THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
One World Trade Center, Suite 1369, New York, N.Y. 10048
Paper to be presented at the Ship Vibration Symposium
Arlington, Va., October 16–17, 1978

Noise Prediction and Prevention in Ships

A. C. Nilsson, Visitor, Det Norske Veritas, Oslo, Norway

ABSTRACT

The prediction and prevention of structure-borne sound on board merchant ships are discussed.

Results of full-scale and model-scale experiments have indicated that the main power flow in the vertical direction in a ship structure is determined by the propagation of flexural waves in the plate elements. Based on these results a prediction model is developed. It is found that the velocity level of a deck is a function of the input power at the source, wavenumbers, masses, losses and dimensions of the plate elements in the structure. The attenuation of structure-borne sound is also a function of frequency. Predicted and measured structure-borne sound levels are compared.

The effects of damping layers and resisliently mounted superstructures and accommodation systems are also discussed.

INTRODUCTION

During the last few years recommendations or regulations concerning maximum permissible noise levels on merchant ships have been issued by the authorities in most countries. Some of the recommendations are currently being revised and are expected to be issued as requirements. In general the required noise level in cabins is 60 dB(A). This noise level can generally not be achieved on small ships without noise reducing measures. In Fig. 1 the distribution of measured dB(A) levels in cabins on 15 randomly chosen ships is shown. The ships -all below 20 000 tdw - are built in Norway during 1976 and 1977. The 60 dB(A) level is exceeded in 2/3 of the cabins although noise reduction measures have been considered on some of the ships. The results shown in Fig. 1 emphasise the need for useful and accurate noise prediction programs.

The purpose of a noise prediction program is to make it possible already at the design stage to estimate the noise levels in a ship. If, say, the initial design is found to be acoustically insufficient, the effects of improved sound insulation and general arrangement can readily be calculated. A noise prediction program can also make it possible to find the most economical solution to achieve a certain noise requirement. Ideally, the prediction of noise levels should be standard procedure for every new construction. To make this possible it is necessary that the costs involved for using a program are low.

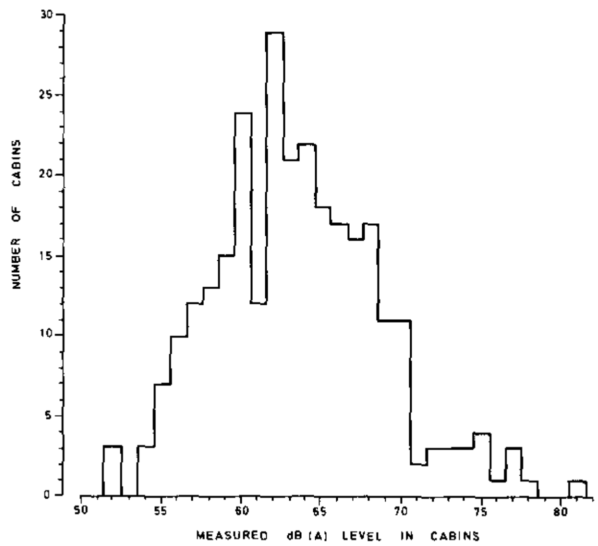

Fig. 1. Distribution of noise levels in 282 cabins on 15 randomly chosen ships.

NOISE GENERATION

The noise situation on board a ship is determined by the sum of noise contributed by the many and varied forms of noise sources. Effective shipboard noise control therefore requires identification and knowledge of all significant noise sources.

The most important noise sources are: main and auxiliary engines, propeller, gear, casing and exhaust systems including funnel, various pumps, compressors, hydraulic systems and fan equipment including air intakes and outlets. In rooms containing noise sources such as engines, fans or pumps the sound pressure level is almost entirely determined by airborne sound. Methods to predict and reduce airborne sound are well known and are extensively treated in the literature. See for example the references /1/ - /4/. Typical sound reducing measures are partitions, hoods, screens and sound absorbing materials.

In accommodation spaces other than those mentioned above, with the possible exception of rooms directly adjoining a source, the noise level is determined by structure-borne sound.

The term structure-borne sound refers to structural vibrations in the frequency range 16 - 20000 Hz. These high frequency vibrations which are well coupled to water and air, radiate audible noise into these media.

Structure-borne sound is directly induced by any mechanical force. The mechanical power transmitted from a source through its connection to the foundation propagates into the structure. The power can propagate in the structure as flexural, longitudinal, transverse and torsional waves. The relative importance of these wave types has been discussed in the references /5/ -/8/.

The resulting energy flux is attenuated as function of the distance from the distrubance. The attenuation depends on losses in the structure and also on the number of obstructions or discontinuities (decks, platforms, frames) in the propagation path. At the receiving end - in for example a cabin - the acoustical power radiated from a structure depends on the velocity level of and the material parameters and dimensions of the structure. To make a prediction of resulting noise levels in an accommodation space possible, the following quantities must be known:

i) source strengths
ii) transmission properties of steel structure
iii) radiation properties of structures at the receiving end

These properties are further discussed below.

Janssen and Buiten /9/ first formulated general expressions for the calculation of noise levels induced by structure-borne sound in accommodation spaces on board ships. Assume that the rms velocity at the foundation of a source is L_V(source) and the resulting rms velocity of a deck is L_V(deck), then

$$L_V(\text{deck}) = L_V(\text{source}) - F \qquad (1)$$

where the function F determines the attenuation of structure-borne sound in the path between source and receiver. F is a function of the number of discontinuities (decks, platforms, frames, etc.) between source and receiver as discussed below.

The structure-borne sound is transmitted from the steel construction to the floor, bulkheads and ceiling in the cabin. The type of mounting of each surface and its connection to the steel construction determine the velocity level difference, ΔL_V, between the steel deck and the radiating surface.

The velocity level L_i of a radiating surface can thus be written as:

$$L_i = L_V(\text{deck}) - (\Delta L_V)_i \text{ dB} \qquad (2)$$

The sound pressure p_i in a room induced by one vibrating surface with the area S_i, velocity u_i and radiation ratio s_i can be written as:

$$p_i^{\,2} = \frac{4(\rho c)^2 \cdot u_i^{\,2} \cdot S_i \cdot s_i}{A} \qquad (3)$$

where ρc is the acoustical impedance of air and A the equivalent absorption area (m^2) in the room. The resulting sound pressure level L_{pi} from one source and one surface is thus:

$$L_{pi} = L_i + 10 \log (4\, S_i\, s_i/A) \quad \text{dB} \qquad (4)$$

The reference sound pressure in the equation above is set to equal 20μPa. The velocity level L_i is given in dB re. 50 nm/s.

