prevention of vibration (Guideline B3.1.12.5) is that no accommodation or recreational or catering facilities should be exposed to excessive vibration (ILO, 2006).

2.2 Code on Noise Levels on Board Ships, 2012 - IMO Resolution MSC.337 (91)

The IMO adopted the Code on Noise Levels on Board Ships (Noise Code 2012) on 30 November 2012 in Resolution MSC.337 (91) (IMO, 2012). The important difference between Noise Code 2012 and Noise Code 1981 is that the former is legally treated as a mandatory instrument under IMO SOLAS regulation II-1/3-12 (IMO, 2014) while the latter was not a mandatory instrument and only provided guidance (IMO, 1981). However, there are also provisions of Noise Code 2012 that remain recommendatory, options for compliance, or informative in nature. It entered into force on 1 July 2014 (IMO, 2012).

Noise Code 2012 is applicable to all new ships of 1,600 GT and over. Unlike Noise Code 1981, the definition of “new ship” is given in the code and it means a ship to which the code applies in accordance with SOLAS regulation II-1/3-12.1. This would mean that it is applicable mandatorily to such ships with keel laid on or after 1 July 2014.

Similar to Noise Code 1981, there are provisions in Noise Code 2012 that the code may also apply to existing ships of 1,600 GT and over, and to new ships of less than 1,600 GT, as far as reasonable and practicable to the satisfaction of the flag administration. However these are non-mandatory provisions. The types of vessels that Noise Code 2012 are not applicable are similar to Noise Code 1981 except with the additions of high-speed craft, pile driving vessels, and dredgers.

Dispensations from certain requirements may be granted by the flag administration in special circumstances, if it is documented that compliance will not be possible despite relevant and reasonable technical noise reduction measures. Such dispensation shall not include cabins, unless exceptional circumstances prevail.

On completion of the construction of a ship, or as soon as practicable thereafter, measurement of noise levels in spaces specified in the code including accommodation spaces shall take place under
sea trial and in port operating conditions and shall be suitably recorded in a survey report. Sound pressure level readings shall be taken in decibels using an A-weighting [dB(A)] and C-weighting [dB(C)] filter, if noise levels exceed 85 dB(A). The limits for noise levels [dB(A)] are specified for the accommodation spaces in Table 1.

Table 1: Limits for noise levels specified for the accommodation spaces.
Source: Code on Noise Levels on Board Ships (IMO, 2012)

<table>
<thead>
<tr>
<th>Designation of Rooms and Spaces</th>
<th>Ship Size</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1,600 up to 10,000 GT</td>
</tr>
<tr>
<td>Cabin and hospitals</td>
<td>60 dB(A)</td>
</tr>
<tr>
<td>Mess rooms</td>
<td>65 dB(A)</td>
</tr>
<tr>
<td>Recreation rooms</td>
<td>65 dB(A)</td>
</tr>
<tr>
<td>Open recreation areas (external recreation areas)</td>
<td>75 dB(A)</td>
</tr>
<tr>
<td>Offices</td>
<td>65 dB(A)</td>
</tr>
</tbody>
</table>

2.3 ABS Guide for Compliance with the ILO Maritime Labour Convention, 2006 Title 3

This ABS guide “is based on the MLC objective requirements and ABS’ interpretation of the intent of the Part A requirements and on what ABS considers satisfactory compliance with the Part A subjective requirements” of the MLC. There is a disclaimer in the guide that states that additional or different criteria by individual flag administrations may also be applicable and that adherence to the guide’s criteria does not guarantee acceptance by any flag administration of compliance with MLC Regulation 3.1. The guide also provides the assessment criteria and measurement methodology for a vessel seeking to obtain a free ABS MLC accommodation related notation (MLC-ACCOM) (ABS, 2014b).

The vibration loads imposed on the body is restricted to motions transmitted from surrounding structures to the entire human body through the feet of a standing person in the frequency range of 1 to 80 Hertz (Hz). Motions transmitted to the body of a seated or reclining person have been omitted from the guide as the motions transmitted through the feet are expected to be the highest vibration levels to which crew will be exposed. Whole-body vibration measurements shall only be taken in manned accommodation spaces. A space is considered “manned” if it is occupied by a seafarer for twenty (20) minutes or longer at a time for regular, routine daily activities.

The maximum vibration levels in Table 2 shall not be exceeded under normal operating conditions. The assessment can be based on acceleration or velocity values. The notation in Table 2 applies to
the vibration levels occurring on the deck supporting the human body in the three translational (x-, y-, and z-) axes. The vibration levels are computed for each axis individually, as well as combined as a multi-axis acceleration value. Each is expressed as a frequency weighted root-mean-square ($a_w$) value. The multi-axis $a_w$ level must be less than or equal to the maximum level expressed in Table 2 to meet the vibration criteria,

Table 2: Maximum Weighted Root-Mean-Square Acceleration Level.
Source: Guide for Compliance with the ILO Maritime Labour Convention, 2006
Title 3 Requirements (ABS, 2014b)

