VOLUME 2

SCHEDULE 'D'

SPECIFICATION, CONTROL, REDUCTION & MEASUREMENT OF ACOUSTIC NOISE AND VIBRATION

INTRODUCTION

Onboard Noise

The research duties of the vessel require an environment that will allow the scientists and ship's staff to function in the most efficient manner possible, both whilst at work and when relaxing or sleeping. The limits contained in this document, in Tables D1 and D2, are based upon these requirements. All noise limits are sound pressure levels in dB re: 20 μ Pa.

It should be noted that the measures needed to achieve a low underwater radiated noise (URN) signature will also have a beneficial effect on the noise levels achieved throughout the accommodation areas and laboratories.

Onboard Vibration

The levels of vibration are to be controlled throughout the vessel to (i) reduce the stress in structural or mechanical components; (ii) minimise adverse effects on equipment reliability or operational efficiency; (iii) prevent interference with the performance of scientific and ship's staff duties; and (iv) prevent discomfort to personnel when off duty or sleeping. To meet these obligations, limits are set forward in Table D3.

Underwater Noise

The vessel will undertake research and survey tasks that necessitate compliance with this document and the Statement of Requirements NERC Underwater Radiated Noise "URN1 or URN2" (10 knot) and "URN3" (5 knot) as set forward in Figures D1 or D2 and Figure D3. This requirement is a modification of the recommendation made in Cooperative Research Report No. 209 by the International Council for the Exploration of the Sea (ICES). The NERC specification and its adjuncts contained in this document place limits on the allowable level of underwater radiated and vessel self noise for the purpose of:

a) ensuring all oceanographic acoustic instruments operate at a high efficiency when the vessel is free running at all speeds at least up to 10 knots; and

b) ensuring that the overall objectives set by the NOCS geophysics are met with the vessel operating at 5 knots and towing a load of up to 3 tonnes.

Obligations of Shipyard

The shipyard, or their nominated noise & vibration consultant, shall undertake noise and vibration calculations to ensure that the vessel, when delivered, will comply with all the specified noise, vibration and underwater noise limits as set forward in the Statement of

Requirements (SOR). This shall include finite element models to predict the vibration response both locally (for major equipment and structures) and globally (hull girder vibration). In addition, airborne and structure-borne noise prediction models shall be used to assess noise control measures in all areas of the vessel and ensure compliance with the specified limits. Underwater noise models must be developed to demonstrate that the URN specifications “URN1 or URN2” (10 knot) and “URN3” (5 knot) can be achieved with the proposed propeller design and all proposed equipment/mounting arrangements.

NOISE, VIBRATION AND UNDERWATER NOISE LIMITS

Onboard Noise Requirements

Regulations

In terms of work area noise, it should be noted that the Control of Noise at Work Regulations came into force in the UK on the 6th April 2006. However, these regulations excluded ships, which have been dealt with separately. The obligation to comply with EU Directive 2003/10/EC is now met, for owners, operators and managers of ships, under “The Merchant Shipping and Fishing Vessels (Control of Noise at Work) Regulations 2007. These regulations, which protect workers from the risks related to exposure to noise at work, came into force on the 23rd February 2008. Guidance on these regulations can be sought from the MCA website by perusal of Marine Guidance Note MGN 352. The link to this guidance is as follows:

http://www.mcga.gov.uk/c4mca/352-2.pdf

Whilst the above limits provide guidance on occupational noise, this does not address the issue of noise in rest/ sleep areas onboard ships. In this regard, the UK Department of Transport issued a useful guide in its “Code of Practice for Noise Levels in Ships” DTp, HMSO 1990 – now governed by the Maritime and Coastguard Agency (MCA) which is an executive agency of the Department for Transport. Whilst this document provides good guidance on attainable noise levels in ships, it was published in 1990 and therefore needs to be updated to be consistent with the above regulations for work area noise. The MCA state that they intend to revise and update this code of practice to bring it into line with the 2008 regulations; the aforementioned Code should therefore be used with caution with respect to work area noise.

Worldwide advice on suitable noise limits for ships is set forward in the International Maritime Organisation (IMO) “Noise Levels on Board Ships” (1982) which also sets forward noise limits in various work / rest/ sleep areas. Further guidance on noise can also be taken from the various classification societies (Bureau Veritas, Lloyds, Det Norsk Veritas etc) each of which has their own comfort standard with respect to both noise and vibration. These limits are generally lower than both the IMO and DTp guidance documents.

The vessel is to be constructed in order to meet all relevant regulations. Where the SOR requires a higher standard for particular operational reasons then the requirements of the SOR shall prevail.

SOR Limits

A research vessel is considered to be a particularly critical platform as some participants may not be used to being at sea and are therefore likely to be more noise sensitive than those who are sea experienced. In view of this consideration all due care is to be taken to minimise noise and vibration levels for all participants. In particular, intermittently noisy sources, such as thrusters (DP operations) and the vessel stabilisation system (heavy seaway conditions), need to be carefully selected/ designed to minimise their noise impact on cabin areas. This is particularly important as intermittent sources are notorious for causing disturbance, even when noise levels are lower than continuously operating equipment. Any sudden change in noise level, particularly if it is tonal in nature, is likely to cause a reaction (or sleep disturbance) and therefore the noise level increase should be kept as low as practicable. It should be noted that this increase is often much more significant on inherently quiet research vessels as these tend to have low background noise levels internally due to the low structure-borne energy present in the hull – which is enforced due to the URN requirements.

In view of these concerns, noise limits have been specified in Table D1 both for full speed conditions (100% MCR on the propulsion motors) and somewhat higher limits in selected receiver areas when intermittently operated equipment is in use. For the purpose of this specification, it is to be assumed that the latter limits must be met:

(i) whilst the vessel is in DP mode with [50%] load on all thrusters*; and
(ii) whilst the vessel is in heavy seaways [Beaufort Force 6 or greater] with the roll stabilisation system in operation**.

The intermittent source requirements will be assessed separately.

To minimise the noise impact of intermittently operated equipment, all such items must be selected not only on the basis of cost/ mechanical efficiency/ size etc but also on the basis that their design is such that strong tonal characteristics are not exhibited.

The area noise limits required for this vessel are detailed in Table D1.