The total level L_{tot} in an accommodation space caused by all the surrounding radiating surfaces is for each source determined through:

$$L_{tot} = 10 \log [\Sigma 10^{L_{pi}/10}] \text{ dB} \qquad (5)$$

The summation is made over all the surfaces.

In every accommodation space the contributions from the various noise sources are added logarithmically for each of the nine octave bands starting at 32 Hz and ending at 8 kHz. The resulting A-weighted sound level can thereafter be calculated.

SOURCE STRENGTH

In order to define the strength of a source it is in general sufficient to determine the velocity level perpendicular to the plating at the foundation of main and auxiliary engines, gear, pumps, etc. and in the hull plating above the propeller. Semi-empirical formulae for the prediction of these velocity levels have been formulated by Janssen and Buiten /9/. Plunt /10/ has later indicated that the velocity level induced by main engines is well correlated to the horsepower and rpm of the engines. The importance of the impedance of the engine foundation is now also being investigated. So far the reliability of the final results - i.e. the noise levels in the accommodation spaces - is however much improved if the input data are based on direct measurements. Access to a data bank is therefore essential in order to make a sufficiently accurate noise prediction today.

For the determination of the structure-borne sound induced by propellers the existing empirical formulae are less satisfactory /11/ than those for engines. However, a combination of the prediction formulae discussed in the references /9/ and /12/ yield the best agreement found so far between calculated and measured velocity levels induced above the propeller. The result can be written as:

$$L_V(\text{propeller}) = C + 10 \log (M \cdot N) + 40 \log D + 30 \log \omega - 20 \log f \qquad (6)$$

where M is the number of propellers, N number of blades/propeller, D the propeller diameter, ω the rpm and f the frequency. C is a constant. Work is, however, in progress for the investigation of the importance of such parameters as wakefield, pitch, tip clearance, plate dimensions, etc.

The relative importance of the main sources has been discussed in ref. /13/. The result is summarized in Fig. 2.

In certain cases it is possible to reduce the noise level induced by a source. For the auxiliary engines this can be achieved if these are resiliently

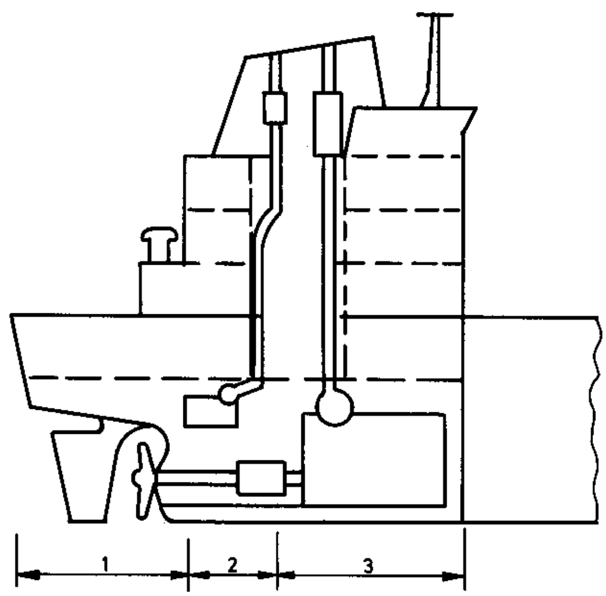

Fig. 2. The relative importance of noise sources as function of frequency.

Source	1	2	3
Propeller "Non-cavitating"	32 - 2000 Hz	32 - 500 Hz	
Propeller Cavitating	32 - 2000 Hz	32 - 2000 Hz	32 - 2000 Hz
Medium Speed Diesel Engine	250 - 2000 Hz	250 - 2000 Hz	32 - 2000 Hz
Slow Speed Diesel Engine		32 - 500 Hz	32 - 500 Hz

mounted. In ref. /14/ the possibility of mounting the main engines resiliently is discussed. Propeller-induced structure-borne sound can on small ships be reduced by means of a compliant layer on the hull above the propeller /15/. Visco-elastic layers at the foundation of small sources such as pumps can yield very good results /16/.

PROPAGATION

The structure-borne sound induced in a ship structure by propeller, main and auxiliary engines is transmitted as longitudinal, torsional, transverse and flexural waves. These wave types are coupled. If for example a pure flexural wave impinges on a junction, other wave types are induced. At the receiving end of the transmission line only the flexural waves need to be considered. The reason is of course that other wave types are weakly coupled to the sound field in a room.

Measurements reported by Kihlman and Plunt, /17/ and /18/, indicate that the velocity levels perpendicular and parallel to the shell can be of the same magnitude. If this inplane motion of the shell is caused by longitudinal waves travelling in the vertical direction, these waves - rather than the flexural waves - would determine the velocity level perpendicular to the plating of a deck. On the other hand, if the inplane motion is determined by transverse waves induced by the relative vertical motion of the frames, then the flexural waves would be of the greatest importance. The magnitude and the direction of the energy flux of the various inplane waves remain to be fully investigated.

If all wave types were to be included in a propagation model the resulting computations would be very complex. In general it is therefore assumed that one wave motion dominates and determines the power flow in the structure.

One of the first propagation models /9/ was developed by Janssen and Buiten in 1973. This semi-empirical method was originally developed for passenger ships. In the model it is assumed that the attenuation of structure-borne sound, induced by a source on the tank top, is 5 dB for each of the first four decks or platforms. For the subsequent decks the attenuation is set to be 2 dB/deck. For certain types of ships the model yields very good or acceptable results at least as regards the resulting dB(A) levels in the accommodation spaces. However, for other classes of ships very large discrepancies between measured and predicted velocity levels have been observed. Full-scale measurements of the attenuation of structure-borne sound have been reported by Buiten /19/, Kihlman and Plunt /17/ and Suhara /20/ among others. These results indicate that the attenuation of structure-borne sound over a discontinuity like a deck is a function of frequency and also of the geometry of the steel structure.

The propagation of flexural waves in a plate coupled to parallel frames has been investigated by Fahy and Lindquist /5/. In the paper it is concluded that the frames act like wave guides, i.e. the power injected into a plate element is mainly contained by the frames. Above the cut-off frequency the energy flux in the plate is much greater than in the frames. This conclusion is confirmed in /6/.