<table>
<thead>
<tr>
<th>Notation</th>
<th>Frequency Range</th>
<th>Overall Frequency Weighted r.m.s. Acceleration</th>
<th>Maximum RMS Level</th>
</tr>
</thead>
<tbody>
<tr>
<td>MLC-ACCOM</td>
<td>1 – 80 Hz</td>
<td>$a_w$ (mm/s²)</td>
<td>Transit Conditions (mm/s²)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>214 (6.0 mm/s)</td>
</tr>
</tbody>
</table>

The maximum acceleration or velocity values for a vessel in transit conditions in Table 2 are similar to the upper values for crew accommodation areas of ISO 6954:2000 as shown in Table 3. However, the maximum acceleration or velocity values for a vessel in dynamic positioning (DP) conditions are set higher. The reason for this is because during DP operations, it is common to use transverse tunnel thrusters in the bow and even the stern, and such thrusters cavitate at relatively low operational speed. The cavitating impeller of the thruster will then induce high vibrations in the surrounding tunnel wall and hull plating (Fischer, 2000b).

Table 3: Overall frequency-weighted r.m.s. values from 1 Hz to 80 Hz given as guidelines for the habitability of crew accommodation areas on a ship. Source: ISO 6954:2000 (ISO, 2000)

<table>
<thead>
<tr>
<th></th>
<th>Crew Accommodation Areas</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>mm/s²</td>
</tr>
<tr>
<td>Values above which adverse comments are probable</td>
<td>214</td>
</tr>
<tr>
<td>Values below which adverse comments are not probable</td>
<td>107</td>
</tr>
</tbody>
</table>

Note: The zone between upper and lower values reflects the shipboard vibration environment commonly experienced and accepted
ABS has separate optional notations for ship habitability (HAB, HAB+, and HAB++) (ABS, 2013). The whole-body vibration values for these notations are shown in Table 3 together with the values for its MLC-ACCOM notation for comparison. The maximum acceleration and velocity values for a vessel with HAB++ in Table 4 are similar with the lower values for crew accommodation areas of ISO 6954:2000 as shown in Table 3.

Table 4: Maximum Weighted Root-Mean-Square Acceleration Level.
Sources: Guide for Compliance with the ILO Maritime Labour Convention, 2006 Title 3 Requirements (ABS, 2014b) and Guide for Crew Habitability on Ships (ABS, 2013)

<table>
<thead>
<tr>
<th>Notation</th>
<th>Frequency Range</th>
<th>Acceleration Measurement</th>
<th>Maximum RMS Level in Accommodation Areas</th>
</tr>
</thead>
<tbody>
<tr>
<td>MLC-ACCOM</td>
<td>1 – 80 Hz</td>
<td>$a_w$ (mm/s²)</td>
<td>214 mm/s² (6.0 mm/s) (During Transit Conditions Only)</td>
</tr>
<tr>
<td>HAB</td>
<td>1 – 80 Hz</td>
<td>$a_w$ (mm/s²)</td>
<td>178 mm/s² (5.0 mm/s)</td>
</tr>
<tr>
<td>HAB+</td>
<td>1 – 80 Hz</td>
<td>$a_w$ (mm/s²)</td>
<td>143 mm/s² (4.0 mm/s)</td>
</tr>
<tr>
<td>HAB++</td>
<td>1 – 80 Hz</td>
<td>$a_w$ (mm/s²)</td>
<td>107 mm/s² (3.0 mm/s)</td>
</tr>
</tbody>
</table>

3. IMPACT ON THE DESIGN AND PERFORMANCE OF POST-MLC SHIPS DUE TO NOISE AND VIBRATION REQUIREMENTS

IMPACT DUE TO POST-MLC NOISE REQUIREMENTS

3.1 Ships below 1,600 GT

Although under Standard A3.1.6 (h), the “accommodation and recreational and catering facilities shall meet the requirements…on health and safety protection and accident prevention, with respect to preventing the risk of exposure to hazardous levels of noise…” but this MLC requirement is not prescriptive and does not specify what levels of noise are harmful or the maximum noise limits for the accommodation spaces.

It is only under MLC Guideline B3.1.12.4, which is non-mandatory that there are references to the limits for noise levels for working and living spaces. One of the references is the specific protection recommended by the International Maritime Organization, which means the Code on Noise Levels on Board Ships (Noise Code). At the time when the MLC was adopted in 2006, the IMO Noise Code was the 1981 version, which was non-mandatory.
The current IMO Noise Code was adopted in 2012 and came into force on 1 July 2014. It is mandatory for vessels that are built after that date and are 1,600 GT and above. Hence there is minimal impact on the design of ships below 1,600 GT due to Post-MLC Noise requirements unless the ship owners decide to voluntarily seek compliance to obtain for example the American Bureau of Shipping (ABS) MLC accommodation related class notation. The ABS MLC class notation’s assessment criterion for Noise on board vessels below 1,600 GT is the same as IMO’s Noise Code 2012 for ships above 1,600 GT to 10,000 GT (ABS, 2014b).

3.2 Ships above 1,600 GT to 10,000 GT

The maximum noise limits of Noise Code 2012 for the accommodation spaces of ships above 1,600 GT to 10,000 GT are the same as Noise Code 1981, and shown in Table 5.