AREA	Noise limit in dB(A) with:	
	the vessel at full speed (i.e. 100% MCR on the propulsion motors)	**either D.P. operations* or the stabiliser** system operating**
MISCELLANEOUS		
1 m from any ventilation inlet or outlet on open decks	80	
On all other areas on open decks - includes diesel exhaust noise	75	
BRIDGE DECK		
Wheelhouse	60	
FORECASTLE DECK		
Cabins	50	60
Ship’s office	60	
Store rooms	75	
Muster station	70	

BOAT DECK		
Cabins	50	60
Purser's office	60	
Store rooms	75	
Laundry rooms	75	
Meteorological laboratory	60	
Emergency generator room (operational)	110	
MEZZANINE DECK		
Library and conference room	55	
Technical office	60	
Mess room	60	
Coffee area/ internet café	60	
Galley (only cooker extract hoods operating)	70	
Galley (any kitchen equipment operating)	80	
Bar and lounge	60	
Video room	60	
Stores and provision rooms	75	
HVAC room	85	
Bond store	75	
General access toilets	60	
Incinerator room	85	
Garbage compaction and treatment room	85	
Container laboratories	60	
UPPER DECK		
Cabins	55	60
Hospital	55	60
All store rooms	75	
All laboratories	60	
Terminal/ server rooms	65	
Electronic workshop	65	
Coffee shop	60	
Change room	75	
Deck workshop	75	
Main hanger	75	
MAIN DECK		
Cabins	55	60
Laundry rooms	75	
All store rooms	75	
Bottom equipment rooms	75	
Electrical workshop and store	75	
Thruster room (thrusters operating/ not op.)	110/ 85	

Winch room/ winch unit room	85	
Engineer's change room	65	
Engineer's office	60	
Fitness centre	65	
Azimuthing thruster rooms	110	
Hydraulic machinery room	110	
LOWER DECK		
Generator room	110	
Engine control room	60	
Separator and auxiliary equipment room	85	
Engineer's workshop	85	
Engineer's store	75	
Domestic/chilled water/clean SW plant room	85	
Switchboard and electrical equipment room	85	
Auxiliary equipment room	85	

TABLE D1: Noise Limits under Two Operating Conditions for Discovery Replacement Vessel

Operating Conditions associated with Table D1:

- (i) vessel to be at a displacement and trim representative of service conditions;
- (ii) vessel going ahead at full speed (100% MCR on propulsion motors);
- (iii) vessel on a straight course (rudder variations < +/- 2 degrees of helm);
- (iv) vessel to be fully outfitted and ready for service;
- (v) all normally operating plant to be in use during the trials;
- (vi) all HVAC and ventilation systems to utilise mid range positions or a mutually agreed setting;
- (vii) vessel to be proceeding in a depth of water not less than five times the draught of the ship;
- (viii) trials are to be conducted in conditions less than sea state 3 and winds less than Beaufort force 4 (excludes DP or stabiliser trials).

Measurement Equipment/ Conditions associated with Table D1:

- (i) noise measurements to be carried out using a precision grade (Grade 1) sound level meter;
- (ii) The 'A' weighted noise level and 1/3rd octave band noise levels (20 Hz – 10 kHz minimum) to be measured at all locations;
- (iii) noise measurements to be taken at room centres at head height (1.7 m) and time averaged over a spatial sweep (to avoid standing wave effects);
- (iv) in large spaces (> 7m long) additional measurements are required (typical 5 m spacing, locations to be agreed) and all measured values to be arithmetically averaged.

Sound Insulation Limits

The sound insulation requirements for noise transmission between adjacent areas, such as cabins, are specified in various documents, including the IMO "Noise Levels on Board Ships" (1982). This document calls for the airborne sound insulation properties for bulkheads and decks to be no less than an I_a value of 30 dB between cabins and not less than 45 dB between mess-rooms/ recreation rooms and cabins.

The sound reduction index R_W, (previously known as Ia) is measured in dB and is a measure of the transmission loss afforded by a panel, wall or other such structure. It is independent of the partition area and the reverberation time of the receiving room as corrections are made for both these parameters under laboratory test conditions. When such panels are installed in-situ, the actual level difference will again be corrected for partition area and the reverberation time in the receiving room; but in this case the measured level difference will also include the energy transmitted via any flanking paths or other building components. The measured parameter then becomes the weighted apparent sound reduction index, R'_W.

In this application, where the vessel will be relatively quiet due to the low structure-borne vibration energy within the shell plating, the acoustic performance of the partitioning system becomes even more important – due to the enhanced audibility of various sources when background noise levels are low. Thus if cabins are to contain any form of entertainment systems, such as televisions or hi-fi equipment, then to limit their audibility in adjacent areas, the partitioning must have a minimum acoustic performance. Similarly, low background noise also causes speech intelligibility to increase due to speech in corridors/ adjacent cabins.

The sound insulation requirements for decks or partitioning materials are set forward in Table D2 below.

AREA to AREA	Building Element Sound Insulation (R_w) and in-situ Sound Insulation (R'_w)	
	Weighted Sound Reduction Index, R_w	Weighted Apparent Sound Reduction Index R'_w
All cabins areas to cabin areas (includes hospital)	38	36
All cabin areas to corridors	38	36
All cabin areas to noisy public spaces (messrooms, recreation rooms, bars, cafes etc.)	50	48
All cabin areas to machinery spaces* (e.g. HVAC rooms) or engine casings	50	48
Conference room/ offices to noisy public spaces	50	48
Conference room/ offices to corridors	38	36

TABLE D2: Sound Insulation Requirements for Discovery Replacement Vessel

* This does not apply to the diesel generator room where source noise levels are considerably higher than HVAC rooms etc.

The weighted apparent sound reduction index values (R'_W values) will be measured in situ and compared with the permissible values set forward in Table D2. Providing that the partitioning materials have been selected to meet the appropriate weighted sound reduction index value (R_W) then the insulation requirements should be met provided that all flanking transmission paths are reasonably well controlled/ avoided.

Onboard Vibration Requirements

In terms of occupational vibration on ships, there are general purpose standards, such as ISO 2631-1 (1997) which set forward a means of assessing whole body vibration levels. The requirements on whole body vibration and hand arm vibration have now been incorporated into legislation in a similar manner to the noise regulations. The obligation to comply with EU Directive 2002/44/EC is now met, for owners, operators and managers of ships, under "The Merchant Shipping and Fishing Vessels (Control of Vibration at Work) Regulations 2007. These regulations, which protect workers from the risks related to exposure to vibration at work, came into force on the 23rd February 2008. Guidance on these regulations can be sought from the MCA website by perusal of Marine Guidance Note MGN 353. The link to this guidance is as follows:

http://www.mcga.gov.uk/c4mca/353-2.pdf

The vibration standard ISO 6954-2000 "Mechanical Vibration – Guidelines for the measurement, reporting and evaluation of vibration with regard to habitability on passenger and merchant ships" sets forward a means of assessing onboard vibration levels in both work and rest areas. It is graded into three categories of area viz. passenger cabins, crew accommodation areas and work areas. The classification societies use the same measurement methodology set forward in that standard, but specify many more areas and various grades of class requirements (e.g. Lloyds PCAC1 to PCAC3).

The limits set forward in Table D3 below broadly correspond to the "adverse comments not probable" category and are required to be achieved generally throughout the vessel when operating at 100% MCR on the propulsion motors.