The transverse bending motion of a plate connected to a frame is not possible without longitudinal waves being induced in the plate. This is discussed in /6/. The velocity level of the inplane waves is however much lower than that of the flexural waves. This implies that the velocity level of the inplane waves in a ship structure is not, as sometimes proposed, caused by the bending of the entire structure.

The Statistical Energy Analysis (SEA) /21/ is often suggested to be a suitable method for the prediction of the transmission of a structure-borne sound in ships. The SEA method is, however, generally used on structures less complicated than a ship. Further, the applicability of the SEA method on more or less periodic structures like a ship has not been fully investigated. The major conditions for the successful application of the SEA method are that the coupled systems are resonant and that the modal densities of each system are sufficiently high. These conditions are not always fulfilled for ship structures. The first resonance for flexural waves in part of the main deck can be as high as 200 Hz. The modal density for longitudinal waves is even lower than for flexural waves.

Jensen /22/ has applied the SEA method to calculate resulting velocity levels in a scale-model of a narrow section of a ship. Except in the low frequency range predicted levels show good agreement with measurements. However, no comparisons are made with full-scale measurements. In /22/ only flexural waves are considered. The fair agreement between measured and predicted levels implies that the longitudinal waves are of less significance than suggested by Kihlman and Plunt, /17/ and /18/. In /22/ Jensen assumes that the structure is built up of plate elements. The dimensions of these elements are determined by the dimensions of the scale model. Any possible effects due to the frames are disregarded. The mathematical model is not immediately applicable to full-scale ship structures.

Sawley /23/ has used the SEA method to study noise and vibration problems in a small motor vessel. Sawley attributes the good agreement between measurements and calculations to the fact that the distance between source and receiver was sufficiently small so as to minimize significant errors. The main sources of error are said to be the coupling loss factors for the structures.

Chernjawski and Arcidiacono have in /24/ discussed the SEA method for the evaluation of the transmission of structure-borne sound, i.e. flexural waves in ship structures. No applications to practical problems are mentioned.

Another example of the use of the SEA method has been reported by Gibbs and Gilford /25/. The sound transmission in a 1/4 scale model of a concrete structure is investigated. A comparison between theoretical and experimental results indicates that the energy of a plate element is mainly determined by flexural waves. For low frequencies, measured values were generally lower than predicted. The reason is said to be the break-down of the SEA method in this frequency range.

The finite element method (FEM) has been used with great success on problems concerning low frequency vibrations of ships. This method can not directly be extended to comply with high frequency problems. Besides, even in the very low frequency range the cost of a FEM calculation is very high.

An analytical method to determine the vibrations of grillages is discussed by Heckl in /26/. The technique is discussed further in /27/. The displacement of each element in a grillage is determined by the boundary conditions. At each junction the conditions concerning continuity and equilibrium must be fulfilled. The result is a system of equations which readily can be solved by the use of a computer at a minimal cost. The application of this method to the problem concerning the transmission of structure-borne sound in a superstructure is discussed below.

PROPAGATION MODEL

A ship is a rather complicated structure. It is therefore necessary to make certain assumptions concerning the propagation paths and wave fields in order to obtain a simple and useful model describing the transmission of structure-borne sound. Further, it is an advantage first to investigate the transmission problem in a comparatively simple but still representative part of the ship. If the initial and approximate model is verified through measurements it can be enlarged to comprise the entire construction.

This first part of the investigation has therefore been limited to superstructures, typical of medium-sized tankers. The model discussed below is confined to problems concerning the propagation of structure-borne sound in the vertical direction from the main deck and up.

The basic physical model is shown in Fig. 3. A section limited by the frames and the deck dimensions are indicated in the Figure. The propagation paths such as bulkheads, pillars and staircases inside the structure have been disregarded. It is assumed that the total power flow to the structure is known. The kinetic energy of the decks is to be determined as functions of this input power. The result will thus also yield the velocity level differences between the deck plates.

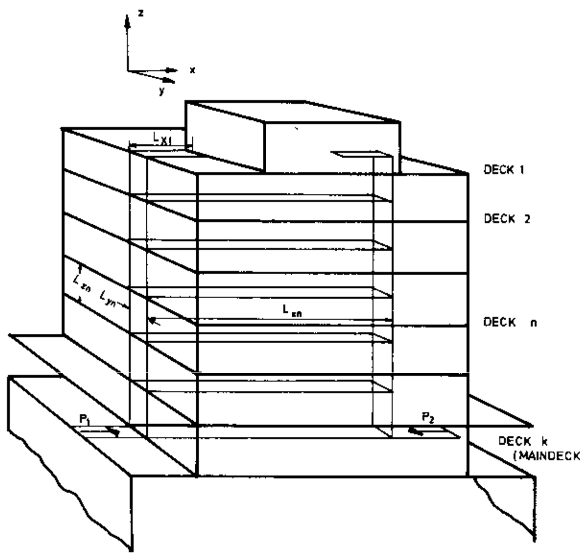

Fig. 3. Model of superstructure.

Four cases have been considered. The power flow from one deck to another is determined by either flexural or longitudinal waves. For each wave type it is assumed that the construction can be approximated by either a wave guide or a beam model. For the wave guide model it is assumed that the waves propagate in the plate elements limited by the frames and decks. The coupling between the plates is caused by bending moments at each junction. The translatory motion of the frames and junctions is neglected. The junctions are allowed to rotate parallel to the y-axis. See Fig. 3.

In the second case - the beam model - the beam is considered to be built up of the frames and part of the plate elements as suggested in reference /6/. The mathematical descriptions of the four cases are analogous. The case concerning flexural waves propagating in a wave guide model is summarized below.

Consider one plate with the dimensions L_x and L_y in the x and y directions respectively. See Fig. 3. The displacement of the plate is assumed to be zero along the edges $x = 0$, $x = L_x$ and $y = 0$, $y = L_y$. The plate is excited by bending moments at the edges $x = 0$ and $x = L_x$. The differential equation for flexural waves on the plate element is

$$\nabla^2(\nabla^2 W) - \kappa^4 W = 0 \qquad (7)$$

In this equation, $W(xy)$ is the displacement in the z-direction. The wave number κ is complex to account for the losses and can be written as:

$$\kappa = \kappa_o(1+ i\eta/4) \qquad (8)$$

where κ_o is the real part of the wave number and η is equal to the total loss factor for the plate.