Table 5: Maximum Noise Limits for Accommodation Spaces of Ships above 1,600 GT to 10,000 GT

<table>
<thead>
<tr>
<th>Accommodation Spaces</th>
<th>Maximum Noise Limit dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cabins and hospitals</td>
<td>60</td>
</tr>
<tr>
<td>Mess rooms</td>
<td>65</td>
</tr>
<tr>
<td>Recreation room</td>
<td>65</td>
</tr>
<tr>
<td>Offices</td>
<td>65</td>
</tr>
<tr>
<td>Open recreation areas</td>
<td>75</td>
</tr>
</tbody>
</table>

Ship noise can be divided into airborne noise and structure-borne noise according to the nature of the sound source. The airborne noise spreads through the air, while structure-borne noise spreads through the structure of the vessel.

Noise, emanating from the casing of any mechanical source, intake/exhaust, or from fans, is considered airborne noise. The acoustic source also generates structure-borne noise via the attachment point to its foundation. Received noise in a compartment results when the structure-borne noise radiates from the structure back to the air (ABS, 2014a).

According to the International Association of Classification Societies (IACS), the sources of noise on board a ship include main propulsion system, auxiliary machinery, and the heating, ventilation, and air-conditioning (HVAC) and piping systems. There are three main routes of transmission of ship noise:
- Airborne noise radiated directly to the air by main or auxiliary machinery system;
- Structure-borne noise spread along the hull structure through mechanical vibration and radiated outward;
- Fan noise and airflow noise transmitted through the duct of the ventilation system.

Mechanical vibrations are the largest source of noise on board the ship (IACS, 2013).

The mechanical vibrations from the propulsion system and machinery are transmitted by different types of sound waves, resulting in the structure-borne energy being partially transmitted through obstacles and somewhat dampened by conversion to heat and radiation along the path from the source to the receiver. Thus, the greater the number of obstacles along the noise/vibration path, the lower the structure-borne noise will be at the receiver space. Such barriers include deck/bulkhead intersections, frames, and other non-uniformities in the structure (ABS, 2014a).

The problem for smaller vessels, especially those with high horsepower propulsion systems and thrusters such as Anchor Handling Tug (AHT) vessels, is that the mechanical vibrations are not actually dampened by the mass of the vessel due to their lighter structures and smaller frames. That is why the Noise Code 2012 is only applicable to vessels 1,600 GT and above.

A tunnel thruster cavitates at relatively low operational speed, and the cavitating impeller of the thruster will then induce high vibrations in the surrounding tunnel wall and hull plating. According to Fisher, in a study of offshore support vessels (OSV), compartments two decks above the thruster room, where the mess and recreation rooms were located, have a mean level of 77 dB(A). On the third deck above the thruster room, where some sleeping berths were located, mean levels amounted to 70 dB(A) (Fischer, 2000b). Although the GTs of the OSVs were not mentioned in the above study, as if assuming that it was around 1,600 GT, which is not uncommon, then all these levels stated in the above study were in excess of Noise Code 2012 maximum noise limits of 60 dB(A) for berths and 65 dB(A) for mess or recreation rooms of above 1,600 GT to 10,000 GT ship.

3.3 Ships above 10,000 GT

The maximum noise limits of Noise Code 2012 for the accommodation spaces of ships above 10,000 GT are shown in Table 6.
Table 6: Maximum Noise Limits for Accommodation Spaces of Ships above 10,000 GT


<table>
<thead>
<tr>
<th>Accommodation Spaces</th>
<th>Maximum Noise Limit dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cabins and hospitals</td>
<td>55</td>
</tr>
<tr>
<td>Mess rooms</td>
<td>60</td>
</tr>
<tr>
<td>Recreation room</td>
<td>60</td>
</tr>
<tr>
<td>Offices</td>
<td>60</td>
</tr>
<tr>
<td>Open recreation areas</td>
<td>75</td>
</tr>
</tbody>
</table>

In a paper presented at the Tanker Structure Co-operative Forum 2013 Shipbuilders Meeting, Wu and Li used Statistical Energy Analysis (SEA) method to predict the noise levels of the accommodation spaces on board a new large oil tanker design. Based on the calculations of their SEA model, it was found that predicted noise levels in the hospital, treatment room, and two cabins on the upper deck of the large oil tanker exceeded the Noise Code 2012 maximum noise limits of 55 dB(A) for cabins and hospitals of ships above 10,000 GT. After control measures had been taken in the ship design, the noise levels in the accommodation spaces of the actual ship were measured and found to be able to meet the maximum noise limits of Noise Code 2012 (Wu and Li, 2013).

The above example shows that noise problems do not only occur in lower decks due to their proximity to the engine room and propulsion system but also in the upper deck of the large ship where the proximity to intake and exhaust stacks can also cause noise problems. The example has also shown that noise prediction models are useful tools in ship design and help to detect noise problems before they occur after a ship is built.