AREA	**Vibration limit ISO 6954:2000 (1 – 80 Hz)**	
	Vibration acceleration level in mm/s^2	**Vibration velocity level in mm/s**
All scientist and crew cabins, hospital	71.5	2.0
Library/ conference room, open decks*	89	2.5
All laboratories and offices	107	3.0
Mess rooms, bars, cafes, video rooms, recreation rooms, wheelhouse	125	3.5
All work areas	143	4.0

TABLE D3: Vibration Limits for Discovery Replacement Vessel

* In the aft areas of the vessel an excess of up to 0.5 mm/s is permitted above the propellers

Operating Conditions associated with Table D3:

(i) vessel to be at a displacement and trim representative of service conditions;
(ii) vessel going ahead at full speed (100% MCR on propulsion motors);
(iii) vessel on a straight course (rudder variations < +/- 2 degrees of helm);

(iv) all normally operating plant to be in use during the trials;
(v) vessel to be proceeding in a depth of water not less than five times the draught of the ship;
(vi) trials are to be conducted in conditions less than sea state 3 and winds less than Beaufort force 4.

Measurement Equipment/ Conditions associated with Table D3:

(i) vibration measurements to be carried out using a meter that complies with the requirements of ISO 6954: 2000 i.e. contains the appropriate weighting filters;
(ii) vibration measurements to be taken tri-axially at deck extremities and in the vertical direction only at all other locations;
(iii) where tri-axial measurements are taken, the assessment is to be carried out in each direction separately;
(iv) vibration measurements to be taken on solid structure (not plate centres) and time averaged over a period of not less than 60 seconds.

Underwater Radiated Noise (URN) Requirements

The vessel will undertake research and survey tasks that necessitate compliance with this document and the following Statement of Requirements.

For DC Motor Driven Vessels NERC Underwater Radiated Noise “URN1” and “URN3” as detailed within Section 5.15 of the SOR must be met.

For Azimuth Driven Vessels or AC Motor Driven Vessels NERC Underwater Radiated Noise requirements “URN2” and “URN3” as detailed within Section 5.15 of the SOR must be met.

The NERC specification and its adjuncts contained in this document place limits on the allowable level of underwater radiated and vessel self noise for the purpose of:

a) ensuring all oceanographic acoustic instruments achieve maximum operational capability when the vessel is free running at speeds up to and including 10 knots – requirements “URN1” or “URN2”;

b) ensuring that the underwater noise level at 250 m astern of the vessel, and at a depth of ~10 m, will broadly comply with the NOCS geophysics requirement not to exceed 5 μbar RMS over the frequency range 5 – 300 Hz with the vessel operating at 5 knots and towing a load of up to 3 tonnes – requirement “URN3”;

The 10 knot requirement is very similar to requirements set forward in Cooperative Research Report No. 209 by the International Council for the Exploration of the Sea (ICES). However, the criterion has been reduced from 11 knots to 10 knots to enable the propeller designer to adopt a design which will readily meet both the 10 knot (unloaded) and the 5 knot (loaded) criteria within this SOR.

The 5 knot requirement is a modification of the ICES CRR No. 209 to make the limits more directly applicable to the research work that will be performed by this vessel i.e. low noise astern of the vessel whilst trailing towed arrays.

SECTION 1: Underwater Radiated Noise Requirements for DC Motor Driven Vessels, when Free Running at Vessel Speeds up to and including 10 Knots

To allow full exploitation of the capabilities of advanced acoustic scientific instruments, including echo sounders, sonars, ADCP units, navigation systems and other acoustic equipment referred to in the SOR, the underwater radiated noise level must meet the limit “URN1” shown in Figure D1 when the vessel is free running at all speeds up to and including 10 knots.

These limits are expressed in terms of 1/3rd octave bands as the sound range facilities will measure in 1/3rd octave bands. These values can be converted back to a mean spectrum level (i.e. the average energy level per Hz) by subtracting 10 log (bandwidth) from the 1/3rd octave band values, which is the reverse process of that detailed below.

N.B. ICES CRR 209 presents its guideline limits as spectrum levels rather than 1/3rd octave band values.

The noise limits can be derived as detailed in the following three expressions:

(1) From the 10 Hz to the 20 Hz 1/3rd octave band, the band levels are given by the expression:

Band level = 160 – 20 log (f_{Hz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{Hz} is the Octave Band Centre Frequency (OBCF) in Hz

(2) From the 25 Hz to the 800 Hz 1/3rd octave band, the band levels are given by the expression:

Band level = 135 – 1.66 log (f_{Hz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where again:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{Hz} is the Octave Band Centre Frequency (OBCF) in Hz

(3) And from the 1 kHz to the 50 kHz 1/3rd octave band, the band levels are given by the expression:

Band level = 130 – 22 log (f_{kHz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{kHz} is the Octave Band Centre Frequency (OBCF) in kHz

For the avoidance of all doubt, the bandwidth of each one third octave band shall be given by the expression:

Bandwidth = (OBCF x 2 $^{(1/6)}$) – (OBCF / 2 $^{(1/6)}$)

For guidance it should be noted that the bandwidth is approximately 23.16% of the OBCF for all 1/3rd octave bands.

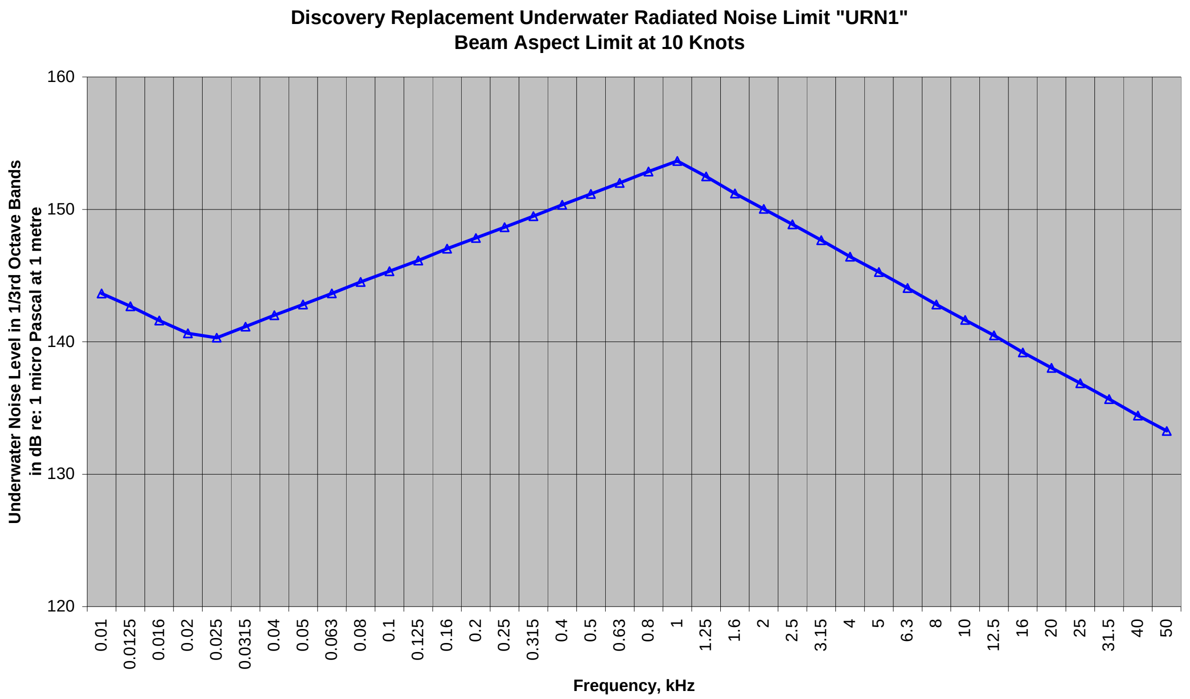

FIGURE D1: Underwater Noise Limit "URN1" for Beam Aspect of a **DC Motor** Driven Vessel when Free Running at all Vessel Speeds up to and including10 Knots