If the bending moments at the edges $x = 0$ and $x = L_x$ are assumed to be constant along the boundaries, then the lowest mode in the y-direction primarily determines the displacement of the plate. Let the shape of this lowest or first mode be described by the function $g(y)$ and let the corresponding eigenvalue be k_1. If the edges of the plate along the frames are assumed to be clamped a complete solution of the displacement of the plate can not readily be derived. An approximate solution can be obtained by representing the mode shape by a simple sine function with an eigenvalue within .4% of the correct value. For simply supported edges however, the motion of the plate, and thus also the function $g(y)$, are well defined.

The kinetic energy of the plate is mainly determined by the motion of the centre part of the plate. Calculations indicate that the choice of boundary conditions is of little or no significance for the final results for frequencies above 200 Hz. In the low frequency range the best agreement between experimetal and calculated results is obtained for the clamped condition. Thus;

$$g(y) = \sin\left(\frac{3\pi y}{2L_y} - \frac{\pi}{4}\right)$$

$$\text{for } y \geq \frac{L_y}{6} \text{ and } y \leq \frac{5L_y}{6} \qquad (9)$$

$$k_1 = \frac{3\pi}{2L_y}$$

Considering the discussion above, the displacement W of the plate can be written as:

$$W(xy) = w(x)\ g(y) \qquad (10)$$

Eq. (10) inserted in (7) yields the differential equation for the function w:

$$\frac{\partial^4 w}{\partial x^4} - 2k_1^2 \frac{\partial^2 w}{\partial x^2} + k_1^4 w - \kappa^4 w = 0 \qquad (11)$$

The general solution to eq. (11) is for $\kappa > k_1$ given by:

$$w = A_1 \sin\kappa_2 x + A_2 \cos\kappa_2 x + A_3 \sinh \kappa_1 x + + A_4 \cosh \kappa_1 x \qquad (12)$$

where:

$$\kappa_2 = |\kappa^2 - k_1^2|^{\frac{1}{2}}$$

$$\kappa_1 = |\kappa^2 + k_1^2|^{\frac{1}{2}} \qquad (13)$$

The displacement is zero at the end parts. The amplitudes $A_1 - A_4$ in the expression (12) can consequently be determined if either the bending moments or angular displacements at the edges are known. In the latter case the boundary conditions become:

$$w = 0\ ;\ \frac{\partial w}{\partial x} = \gamma(0) \quad \text{for } x = 0$$

$$w = 0\ ;\ \frac{\partial w}{\partial x} = \gamma(L_x) \quad \text{for } x = L_x \qquad (14)$$

Based on the equations (12) and (14) the displacement and consequently also the bending moments can be expressed as functions of the angular displacements at the edges. The resulting bending moments at the ends of the plate element can be written as;

$$M(O,y) = D\kappa_2\ [\gamma(0)F_1 - - \gamma(L_x)\ F_1]\ g(y)$$

$$M(L_x,y) = D\kappa_2\ [\gamma(0)F_2 - - \gamma(L_x)\ F_1]\ g(y) \qquad (15)$$

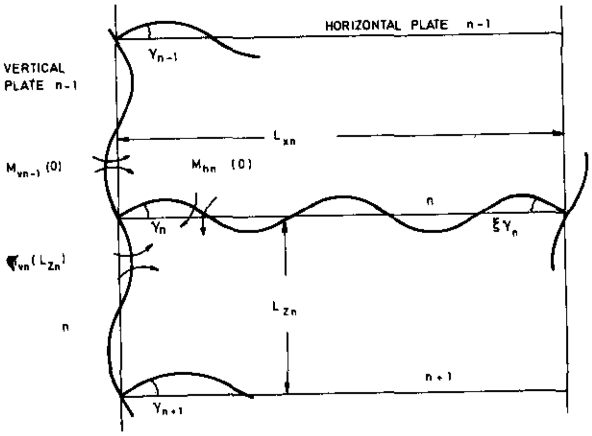

Fig. 4. Resulting bending moments and *angular displacements at a T-*junction.

For frequencies well above the cut-off frequency for the first propagating mode the functions F_1 and F_2 become:

$$F_1 = 1 - \tan\alpha_2$$
$$F_2 = 1/\cos\alpha_2 \qquad (16)$$

where:

$$\alpha_2 = \kappa_2 L_x \qquad (17)$$

The resulting bending moments at a T-junction are shown in Fig. 4. $M_{v,n-1}(0)$ denotes the bending moment at the lower end of the vertical plate element n-1. $M_{vn}(L_{zn})$ is the corresponding moment at the top end of the vertical plate n. $M_{hn}(0)$ describes the moment at the end x = = 0 of the horizontal plate element n.

The condition for equilibrium at a T-junction is:

$$M_{v,n-1}(0) + M_{h,n}(0) = M_{v,n}(L_{zn}) \qquad (18)$$

Considering now the equilibrium of the bending moments at all the junctions in the structure, a system of equations relating all the angular displacements can be derived. The angular displacements can be solved as functions of a real or fictitious power input to the structure.

The space average of the velocity squared of the horizontal plate element discussed above can in the high frequency range be written as;

$$\langle v^2 \rangle = \frac{\omega^2 \kappa_2 [|\gamma(0)|^2 + |\gamma(L_x)|^2]}{6 L_x \eta \kappa^4} \qquad (19)$$

The velocity level difference between two plate elements can thereafter be calculated.

For the case of simple box like superstructures complete expressions are derived in reference /7/. A more general formulation is presented in ref. /28/.

FULL SCALE MEASUREMENTS, PROPAGATION

Velocity level and reverberation time measurements were made on two laid-up medium-sized (130.000 tdw) tankers. The configuations of the superstructure are indicated in Figure 3. The velocity level measurements were made with the auxiliary engines as noise sources in otherwise quiet ships. The reverberation time measurements were made with the use of a vibrator. During the measurements the accelerometers were on all decks mounted at the same positions relative to the sides of the deck house and the frames. All measurement positions were in the same vertical plane, aft of the bridge wings.

The velocity level differences between the decks in the superstructures on the two ships, A and B, were first calculated according to the longitudinal wave models discussed above. Based on these models the resulting velocity level differences were far too small as compared to the measured results. The differences could be of the order 20 dB. The results indicate that a pure longitudinal wave model over-estimates the power flow from the main deck up and into the structure.

The results obtained from the flexural wave model, assuming a wave guide propagation pattern, show far better agreement with measurements.