**IMPACT DUE TO POST-MLC VIBRATION REQUIREMENTS**

Although under Standard A3.1.6 (h), the “accommodation and recreational and catering facilities shall meet the requirements…on health and safety protection and accident prevention, with respect to preventing the risk of exposure to hazardous levels of vibration...” but this MLC requirement is not prescriptive and does not specify what levels of vibration are hazardous or the maximum vibration limits for the accommodation spaces.

It is also similar under the non-mandatory MLC Guideline B3.1.12.5, which only states that no accommodation or recreational or catering facilities should be exposed to excessive vibration. Without specific MLC requirements or guidelines on vibration, a ship owner or ship designer can only turn to the classification societies for more specific guidance. An example of such guidance is
the ABS Guide for Compliance with the ILO Maritime Labour Convention, 2006 Title 3 Requirements mentioned earlier in this paper. The ABS guide adopts ISO 6954:2000, which is a standard for evaluating the shipboard vibration. This ISO standard is also adopted by other classification societies in their class rules for shipboard vibration prior to MLC. Hence, at this point of time, the impact on ship design due to Post-MLC vibration requirements will be the same as Pre-MLC, but after the year 2000.

3.4 Design Considerations
According to Asmussen et al. (Asmussen et al., 2001), low-cost building and operation aspects of a ship are increasingly influencing the design, resulting in vibration problems happening more often. The following design trends are contributing to the vibration problems:

- Light-weight construction and, therefore, low values of stiffness and mass
- Arrangement of living and working quarters in the vicinity of the propeller and main engine to optimise stowage space or to achieve the largest possible deck openings of container ships
- High propulsion power to achieve high service speed
- Small tip clearance of the propeller to increase efficiency by having a large propeller diameter
- Use of fuel-efficient slow-running main engines

Excitation; Stiffness; Frequency Ratio; and Damping are the four elements of importance to consider with regards to vibration in ship design (ABS, 2006) and any of the followings will contribute to vibration reduction (Yucel and Arpaci, 2013):

i) Reduce exciting force amplitude, $F$

The propeller induces pressure fluctuations on the aft of the hull due to the moving blades, resulting in vertical vibration of the hull girder in response to the propeller excitation (force). This propeller-induced ship vibration may be reduced by changing the unsteady propeller hydrodynamics. The following formulae proposed by Holden et al. as cited by Yucel and Arpaci (Yucel and Arpaci, 2013, Holden et al., 1980) are used to illustrate how this can be done:

For non-cavitating pressure,

$$P_0 = \frac{(ND)^2}{70} \frac{1}{2^{1.5}} \left( \frac{K_0}{d/R} \right)$$  \hspace{1cm} (1)
For cavitating pressure,

$$P_c = \frac{(ND)^2}{160} \frac{V_s(w_{T_{\text{max}}} - w_e)}{\sqrt{h_a + 10.4}} \left( \frac{K_c}{d / R} \right)$$ \hspace{1cm} (2)$$

The total pressure impulse, which combines both the cavitating and the non-cavitating components, is then calculated from:

$$P_z = \sqrt{P_o^2 + P_c^2}$$ \hspace{1cm} (3)$$

Where $P_o =$ Non-cavitating pressure; $P_c =$ Cavitating pressure; $P_z =$ Total pressure impulse;

$N =$ propeller rpm; $D =$ propeller diameter, in m; $V_s =$ ship speed, in m/s; $z =$ blade number

$R =$ propeller radius, in m; $h_a =$ depth of shaft centreline, in m; $d =$ distance from $0.9R$ to a position on the submerged hull when the blade is at the T.D.C. (Top Dead Centre) position (m)

$w_{T_{\text{max}}} =$ maximum value of Taylor wake fraction in the propeller disc

$w_e =$ mean effective full scale Taylor wake fraction

$K_o =$ $1.8 + 0.4(d/R)$ for $d/R \leq 2$

$K_c =$ $1.7 + 0.7(d/R)$ for $d/R < 1$

$K_c =$ $1.0$ for $d/R > 1$

Looking at the above formulae, both $P_o$ and $P_c$ can be reduced if the propeller RPM and diameter are reduced. The trade-off to such reductions will be lower propeller thrust.

ii) Increase the stiffness, $K$

Stiffness is defined as spring force per unit deflection. In general, stiffness is to be increased rather than decreased when variations in natural frequency are to be accomplished by variations in stiffness. It is not a recommended practice to reduce system stiffness in attempts to reduce vibration. The following formula for the un-damped natural frequency of a simple mass-spring system will illustrate this:

$$\omega_n = \frac{K}{\sqrt{m}}$$ \hspace{1cm} (4)$$

where $\omega_n =$ angular natural frequency in radians per second; $K =$ stiffness of the spring in N/m; and $m =$ mass of the spring in kg

From the above equation, the un-damped natural frequency $\omega_n$ will increase when stiffness, $K$ is increased.
iii) Avoid resonance
Avoid values of frequency ratio near unity, that is $\omega/\omega_n = 1$, which is the resonant condition. At resonance, the excitation is opposed only by damping. $\omega/\omega_n$ can be varied by either excitation frequency $\omega$ or natural frequency $\omega_n$. $\omega$ can be changed by the RPM of a relevant rotating machinery source, or, in the case of propeller-induced vibration, by changing the propeller RPM or its number of blades. $\omega_n$ can be changed by system mass and/or stiffness. Increasing the stiffness is the usual and preferred approach.