SECTION 2: Underwater Radiated Noise Requirements for Azimuth Driven or AC Motor Driven Vessels when Free Running at Vessel Speeds up to and including 10 Knots

Once again, to allow full exploitation of the capabilities of all acoustic scientific instruments, including echo sounders, sonars, ADCP units, navigation systems and other acoustic equipment referred to in the SOR, the underwater radiated noise level must meet the limit "URN2" shown in Figure D2 when the vessel is free running at all speeds up to and including 10 knots. In this case, the underwater radiated noise shall exclude the propulsion motors and propellers below 1 kHz but shall include the propulsion motors and propellers at 1 kHz and above.

These limits are expressed in terms of 1/3rd octave bands as the sound range facilities will measure in 1/3rd octave bands. These values can be converted back to a mean spectrum level (i.e. the average energy level per Hz) by subtracting 10 log (bandwidth) from the 1/3rd octave band values, which is the reverse process of that detailed below.

N.B. ICES CRR 209 presents its guideline limits as spectrum levels rather than 1/3rd octave band values.

The noise limits can be derived as detailed in the following two expressions:

(1) From the 10 Hz to the 800 Hz 1/3rd octave band, the band levels are given by the expression:

Band level = 135 – 1.66 log (f_{Hz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{Hz} is the Octave Band Centre Frequency (OBCF) in Hz

(2) And from the 1 kHz to the 50 kHz 1/3rd octave band, the band levels are given by the expression:

Band level = 130 – 22 log (f_{kHz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{kHz} is the Octave Band Centre Frequency (OBCF) in kHz

For the avoidance of all doubt, the bandwidth of each one third octave band shall be given by the expression:

Bandwidth = (OBCF x 2 $^{(1/6)}$) – (OBCF / 2 $^{(1/6)}$)

For guidance it should be noted that the bandwidth is approximately 23.16% of the OBCF for all 1/3rd octave bands.

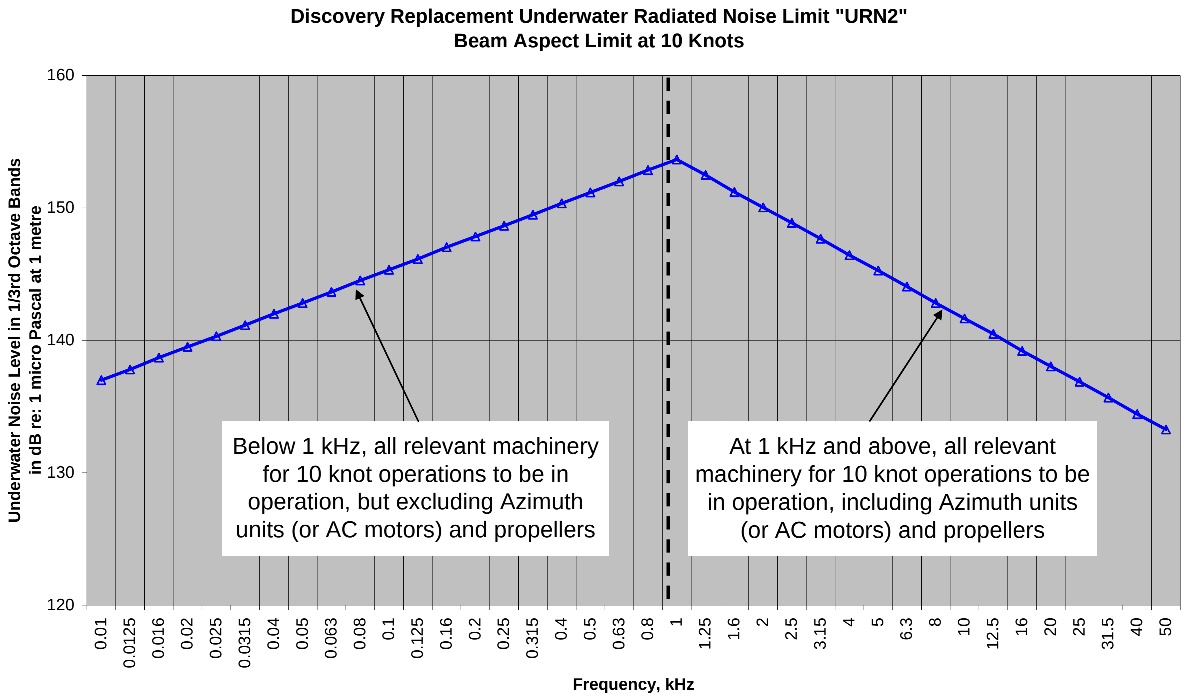

FIGURE D2: Underwater Noise Limit "URN2" for Beam Aspect of **Azimuth or AC Motor Driven Vessels** when Free Running at all Vessel Speeds up to and including10 Knots

SECTION 3: Underwater Radiated Noise Requirements for DC Motor Driven or Azimuth Driven or AC Motor Driven Vessels when Towing a Load of 3 tonnes at Vessel Speeds up to and including 5 knots

To ensure that towed arrays, and the like, will be subject to an acceptable level of underwater noise when located some 250 m astern of the vessel, and at a depth of ~ 10 m, the underwater radiated noise level must meet the limit “URN3” shown in Figure D3 using a stern aspect assessment. To simulate these towing operations, the vessel must operate at speeds of up to and including 5 knots whilst towing a load of 3 tonnes.

These limits are again expressed in terms of 1/3rd octave bands. But these values can be converted back to a spectrum level (i.e. the mean energy level per Hz) by subtracting the 10 log (bandwidth) from the 1/3rd octave band values, which is the reverse process of that detailed below.

The noise limits for the 5 knot towed condition can be derived as detailed in the following two expressions:

(1) From the 10 Hz to the 800 Hz 1/3rd octave band, the band levels are given by the expression:

Band level = 145 – 1.66 log (f_{Hz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{Hz} is the Octave Band Centre Frequency (OBCF) in Hz

(2) And from the 1 kHz to the 50 kHz 1/3rd octave band, the band levels are given by the expression:

Band level = 140 – 22 log (f_{kHz}) + 10 log (bandwidth) expressed in dB re: 1 μ Pascal at one metre

where:

(i) bandwidth is the bandwidth of the relevant 1/3rd octave band; and
(ii) f_{kHz} is the Octave Band Centre Frequency (OBCF) in kHz

Once again, for the avoidance of all doubt, the bandwidth of each one third octave band shall be given by the expression:

Bandwidth = (OBCF x $2^{(1/6)}$) – (OBCF / $2^{(1/6)}$)

In addition to the 1/3rd octave band limits, narrow band tonal components shall not exceed a source level of 156 dB at 1 metre over the frequency range 10 to 250 Hz when using a measurement bandwidth of ~ 1 Hz.