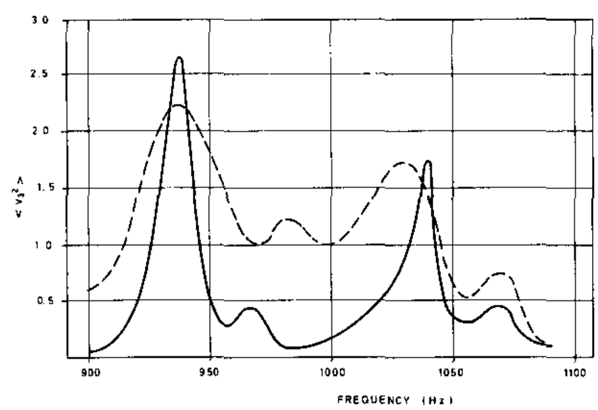

Fig. 5. Calculated (-) and measured (---) velocity squared $\langle v_3^2 \rangle$ for deck 3 on ship B. The band-*widths for the curves are not* the same.

The influence of the periodic character of the structure on the transmission process is indicated in Figure 5. In this Figure the predicted velocity, or rather the quantity $\langle v_3^2 \rangle$, for the third deck on ship B is shown as function of frequency within the 1 kHz 1/3 octave band.

A narrow band analysis of the measured quantity $\langle v_3^2 \rangle$ within the 1 kHz 1/3 octabe band is shown in the same Figure. As before the velocity refers to the third deck on ship B. The form of the peaks in Figure 5 depends on the bandwidth of the analysis and also on the frequency spectrum of the input power. The theoretical curve and the measured curve can therefore not be compared directly. It is, however, quite clear, that the velocity has two distinct maxima at approximately 930 and 1030 Hz within the frequency band. Two secondary maxima can also be observed. For a free steel plate of the same dimensions as the horizontal plate 3 in ship B, the number of resonances within the 1 kHz 1/3 octave band is of the order 200. The actual response of a plate element in a supersturcture is thus not directly comparable to the response of a free plate or to the response of any other separate part of the total structure.

Calculated (flexural wave model) and measured velocity level differences between the five lowest decks in the superstructure are compared in Figure 6. Measured and predicted values are all based on 1/3 octave band analyses. The quantity ΔL_{mn} referred to in the Figures is equal to the velocity level difference between the decks m and n. The main or the lowest deck is for both ships referred to as deck number 6. The level differences shown in the Figures are calculated for the case that the plate elements are clamped along the frames.

For ship A the level differences between the main and the upper decks increase with increasing frequencies up to about 200 Hz. In this frequency range the first resonance for the substructures on the main deck starts to be fully developed. The velocity of the main deck is consequently high in this region. This results in a large velocity level difference between the main and the upper decks. Even for higher fre-

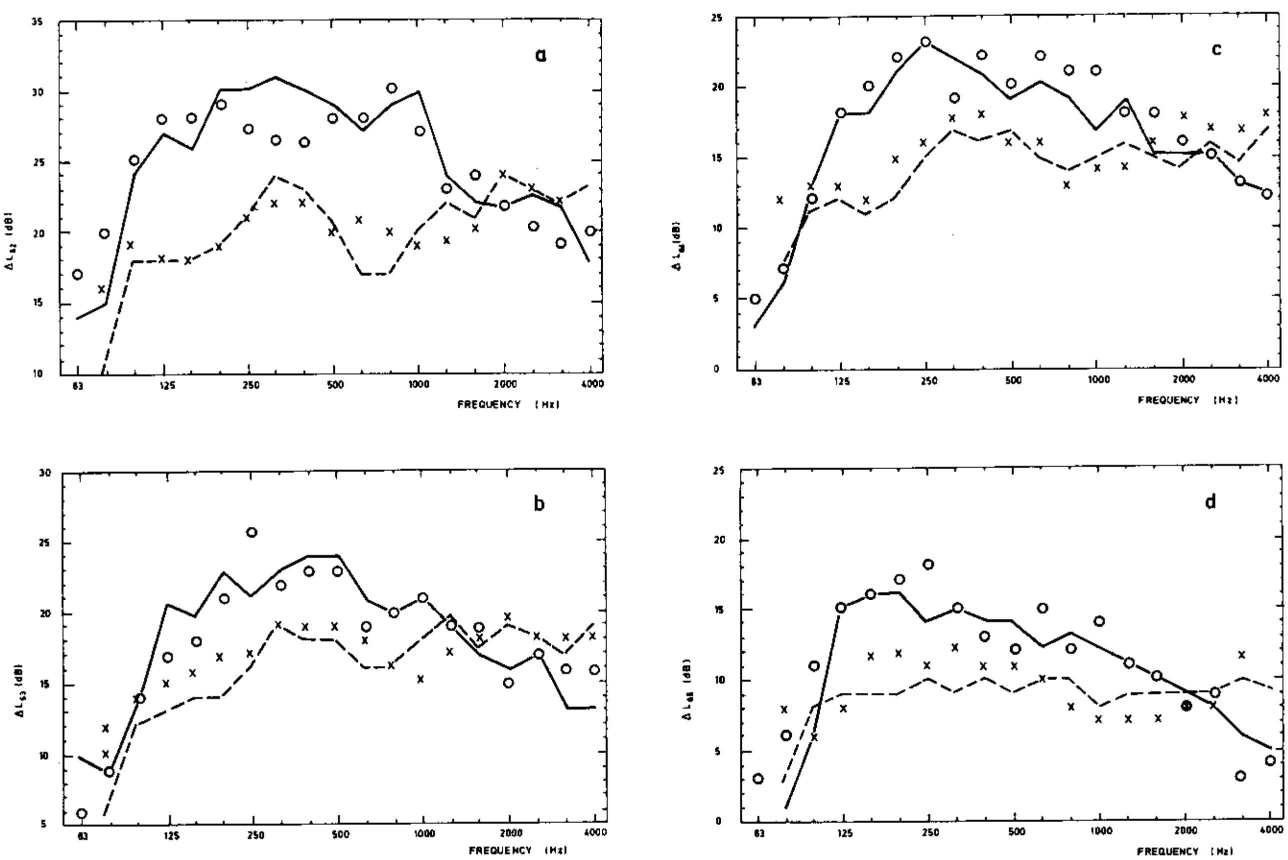

Fig. 6. Velocity level differences between the decks 6 and 2 (a), 6 and 3 (b), 6 and 4 (c), 6 and 5 (d) Compare Fig. 3.