iv) Increase damping
Damping of structural systems in ships is small. Therefore, except very near resonance, the vibratory amplitude is approximately damping independent. Furthermore, damping is difficult to increase significantly in systems such as ships; $\zeta$ is, in general, the least effective of the four parameters available to the designer for implementing changes in ship vibratory characteristics. The formula of the damping ratio, $\zeta$ of the mass spring damper model is used to illustrate this:

$$\zeta = \frac{c}{2\sqrt{Km}}$$

where $c =$ damping coefficient in Ns/m; $K =$ stiffness of the spring in N/m; $m =$ mass of the spring in kg; and $\zeta =$ damping ratio

Typical values for damping ratio, $\zeta$ are 0.005 to 0.01 for steel, and 0.05 to 0.10 for rubber (Lamancusa, 2002). The damped natural frequency, $\omega_d$ is related to the un-damped natural frequency, $\omega_n$ by the following formula:

$$\omega_d = \omega_n \sqrt{1 - \zeta^2}$$

From the above formula, it can be seen that the damped natural frequency is less than the un-damped natural frequency, but the difference is negligible as the damping ratio, $\zeta$ is usually relatively small as mentioned above.

3.5 Calculations of Natural Frequencies
As seen from the previous section, being able to derive the natural frequencies of the ship’s structural systems will be very useful to prevent resonance. The transitions between ship motions, ship vibrations and ship acoustics are smooth. In the field of vibration, it is possible to distinguish
between three different phenomena: global hull vibrations, vibrations of substructures and local vibrations (Asmussen et al., 2001).

The most accurate but time-consuming method for the calculations of natural frequencies of the ship’s structural systems is to build a Finite Element Analysis (FEA), model. The FEA model can be used for the global hull structures, substructures, as well as the local structures. Substructures and local structures also need to be considered in the vibration analysis process as they have their natural frequencies, which again should not coincide with primary excitation frequencies (ABS, 2014a). There is a strong interaction between local vibrations of structures and ship’s acoustics. This relationship is manifested by the fact that a ship whose local structures have been consistently designed on vibration also gains acoustic advantages (Asmussen et al., 2001).

3.6 Vibration Prediction in the Design of Vessels

Finite Element Analysis (FEA) also known as Finite Element Model (FEM) can be used for vibration prediction. Using the FEM and a double hull shaped ship of 18,000 deadweight (DWT) chemical tanker, the results of propeller-induced vibration analysis done by Yucel and Arpaci have shown that the vibration velocities of crew accommodation areas remain under the limits prescribed by ISO 6954:2000 (Yucel and Arpaci, 2013).

Another example is in the building of two RO-RO vessels "José María Entrecanales"(C-509) and "Super-Fast Baleares" (C-510) at the Puerto Real shipyard of Navantia. The FEM was applied to avoid dynamic amplification phenomena due to possible resonance in the vessel structure; to identify natural frequencies and prevent their coincidence with the primary excitation sources of the ship: forces and moments of the main engines excitation and forces coming from the pressure pulses induced by the propeller on the sternpost. The dynamic FEM model helped the shipbuilder obtained the "Expected Vibration Levels" in the different areas throughout the vessel structure. Based on the comparison between the "Expected Vibration Levels" with the limits required in the ISO Standard 6954:2000 demanded in the shipbuilding specifications, decisions were taken. It then allowed the shipbuilder to validate the structure from a dynamic behaviour point of view or introduce structural modifications that guaranteed compliance within the required limits (Beltrán, 2011).
4. POSSIBLE SOLUTIONS TO MEET POST-MLC NOISE AND VIBRATION REQUIREMENTS

4.1 Acoustical Design Practices

It is important to incorporate proper acoustical practices during the initial stage of a ship design. Such methods include using relatively quiet machinery and propulsors, optimal general arrangements, and placing compartments with less stringent noise criteria closer to major noise sources (ABS, 2014a). Both IMO and IACS also suggest these practices.

IMO recommends that accommodation should be sited both horizontally and vertically as far away as is practicable from sources of noise such as propellers and propulsion machinery and where applicable to the separation of accommodation spaces from machinery spaces by empty spaces, sanitary and washing rooms. Machinery casings, where practicable, should be arranged outside superstructures and deckhouses containing accommodation spaces. If this is not feasible, passageways should be organised between the casings and accommodation areas, if practicable (IMO, 2012).

IACS proposes that sources of noise such as engines, fans, rotating equipment, to the extent possible, should be isolated and located away from work and living spaces. To reduce noise transmitted to accommodation cabins, the crew accommodations areas should be arranged in the middle or rear of the superstructure or on the poop deck and above. Equipment that by its design or quality are lower noise and vibration should be selected instead (IACS, 2013).

Although some of the above useful acoustical practices may seem simple, they are effective in controlling noise and should be relatively lower in cost to implement compared to noise treatments. Hence they should not be ignored during the initial ship design and should be incorporated as much as feasible.