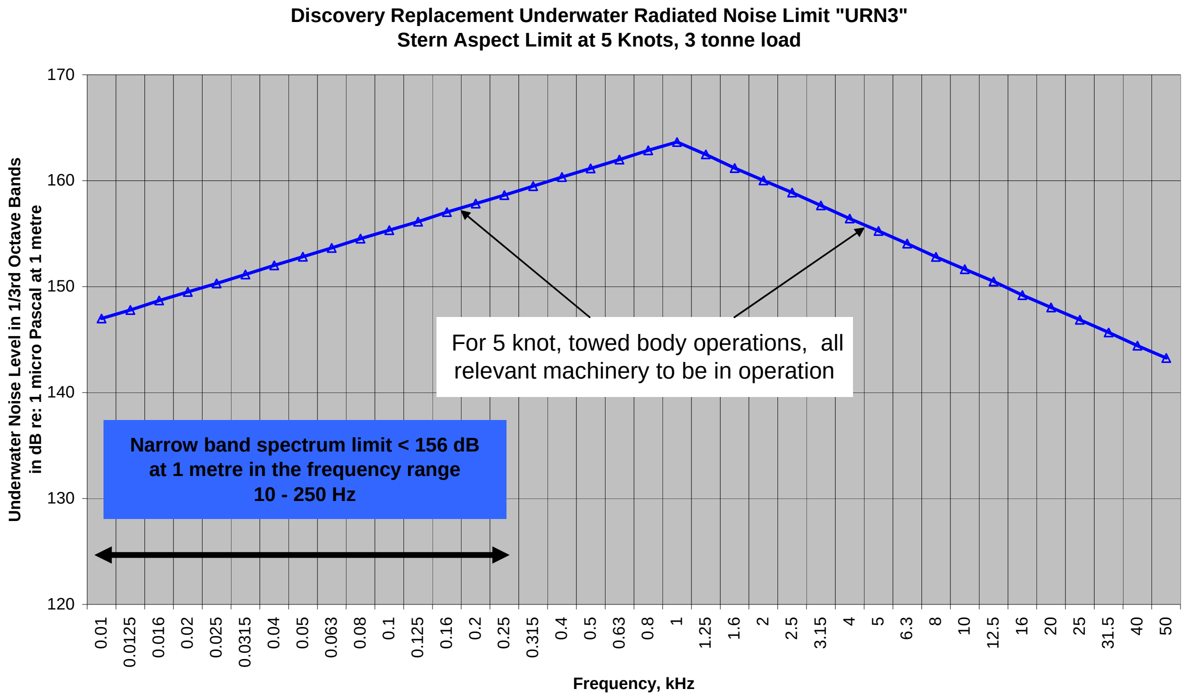

FIGURE D3: Underwater Noise Limit "URN3" for Stern Aspect of the Vessel when any Drive System (DC/AC/Azimuth) is used on the Vessel, whilst Towing a Load of 3 Tonnes at Vessel Speeds of up to 5 Knots

NOISE AND VIBRATION GUIDANCE

General Noise & Vibration Control Measures

Structure-borne noise should be controlled from all rotating and reciprocating machines should be controlled by appropriate measures. Below is a list of suitable techniques:

- select inherently well balanced machines (mainly applies when soft mounting is not suitable), the application of reciprocating or positive displacement pumps/compressors is to be avoided as far as possible and where this is not possible special measures are to be adopted to counteract their noise and vibration influences;
- the use of anti-vibration mounts beneath all suitable rotating and reciprocating equipment;
- the use of anti-vibration mounts beneath all ancillary pieces of plant where appropriate e.g. diesel exhaust supports;
- the use of flexible bellows or hose on the supply and discharge lines from soft mounted pumps, fans etc;
- the use of flexible pipework supports on noisy supply and discharge lines from centrifugal pumps, hydraulic pumps, compressors etc;
- design suitably robust seatings for all items of equipment, including soft mounted machines – stiffen seatings as appropriate;
- ensure all pipe supports are located on suitably stiff structure, ideally onto stiffeners - never at plate centres;
- do not provide mechanical links (pipe restraints) between low noise lines, where no special precautions have been made, and noisy lines where special measures have been adopted.

In addition, other measures may need to be adopted to control noise & vibration in appropriate circumstances. Below is a list of possible measures:

- the use of constrained layer damping systems, damping tiles or damping cassettes on shell plating to limit noise radiation;
- the use of specialised fireproof materials on the shell plating to limit airborne noise radiating from the generator compartment;
- the use of specialised damping materials on the tank top to limit structure-borne noise radiation from propulsion motors or generators.

Specific Equipment Noise & Vibration Control Measures

Main Diesel Alternator Sets

Structure-borne noise is to be controlled by double isolating the main diesel generator sets. Conventionally this can be accomplished by either: using a skid mounted diesel alternator set that is soft mounted from the raft structure, which is then in turn soft mounted from its seatings; or by hard mounting the alternator to the raft and using a flexible coupling between the diesel engine and the driven alternator, with again mounts beneath the raft. If the former option is preferred, then it is important to select a stiff skid to ensure that mounts “see” a high mechanical impedance and to ensure that shaft alignment is maintained between the diesel and the alternator. The latter option, where the alternator is hard mounted to the raft, is generally preferred as the alternator casing then stiffens up the raft structure thereby raising the raft flexural/ torsional modes. However, in that instance the alternator must be of a herringbone design, with skewed slots, as the alternator forces will only then be attenuated by a single stage isolation system. For design guidance, the first raft flexural/ torsional mode should be set to be not less than 10% above the firing frequency of the diesel engine.

The diesel engine must be selected to minimise the diesel orders, or line components, as far as practicable. This may be accomplished by good design practice. All appropriate design studies must be carried out to ensure that the shafting does not exhibit resonant characteristics at any relevant speed. A torsional vibration study must also be carried out and vibration dampers selected to minimise vibration levels on the set.

The two sets of anti-vibration mounts required for the rafted diesel alternator set must be selected through an iterative design process to optimise the attenuation and minimise the transmitted vibration forces such that the relevant seating vibration limits, as advised by the shipyard’s consultant are achieved. These vibration limits must be compatible with the underwater radiated noise criteria “URN1” and “URN2” as set forward in Figures D1 and D2 in the section on “Underwater radiated noise requirements”.

The mounting system must also be selected to ensure that the whole body movements at the top of the set, under the most extreme seaway conditions, are kept to acceptable limits i.e. can be accommodated by the turbocharger bellows units. Where appropriate a buffer system is to be employed to limit the motion of the set under heavy seaway conditions or in the worst case collision situation. The buffers must be left to provide clearance under all normal operational sea-states [to be defined]. The seating structure on the tank top must be of high mechanical impedance to optimise the acoustic performance of the raft mountings.