—— predicted - ship A; --- predicted - ship B
ooo measured - ship A; xxx measured - ship B

quencies predictions, based on the flexural wave model, as well as measurements show that the level differences vary considerably with frequency. This type of frequency dependence has previously been attributed to the effects of the longitudinal waves.

The exterior of the superstructures on the two ships are fairly similar. However, the differences between the plate dimensions and deck constructions result in quite different velocity levels. The velocity level difference between the first two decks is up to 7 dB larger on ship A than on ship B. The quantity ΔL_{62} is in the 1 kHz third octave band 30 and 20 dB for the ships A and B respectively. The corresponding level difference on a typical passenger ship would according to /9/ be of the order 8 dB. The examples above indicate that no simple and general rule can be formulated concerning the attenuation of structure-borne sound in a steel structure. The dimensions of the sub-structures must always be considered.

A corresponding investigation concerning the attenuation of structure-borne sound due to frames etc. is presented in ref. /8/. The parameter F discussed above - eq. (1) - is a function of the attenuation in the horizontal and vertical directions.

ATTENUATION OF STRUCTURE-BORNE SOUND

In general nothing is made to increase the attenuation of structure-borne sound in the propagation path between source and receiver. There are however, notable exceptions. Noise levels in superstructures can be effectively reduced if the entire structure is resiliently mounted on the main deck. Grünzweig und Hartmann has at the end of 1977 designed more than 150 of these superstructures. The first resonance frequency for the vertical motion of the simple mass-spring system is typically of the order 4 - 8 Hz. Measurements made by Grünzweig und Hartmann on two sister ships (1599 brt), one with a conventional and the other with an elastically mounted superstructure, indicate that the noise levels in the accommodation spaces on the 1st poop deck can be decreased by 10 dB(A) or more by the resilient mounting. On the deck below the noise reduction is somewhat less.

Full- and model scale experiments with damping layers are discussed in the references /29/ - /31/. In these investigations part of the steel structure between source and receiver were treated with damping layers. The noise reduction in cabins above and away from the treated areas were reported to be insignificant.

Constrained damping layers on steel structures directly facing a cabin can however decrease the noise level in a cabin by 4 - 5 dB(A) as discussed in ref. /32/.

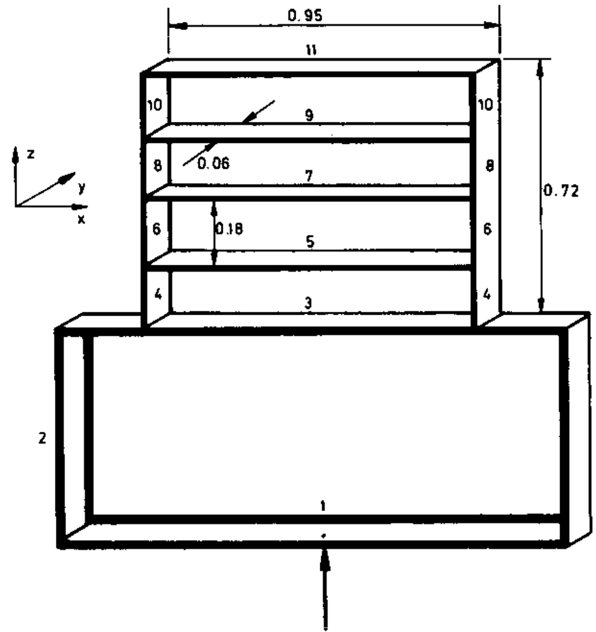

Fig. 7. Scale model for measurements of the attenuation of structure-borne sound. The thickness of the plates 1 - 3 is 1 mm and of the plates 4 - 11 0.6 mm. Other dimensions in m.

Various methods to increase the attenuation of structure-borne sound have been investigated by means of model scale measurements. The basic model is shown in Fig. 7. The width of the model is equal to the distance between two frames. The reason being, as discussed above, that a ship structure can be considered as a wave guide model i.e. the main power flow in the vertical direction is contained by the frames.

The following configurations have been investigated (compare Fig. 7):

1. Bare steel model
2. Damping layers on the plates 4 and 6
3. Damping layers on the plates 5 and 7
4. Damping layers on the centre third part of the plates 5 and 7
5. Damping layers on plate 3 and on the plates between the elements 2 and 4.
6. The plate elements 3 and 4 connected only via the frames (see Fig. 8).
7. The superstructure (plates 4 - 11) elastically mounted on the main deck (plate 3), (see Fig. 9).

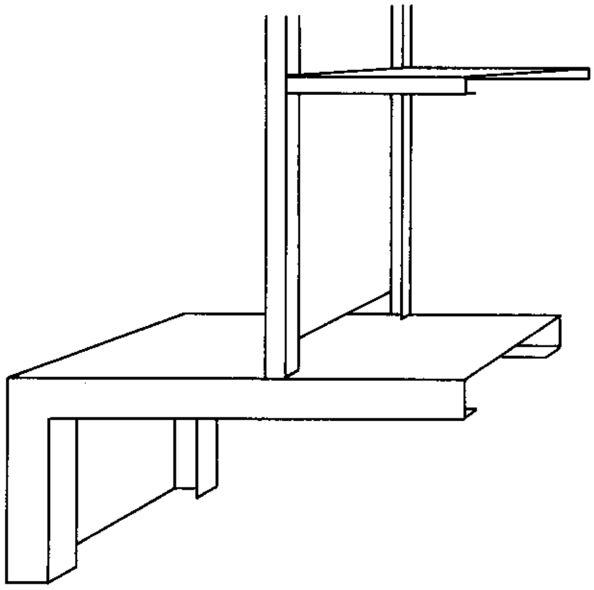

Fig. 8. Scale model 6. The superstructure connected to the main deck via the frames.

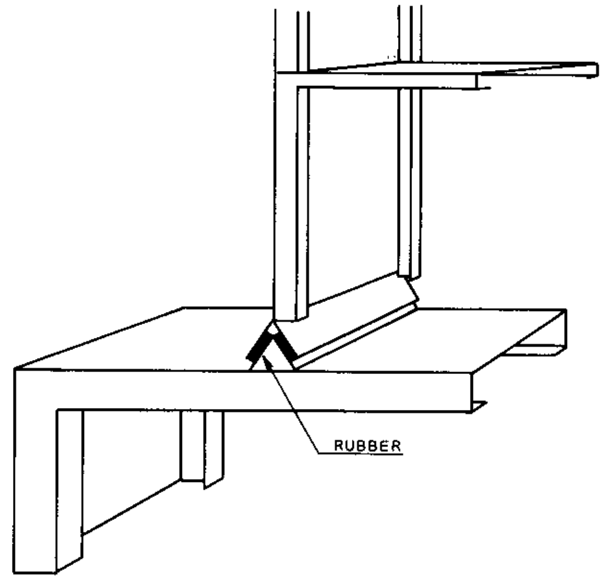

Fig. 9. Scale model 7. The superstructure is elastically mounted on the main deck.