4.2 Noise Predictions in Ship Design

The use of noise prediction programs in ship design is not a novelty as it was already mentioned by Nilsson in his 1978 paper that the purpose of a noise prediction program is to make it possible at the design stage to estimate the noise levels in a ship (Nilsson, 1978). The only difference is that with more computing power available nowadays, noise prediction programs are using three-dimensional (3D) noise mapping models compared to previously used two-dimensional (2D) models.
IACS recommends that noise modelling is used in initial ship design stage and that the models should include the following components:

- Source, acoustic path, and receiver space description
- Machinery source reports (e.g., noise and vibration levels, size and mass, location, and foundation parameters)
- Propulsor source description (e.g., number of propellers (impellers), number of blades, RPM, clearance between hull and tips of propeller, vessel design speed)
- HVAC source description (e.g., fan parameters (flow rate, power, and pressure), duct parameter, louvre geometry, and receiver room sound absorption quality)
- Essential parameters for sound path description include hull structure sizes and materials, (damping) loss factors, insulation and joiner panel parameters
- Receiver space modelling characterised by the hull structure forming the compartment of interest, insulation/coatings, and joiner panels (IACS, 2013).

A hybrid SEA method was used by Fischer to carry out noise predictions for a supply boat with two bow thrusters. The program accurately accounts for the path losses and acoustic characteristics of the ship's structure and compartments and the results as seen in Table 7 show that thruster induced noise can be accurately predicted (Fischer, 2000b).

<table>
<thead>
<tr>
<th>Compartment</th>
<th>Calculated, dB(A)</th>
<th>Measured, dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Room</td>
<td>108</td>
<td>109</td>
</tr>
<tr>
<td>Mess/Galley</td>
<td>87</td>
<td>87</td>
</tr>
<tr>
<td>Berth on the Main Deck</td>
<td>83</td>
<td>82</td>
</tr>
<tr>
<td>Berth on the 01 Level</td>
<td>75</td>
<td>75</td>
</tr>
</tbody>
</table>

According to the IMO, noise predictions and any noise reduction measures planned in the design phase should be documented. Particularly in cases where, according to the noise predictions, it must be expected that compliance with any of the noise level limits of Noise Code 2012 will be difficult to achieve, despite reasonable technical initiatives (IMO, 2012).

From the above, it can be seen that noise modelling and predictions are very useful tools for solving potential noise problems and also meeting Post-MLC noise requirements, especially during the initial
design stage. Hence the ship designer or yard should not disregard their usefulness and should be prepared to use them.

4.3 Evaluation and Selection of Noise Treatments

After the noise predictions and analysis have been carried out during the initial ship design stage, and if the criteria such as the noise level limits of Noise Code 2012 are not met, then noise treatments need to be considered. Even if noise criteria is met at the initial ship design stage, it is very likely that further noise predictions will be carried out in the later stages of ship design, and at these later stages, noise treatments may be found necessary. The choice of optimal noise treatment should be based on accurate noise predictions and evaluated from a practicality point of view. As there is a broad range of noise treatments available, they are grouped under the following sections of the source, path and receiver treatments.

4.3.1 Source Treatments

As mentioned earlier, the main sources of noise on board a ship include propulsion system and machinery (IACS, 2013).

4.3.1.1 Propulsion System Treatment

The cavitating impeller of a thruster induces high vibrations in the surrounding tunnel wall and hull plating (Fischer, 2000b). Similarly, cavitating propellers generate relatively high sound pressure levels in the water around their blades. This high sound pressure induces vibration on the hull, while part of the acoustic energy may travel through the hub and shafts as well. A propulsor can be considered a source of a pure structure-borne sound like hull excitation (ABS, 2014a).

The most effective way to treat propulsor noise is through careful design, and the goal should be to increase the cavitation inception speed, which is the rotation rate at which cavitation begins. Measures such as increasing number of blades and increasing propeller diameter thereby reducing the needed rpm can help in increasing the cavitation inception speed (ABS, 2014a). However, a larger propeller diameter may mean a smaller tip clearance of the propeller, and this can cause vibration problems (Asmussen et al., 2001). Hence, a rule-of-thumb for tip-hull and leading edge-strut/rudder clearance is a minimum 25% of the propeller diameter.

The vibration of hull structure around propulsors may also be reduced by using a damping coating or by increasing hull plate mass and stiffness (ABS, 2014a). Noise and vibration can also be reduced
by the isolation of the thruster from hull structure and reduced cavitation using permanent magnet tunnel thrusters (RR, 2017b).

Increasing impeller pitch, which may produce the same thrust but at a lower rotation rate, which is equivalent to a lower tip speed; and making use of skewed blades, which may relieve the tip overloading and provide a more uniform loading over the entire blade are some of the noise abatement techniques suggested by Fischer. Fischer also suggested that equipment suppliers can supply Prairie type masker air injection systems with their thrusters. The air is injected near the tunnel inlet or the impeller through perforated air emitter pipes located on the perimeter of the duct. This air would be pulled into the thruster, mix with the cavitation, resulting in a gaseous cavitation-ventilation mixture. This mixture would cushion the bubble collapse and thereby reduce the induced vibration and structure-borne noise. The additional air would also provide some absorption in the tunnel (Fischer, 2000b).