The diesel exhaust must contain high performance silencer systems. These may comprise conventional reactive/ dissipative silencers and/ or multiple compact silencers to de-tune strong tonal components (e.g. Wartsila CSS silencers). The overall sound insertion loss of the silencer system shall not be less than 40 dB(A). In addition, the exhaust system must be designed to ensure that it can accommodate all necessary thermal movements/ forces as well as heavily attenuating transmitted vibration forces. This is conventionally achieved by the use of stabilisers combined with anti-vibration supports. The mounts must be capable of withstanding the anticipated operating temperatures; stainless steel mesh mounts are not to be used due to their poor high frequency attenuation characteristics. All elastomeric based mounts must be protected by means of heat sinks etc if temperatures exceed their maximum normal long term working temperature. Mounts must be selected to have a design life of not less than [15 years] at service conditions.

All pipework systems (such as cooling water lines) which cross between the diesel and the adjacent structure, such as the tank top, must have high performance (or multiple) bellows or hose to minimise transmitted energy. High energy lines attached to the diesel engines should not be attached to the deckhead structure – as far as practicable. In addition, the diesel fuel feed system must contain pulsation dampeners on both supply and return lines. The air start system, which hooks up to the diesel engine, must also be de-coupled from the tank top as far as is practicable to minimise diesel energy being transmitted via the air start line. Ideally this is accomplished by running the air start lines along the raft structure for as far as is practicable and using high performance mounts on this pipework run. In addition, high performance bellows must be installed at both ends of this raft mounted pipework. Failure to address these transmission paths could lead to a significant increase in the overall transmission through the rafted double isolation system.

Main Propulsion Motors

If the main propulsion motors are to be large DC motors, then such units will be hard mounted to the hull. In such cases the propulsion motor must be designed and fabricated on low vibration principles. Typically this will involve skewed slots, controlled air gap between rotor and stator, low noise cooling system etc.

The vibration characteristics of the set must be achieved when operating from the shipboard electrical supply under the specified loading levels. The supply system must therefore be as smooth as possible e.g. a 24 pulse rectifier system.

An alternative drive configuration is to use AC main drive motors which are driven at variable speed via frequency converters, or similar, and directly coupled to the propeller. This form of arrangement generates low frequency vibration due to the waveform of the input supply to the motor i.e. harmonic energy. In that eventuality some form of vibration control will be necessary. This may be accomplished by use of a suitable design process, such as active cancellation filters, to minimise these harmonic components. However, it should be noted that, if active techniques are employed, the filters must function adequately over the full vessel speed range up to and including the nominated maximum speed.

In the event that high speed AC motors are to be coupled to azimuth units, then the motors will require some form of vibration control. This may be accomplished by the use of anti-vibration mounts and a flexible coupling to the azimuth unit. Whether a “Z” or an “L” drive configuration is employed, the motors must be mounted on heavy structure. In the event that an “L” drive is employed, then the structural supports will ideally be taken from the deckhead above.

Any frequency converters must also be soft mounted to limit the structure-borne vibration energy entering the hull.

Propellers

Noise radiation from propellers is likely to be one of the most important issues on this vessel as the URN performance of this vessel will be strongly linked to the cavitation inception speed under both 5 knot and 10 knot test conditions. In addition, at the bottom end of the spectrum, the fundamental blade rate frequency and its first few harmonics will dominate the vessel noise radiation (typically at around 10/ 20/ 30 Hz assuming a 5 bladed propeller running at ~ 120 rpm). It is therefore imperative that the propeller is designed to minimise the blade rate energy as well be non-cavitating under both sets of operational conditions. To meet these requirements, it is anticipated that a five bladed, slow running speed, large diameter propeller will be required. The propeller designer must select a suitable propeller profile such that the two sets of design criteria lie within the “operational bucket” thereby minimising underwater noise radiation as far as is practicable.

Whether the main propulsion is to use DC or variable speed AC motors or smaller high speed AC motors coupled to azimuth units, the propellers are to be either fixed pitch or adjustable pitch units. In the latter case, the blade angle may be changed in the light of trials results to a new fixed pitch angle – thereby providing some flexibility in the event that URN limits are not met under either of the two operational conditions. Controllable pitch propellers (CPP units) are not considered to be consistent with the URN requirements for this vessel.

The propellers are to be treated with anti-singing edges to avoid the strong tones that arise in the event of a singing propeller. Further, the propellers are to have a large hull clearance to minimise coupling effects into the aft vessel structure. These clearances must be in excess of class requirements. At least 1m clearance is to be achieved.

Tank tests are to be carried out to assess the suitability of the design in terms of wake field, cavitation inception speed and the magnitude of pressure pulses which arise at blade rate frequencies. These tests are required under free running and loaded conditions.

Azimuth Units

If azimuth drive units are selected for this vessel, then noise radiation from gearboxes is likely to be another important issue on this vessel. With a conventional bevel drive gearbox, the tones at gear meshing frequencies are likely to be extremely strong, although these can be reduced by special hypoid bevel gears.

If azimuth drive units are used then it is considered to be a prerequisite for special attention is paid to the type, design, manufacture, and testing of all gearbox systems and to use units proven in service for low noise and reliability.

The use of an ‘L’ drive configuration is preferred as it will limit the tones to a single gearbox.

Whilst a ‘Z’ drive configuration would also be acceptable, this would mean two gearboxes on each drive train and two sets of gear meshing frequencies to control. Ideally the number of gear teeth will be selected to ensure that all gear meshing tones arise at frequencies greater than 250 Hz at the two principal operating conditions associated with underwater radiated noise limit “URN1/2” and “URN3”.

In addition to the direct radiation of underwater noise from the gearboxes, there is also the issue of shell plate radiation from the hull structure. It is considered that the azimuth unit should be de-coupled from the hull by means of a suitable resilient mounting system. This may be a relatively stiff (or semi-resilient) element to provide attenuation for just the high frequency gear meshing tones whilst permitting only a very limited degree of movement under the application of propeller or other forces. A flexible coupling will be required between the drive motors and azimuth units.

Thruster Units

Noise radiation from thrusters is an important issue on this vessel due to the potential disturbance of scientists and crew members. Thruster units are to be selected not only on the basis of cost/ mechanical efficiency/ size etc but also on the basis that their design is such that strong tonal characteristics are not exhibited. In general, fixed pitch, rather than CPP units, are to be preferred with variable speed control on the motors. Twin contra-rotating propeller units are not considered to be consistent with the onboard noise requirements for this vessel under DP operations.

To minimise onboard noise during DP operations etc, any tunnel thrusters proposed must utilise a double tunnel arrangement with the two tunnels de-coupled by means of anti-vibration mounts.

Since thrusters impart considerable vibration energy into the hull, both via the machinery components and the high dynamic pressure levels on the shell plating, there will also be the need to provide path attenuation to improve noise levels for nearby cabins. This is discussed in the section on “Specific Receiver Areas” below.