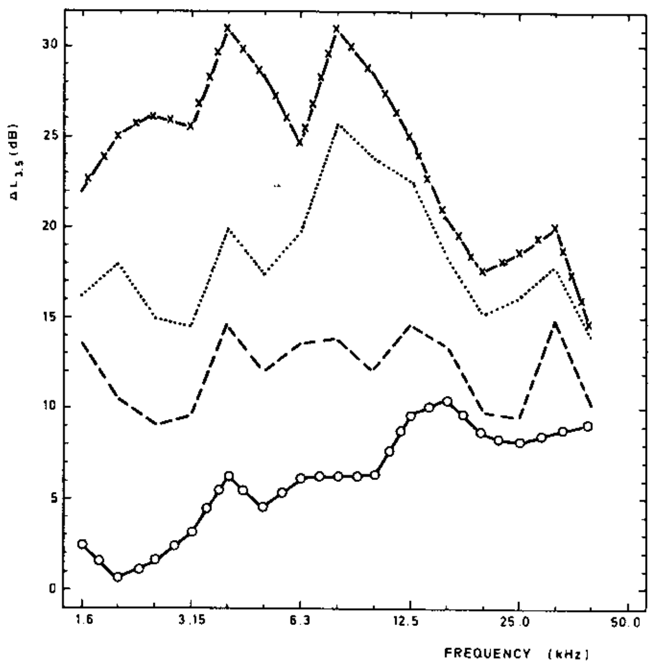

Fig. 10. Measured velocity level differences $\Delta L_{3.5}$ between the plates 3 and 5.

o–o–o Model 1
----- Model 2
x-x-x Model 3
..... Model 4

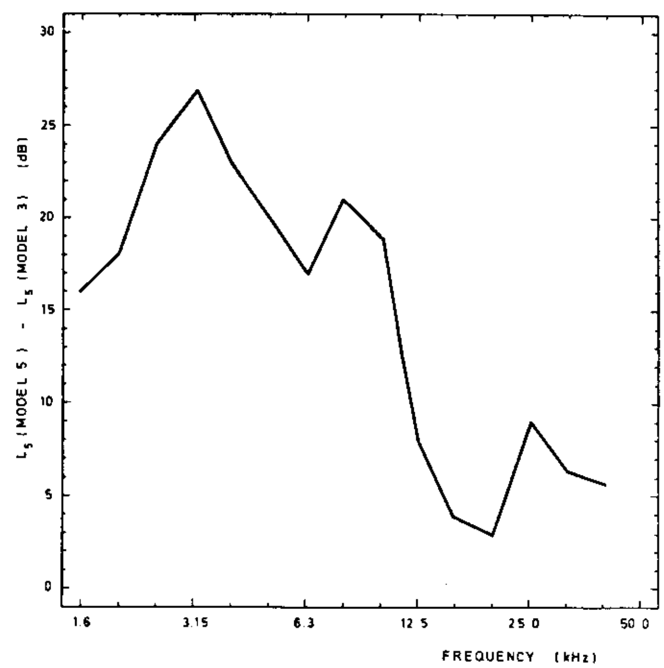

Fig. 11. Comparison between the measured velocity levels of plate element 5 for the models 3 and 5.

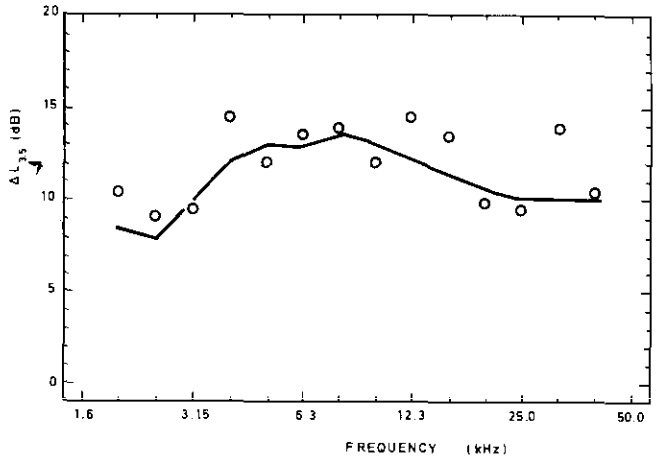

Fig. 12. Velocity level differences between the plates 3 and 5 for model 2.

---- predicted oooo measured

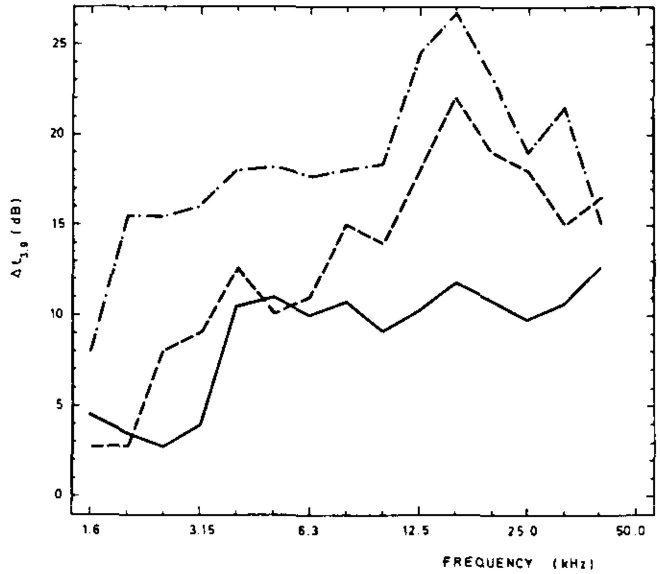

Fig. 13. Measured velocity level differences between the plate elements 3 and 5.