Other propulsion system noise treatments will be discussed under the vibration control procedures of the next paper on vibration as there is a close relationship between the vibrations of structures and the ship’s acoustics (Asmussen et al., 2001).

4.3.1.2 Machinery Treatment
The installation of an acoustic enclosure for gas turbines can produce 10 to 20 dB lower airborne noise levels than an equivalent enclosed diesel engine. Acoustic enclosure for ships equipped with main diesel engines can also be fitted, but this is not commonly used compared to diesel generator sets, which can be supplied with noise shields as enclosures by vendors. Diesel generators come on a skid or sub-base and are amenable to be resiliently mounted (ABS, 2014a, IACS, 2013).

4.3.2 Path Treatments
Noise path is defined as the construction of the source and the receiver space and as mentioned earlier in this report, a noise path can either be structure-borne or airborne. The paths start at the source’s foundation and include all structures up to but excluding the receiver space(s). Acoustic energy is transmitted by elastic waves through the structures in many ways, including direct path and multiple flanking paths. The airborne path is usually a significant factor only within the source spaces and in areas adjacent to source space (ABS, 2014a, IACS, 2013). The use of damping material and sound absorptive insulation for noise reduction within the engine room can be considered airborne path noise treatment. Another airborne path noise treatment also mentioned above is the installation of enclosures of machinery.
In the case of the bow or stern thruster compartment, which is usually not inside the engine room but a compartment of its own, even though the airborne noise level is usually 100 dB(A) or greater, receiver spaces are rarely adjacent to the thruster compartment and airborne transmitted noise is usually not a concern. However, if there is a receiver space with a standard interface to the thruster room, this interface should be treated with a high transmission loss material such as a limp mass-loaded material sandwiched between compliant layers such as fibreglass or mineral wool. This material can be placed on either the source side or on the receiver room side. The noise reduction from this treatment may be between 3 dB and 7 dB in the low to mid frequency range and up to 12 dB at higher frequencies (Fischer, 2000b, ABS, 2014a).

The greater the number of obstacles along the noise/vibration path, the lower the structure-borne noise will be at the receiver space. Such obstacles include deck/bulkhead intersections, frames, and other non-uniformities in the structure. Another treatment is the use of resilient connections of the piping with machinery and structure. Stiffer and heavier foundation top plates for the machinery will also help reduce vibration amplitude around the foundation, and as a result, less energy will spread to remote spaces (ABS, 2014a).

Damping treatments are also effective structure-borne noise path treatments, and this can be in the form of a relatively thin metal or composite layer tiles or viscoelastic tile material attached to hull plating between the frames. These tiles convert the structure-borne energy to heat via a shearing process thereby attenuating the propagation of structure-borne energy to the receiver (Fischer, 2000b).

### 4.3.3 Receiver Treatments

"Floating" cabins or floors can be used to mitigate noise in compartments located near a bow or stern thruster. A floating room has finish surfaces connected to the hull though resilient mounts. The gap or void between the false deck and structural bulkheads is partly filled with thermal and acoustical insulation such as mineral wool. It would reduce both high structure-borne sound and airborne sound and should provide a minimum of 7 dB(A) noise reduction to a maximum of 20 dB(A). The most critical parameter in this system is the height of the gap between this deck and the structural deck. The void or gap between the structural and false deck should be between 50 and 100 mm depending on the desired noise reduction (Fischer, 2000b).
If required, the receiver space can be further treated with a high sound absorbing and high sound insulating ceiling. There is usually a trade-off between high sound absorption and high sound insulation as high sound absorbing ceilings are typically perforated giving the poor sound reduction. However, a ceiling construction providing both could consist of 2 layers of mineral wool above a 0.6 mm steel plate with perforations.

The noise treatments discussed above are not the only ones available in the market. IMO has also suggested the possible applications of noise-cancelling technologies from other industries for the marine industry such as active mufflers, active mounts, noise-cancelled quiet zones and noise-cancelling headsets (IMO, 2012).

POSSIBLE SOLUTIONS TO MEET POST-MLC VIBRATION REQUIREMENTS
As mentioned above, there is a close relationship between the vibrations of structures and the ship’s acoustics (Asmussen et al., 2001), hence some of those solutions to meet Post-MLC Noise Requirements can also be used for meeting the Vibration Requirements, and the converse is true, especially for structure-borne noise.

4.4 Vibration Design Practices
Having good vibration design practices are just equally important as using good acoustical practices during the initial stage of ship design. Such practices include selecting relatively low vibration machinery and providing propeller clearances (ABS, 2014a).