Stabilisation system

The stabilisation system can give rise to onboard noise problems in heavy seaways. Noise typically radiates from the cross-over valve assembly and the cross-over duct supports, both of which are attached to the deckhead above. It is considered imperative to reduce both these two paths. This can be accomplished by: (i) installing the cross-over valve manifold assembly on anti-vibration mounts; (ii) using soft flexible bellows to de-couple this assembly from the rest of the cross-over ductwork; and (iii) using resilient pipe clamps to reduce noise transmission from the cross-over ductwork. The system shall be designed to prevent any contact from significant “slugs” of water impacting on the deckhead above the shell side tanks – via high level control sensors etc. Further, the system shall be designed to revert to a fail-safe status in the event that the stabilisation system becomes inoperable. The Ship’s Staff will then decide whether to allow the stabiliser to operate in the ‘passive’ mode and should be provided with the means by which to achieve this.

Hydraulic equipment

All hydraulic equipment, such as power packs, shall have pulsation dampeners/ silencers installed. The hydraulic lines must be mounted on soft supports from high impedance (stiff) structure to minimise energy from being transmitted into the supporting structure. In addition, the routing of any hydraulic lines must avoid sensitive receiver areas as far as practicable.

Sea Water Sampling Pumps

On many vessels sea water sampling is found to be a noisy operation. Since sampling may be undertaken for long periods of time, it is considered essential to select low noise pumps which minimise the structure-borne input to the vessel. The pumps (2 off) should be Bornemann Twin Screw Non Toxic Pumps Model SLH80-40, or similar, including frequency converters to allow control of capacity. Two pumps are required to provide capacity and redundancy within the system.

General Receiver Area Noise & Vibration Control Measures

Floating Floors

Floating floors are constructions where the floor surface, usually in a cabin, is de-coupled from the underlying structural deck plate by means of an isolation material. This compliant material can comprise individual mounts, mineral wool slabs or may use isolation strips (such as Sylomer). The upper floor structure may be steel, damped steel (i.e. two sets of steel plate separated by a very thin layer of visco-elastic material) or latex cement (if a thicker upper layer is acceptable). A generalised schematic of a floating floor based accommodation system is shown in Figure D4 below.

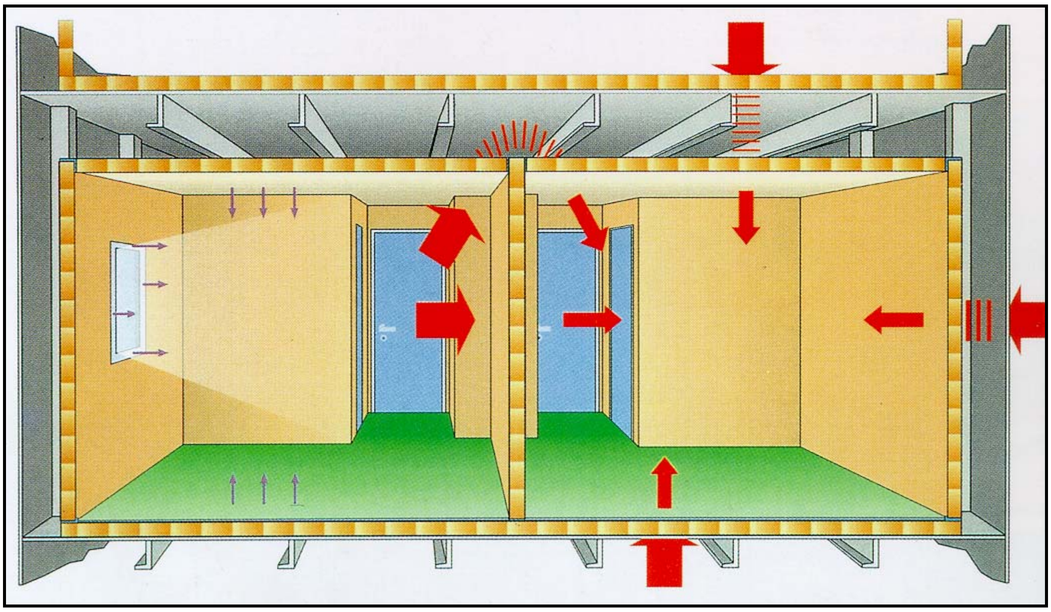

Figure D4: Generalised Schematic showing possible transmission paths into a cabin

On many ships, the most convenient system to install is a steel cassette system where the mineral wool slab is pre-bonded to an upper steel layer. A cassette based floating floor arrangement is shown schematically in Figure D5 below.

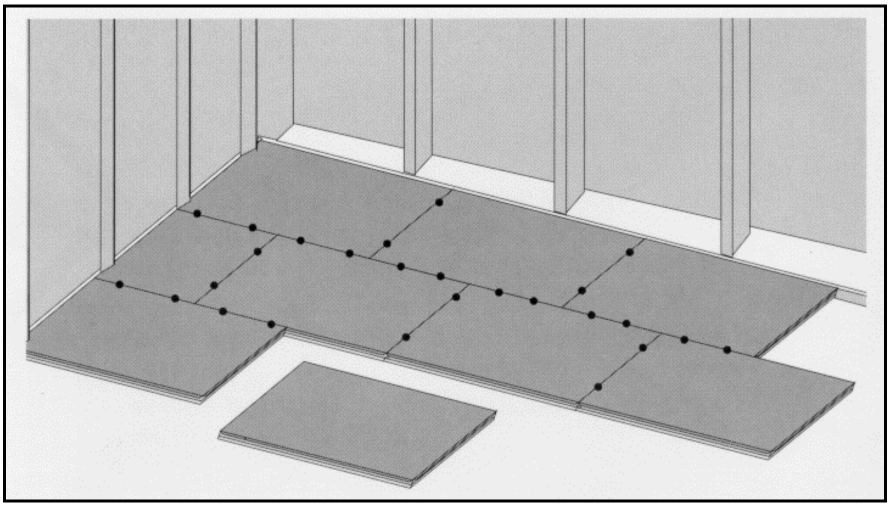

Figure D5: Generalised Schematic showing Floating Floor Slabs or Planks

The floor is generally laid in planks and the planks are then welded together in situ and covered by a deck levelling compound. The edge detail of the floor must be such that there is no contact between the upper plate of the floating floor and the adjacent bulkhead. This is usually accomplished by use of a mineral wool insert piece as shown schematically in Figure D6 below.