4.4.1 Selection of Low Vibration Machinery
In recent years, engine manufacturers have made significant progress in reducing vibratory excitation, to a large extent by moment compensators installed with the engine. The ship designer or owner should look into the steps taken by the engine manufacturer and addressed them in the main engine specifications for the new vessel (ABS, 2006). DNV GL also published vibration limits in its class rules for the assessment of machines, equipment and appliances on board ship. Another available guideline is the ISO 10816-6:1995 standard on engine vibration which applies to reciprocating piston engines mounted either rigidly or resiliently with power ratings of above 100 kW. Marine propulsion engine and marine auxiliary engines are typical examples of the application of this standard (ISO, 1995).
4.4.2 Providing Propeller Clearances
It was mentioned earlier that a rule-of-thumb for tip-hull and leading edge-strut/rudder clearance is a minimum 25% of the propeller diameter (ABS, 2014a). A reduced propeller diameter may have to be selected for an increased hull clearance to solve vibration problem, but this would likely affect vessel performance.

The above vibrational practices have shown that it is important to incorporate such good practices during the initial design to avoid corrective actions, which would be costlier once the ship is in service. IACS also recommends the selection of the main machinery with inertia force and moment balanced and the reduction of exciting force by the application of various kinds of dampers, compensators and balancers (IACS, 2013).

4.5 Vibration Predictions in Ship Design
According to Asmussen, a complete investigation of the vibration behaviour should involve not only an examination of the local structures for the danger of resonance but also a prediction of the vibration level. Predicted amplitudes can then be compared with limit values specified for the ship concerned. Furthermore, some design alternatives can be checked, using results of forced vibration analyses for various variants (Asmussen et al., 2001).

Besides using Finite Element Method (FEM) which according to ABS is the most accurate but time-consuming method for the calculations of natural frequencies of the ship’s structural systems, empirical methods such as Kumai’s formula can also be used. However, Yucel and Arpaci have detected up to almost 30% difference between the natural frequency values calculated empirically and the ones obtained using finite element analysis (Yucel and Arpaci, 2013). Nonetheless, ABS recommends that Kumai’s formula may be used to preliminarily evaluate the calculated two-node natural frequency to verify that the FEM frequencies are in a proper range (ABS, 2006).

From the above, it can be seen that similar to noise predictions; vibration predictions are also very useful tools for solving potential vibration problems and also meeting Post-MLC vibration requirements, especially during the initial design stage. Hence the ship designer or yard should not disregard their usefulness and should be prepared to use them.
4.6 Evaluation and Selection of Vibration Control Treatments

4.6.1 Hull Structure Treatment
According to ABS, other than providing the proper propeller clearances, there is little to affect the design of the hull. A wider and longer deckhouse will have the lesser risk of vibration. However, such deckhouse design may not be available for ships such as containers. The design of the stern part of a vessel is important because it can provide uniform flow to the propeller and also greater propeller clearances. A foundation for critical vibration source such as a two-stroke main diesel engine should have a thick foundation top plate, solid floors, and local gussets between these two members in the way of attachment points regardless of whether the machinery is to be isolation mounted or not (ABS, 2014a).

4.6.2 Propeller Treatment
The suggested propeller treatments by Fischer for noise abatement techniques can also be used for vibration control (Fischer, 2000a). Other possible propeller treatments include the use of azimuthing permanent magnet thruster (RR, 2017a) for propulsion and nozzle for smoothing the flow field in the propeller plane, although it can also generate drag at higher speeds (Pacuraru et al., 2010).

4.6.3 Mechanical Source Treatment
The use of isolation mounts for sources of vibration such as engines has been mentioned earlier for noise treatments, and this can be similarly applied for vibration control treatment (IACS, 2013). However, according to ABS, the isolation mounting provides no vibration reduction below the system's natural frequency and may amplify the vibration in the vicinity of the system resonance. Resilient mounts need careful design as the vibration levels of the machinery itself are higher on resilient mounts than that if the machinery is hard mounted and also another concern is that the excitation frequencies of the machinery should not coincide with isolation system’s natural frequencies. Hence to avoid resonance, the choice of the resilient mounts should be accompanied with calculations of natural frequencies of a machinery-resilient mount system. The lower the natural frequency of softer resilient mounts, the lower the vibratory effect will be on the foundation and adjacent structures (ABS, 2014a).

5. CONCLUSION
In this paper, the requirements of MLC regarding noise and vibration in accommodation spaces, its impact on the design and performance of new ships and the possible solutions to meet such requirements have been examined.
It can be concluded that the impact due to the MLC noise and vibration requirements in accommodation spaces would be on ships above 1,600 GT for noise requirements. It is because of MLC’s reference to the IMO Noise Code 2012 which came into force on 1 July 2014 while the impact for vibration requirements are the same as Pre-MLC requirements as these are non-mandatory at the time of writing and are based on ISO 6954:2000 guidelines.

Some of the possible solutions for new ship design to meet the noise and vibration requirements include excellent acoustical and vibration design practices, noise and vibration predictions during ship design, and noise and vibration control treatments subsequently.

As mentioned in the introduction of the paper, a Sustainable Maritime Transportation System requires improving social sustainability not only in the areas of fair wages, contractual terms but also in the working and living conditions as well as their health and safety on board ships. Hence, by meeting not only the mandatory but also voluntary guidelines of MLC Regulation 3.1, ship operators can ensure that their seafarers will have decent accommodation and recreational facilities on board that are consistent with promoting the seafarers’ safety, health and well-being, thereby helping to contribute to the Maritime Sustainability.
REFERENCES

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