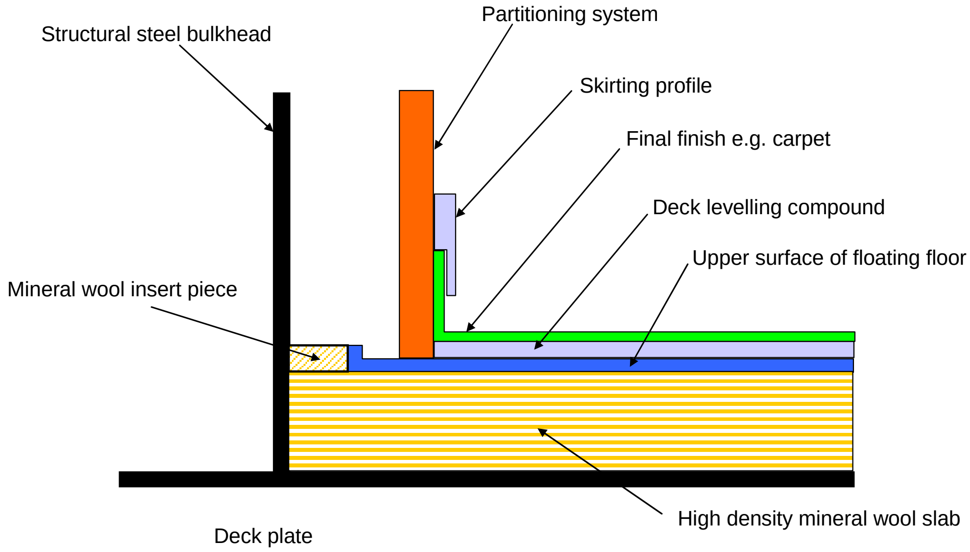

Figure D6: Generalised Schematic of Floating Floor showing Typical Edge Details

There are, however, many other designs that use a mineral wool slab-based system. The individual isolators and strip isolation systems both require more design work as they require a framework to align with the isolators/ strips.

Floating floor systems are most effective when they are installed in large continuous areas between structural bulkheads, with the partitioning systems mounted on the floating floor and ceilings supported by flexible drop hangers from the deckhead above. This design minimises flanking transmission paths. Typically floating floor thicknesses range from 50 to 100 mm when using a steel upper layer.

Floating floors reduce both structure-borne and airborne noise from areas directly below and are therefore ideal, for example, when the receiver area is over a very noisy area (e.g. generator room). However, they are also effective when there is considerable energy in the ship's structure – such as arises when thrusters are in use – providing that a fully isolated cabin arrangement is installed. This should include treatment alongside the shell plating/ port-holes which can be the limiting transmission path.

Constrained Layer Damping Systems

Constrained layer damping systems are damping systems for heavier plates – as used in ship construction work. A flexible visco-elastic damping compound or sheet is bonded to the structural steelwork and then in turn the flexible layer is bonded to an upper constraining layer (usually steel, for reasons of space). The damping efficiency increases with the thickness of the constraining plate; but for practical purposes, the constraining layer is often ~ 1/3rd of the base plate thickness. Thus on top of an 8 mm deck plate, there might be a visco-elastic layer approximately 1 mm thick and a constraining plate typically 3 mm thick so the overall system can be extremely compact. Whilst other constructions can use a thick layer of latex cement as the constraining layer this is less usual on ships due to potential cracking when the vessel flexes under heavy seaway conditions.

A typical constrained layer damping system is shown in Figure D7 when combined with a high performance floating floor system (damped upper layer).

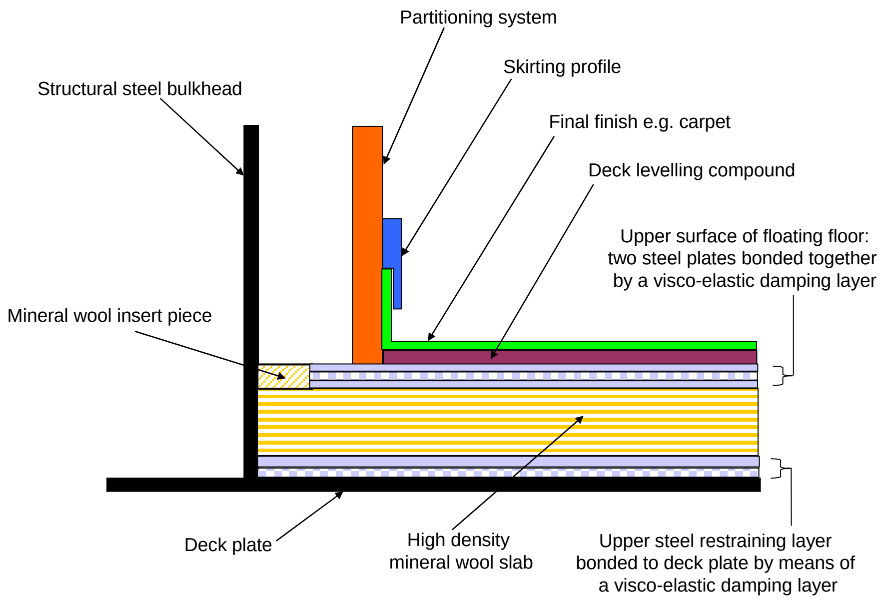

Figure D7: Generalised Schematic showing Constrained Layer Damping System on Deck combined with a high performance (damped) Floating Floor System

Damping Cassettes

Damping cassettes are robust dampers which will provide damping on all lively structures, including heavy shell plating. They consist of a U-Profile backbone into which the visco-elastic material is cast and into which individual fixation legs are imbedded. Damping Cassettes are very simple to fit as they can be welded to flat or curved surfaces and are effective on plate thicknesses from 4 - 50 mm. They are usually installed diagonally on the plating to break-up fundamental modes of vibration – usually in the proximity of the source rather than on the shell plating in receiver areas.

Specific Receiver Area Noise & Vibration Control Measures

Cabins near Forward Thrusters

It is considered that to counteract thruster noise, the combination of both floating floors and constrained layer damping systems is to be used. The constrained layer damping system is installed on the deck plate and then the floating floor system located above. The constrained layer damping system will reduce deck plate panel vibration at the lower frequencies (principally by limiting panel resonances), with the floating floor reducing noise and vibration levels mainly at the higher frequencies. It is not considered that floating floors alone should be used as they are likely to have fundamental natural frequencies (typically 20 – 40 Hz) close to the thruster energy which cause poor performance at these frequencies.

Cabins above Roll Stabilisation System

The noise from the stabiliser valves will be well attenuated following the isolation of the entire manifold assembly and the cross-over ductwork. However, even with a well designed and vibration isolated system, there will still be residual noise passing through the valve manifold/ cross-over duct supports. It is therefore considered that cabins above the roll stabilisation system manifold should also have both constrained layer damping systems and floating floors installed as described above for the thrusters.

General

In view of possible height differences (typically 80 mm assuming a 75 mm thick floating floor), it is recommended that all cabins within the same block should adopt this same floor design. Assuming the layout as set forward in Concept GA this means that all cabins on the Main Deck for'd of Frame 79 and the two Upper Deck cabins and hospital, for'd of Frame 107 should have the combined floating/ damped floor system as detailed above. Similarly, due to the operation of azimuth units in DP mode, or stern thrusters, it should be assumed that all Main Deck cabins aft (between Frames 27 and 49 on Concept GA) will require the same treatment. The treatment for these, and all other cabin areas, should be assessed by the noise model developed by the shipyard's noise and vibration consultants.

Combined with the best possible design measures at source, on the thrusters and roll stabilisation system, it is believed that overall noise levels will be kept as low as is practicable. The use of both constrained layer damping systems and floating floors will also mean that noise from any other intermittently operated sources (such as anchor handling operations, winch noise, or any hydraulically driven deck handling equipment) will be heavily attenuated to give minimum disruption/ sleep disturbance to onboard personnel.