—— Model 1 - - - Model 6
-•-•- Model 7

The loss factors for the untreated steel elements were of the order 0.0008. The visco-elastic layer increased the loss factor to 0.05. The measured velocity level differences $\Delta L_{3.5}$ between the plate elements 3 and 5 (see Fig. 7) are shown in Figure 10 for the models 1 - 4. In Figure 11 the velocity levels of plate 5 for the models 3 and 5 are compared. The input power to the structure was the same in both cases. The velocity levels of the top plate 11 were more or less the same for all models where damping layers were applied. This indicates that the further away the measurement point is from the damped area the smaller is the effect of the damping layer. This is in agreement with the full-scale experiments reported in /29/ and /31/. The highest damping of plate 5 is obtained on model 3 i.e. when the damping layer is applied directly to the plate.

Damping layers on the vertical plate sections increase the attenuation of flexural waves more than of longitudinal waves. Despite this, predictions based on the flexural wave model discussed above compare fairly well with measurements in the frequency range where the model is applicable. This is shown in Figure 12 for model 2.

The loss factor of the treated plates for the models 2 - 5 were increased by the damping layer by the order of 60 times. For a full-scale untreated ship structure the loss factor is fairly high (0.002 - 0.05). The natural losses can not be expected to be increased by more than 5 - 10 times this value, even if constrained visco-elastic layers are applied. This implies that the effect of damping layers on full-scale structures is considerably less than in the scale model.

The resonance frequency for the elastically mounted structure - model 7 - is 140 Hz which corresponds to approximately 7 Hz in full scale. The results for the models 1, 6 and 7 are shown in Figure 13. The measured velocity level differences for model 7 are comparable to the results from the full-scale measurements discussed above. The noise reduction due to the changed boundary conditions, model 6, is of the order 5 dB. Alternatively the boundary conditions between a deck and the vertical plate sections could be changed.

The model scale measurements discussed above are described in detail in reference /28/.

NOISE RADIATION

The resulting sound pressure level in a room is a function of the acoustical power radiated into the room and also of the total absorption in the room. The power radiated by a structure excited by structure-borne sound is a function of the dimensions of the structure, radiation ratio s, the coupling factor ΔL_V between the steel deck and the structure and the velocity level of the deck as discussed above. Data concerning the very important parameters s and ΔL_V can in general not be supplied by the manufactureres of accommodation systems although these parameters determine the acoustic quality of a structure. All acoustical properties of accommodation systems should preferably be measured in situ on board or in a laboratory in a special mock-up or test rig. Fig. 14 shows a test rig which is used for these purposes. The rig is quite simply a section of a ship structure extending from the outer bulkhead to the casing. The dimensions of the deck are 7 x 4.6 sq.m. The structure is excited by a vibrator and the resulting noise levels in the cabins mounted in the rig are measured and determined as functions of the velocity level of the steel deck. Transmission losses for airborne sound can also be measured. A test rig is an ideal set-up for direct measurements of the noise reduction due to a certain modification of an accommodation system. A rig can be used to acoustically optimize a system. Even seemingly small variations of the mounting of bulkheads etc. can reduce the noise level in a cabin.

Noise levels can be effectively decreased by mounting a structure resiliently. The most commonly used application of this principle is floating deck constructions.

On ships floating decks are used to:

i) reduce the acoustic power radiated by the deck,

ii) reduce the structure-borne sound transmitted to bulkheads mounted on top of the floor,

iii) increase the sound transmission loss of the deck.

The following requirements to an effective floating floor must be fulfilled:

1) large velocity level difference between top and bottom constructions,
2) low radiation ratio for the top floor,
3) high noise reduction index.

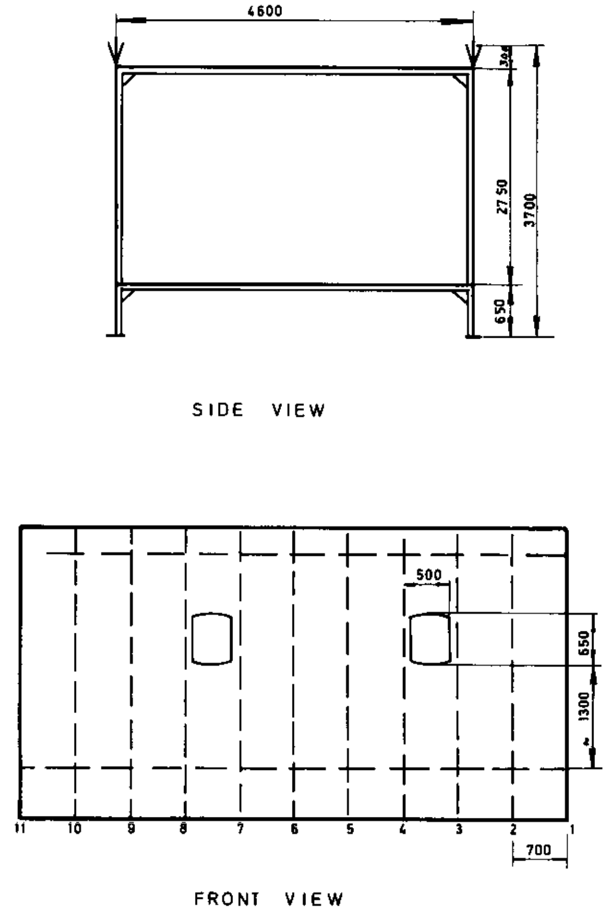

Fig. 14. The DnV test rig for measurements of the acoustical parameters of accommodation systems.

The material and geometrical parameters determining the radiation ratio and reduction index are discussed extensively in the literature. The prediction of the velocity level difference between top and bottom constructions is discussed below and also compared to measurements.

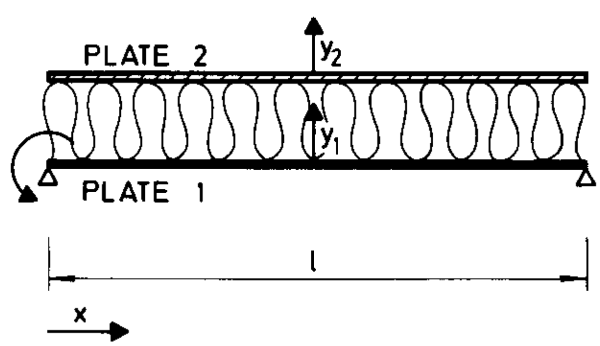

Fig. 15. Model of floating floor.

The basic model is shown in Fig. 15. The bottom plate is assumed to be simply-supported. The plate is excited by a bending moment perpendicular to the supporting parallel beams. The edges of the top plate are free. As a first approximation the dominant motion of the plate is assumed to be in a plane parallel to the beams and perpendicular to the plate. The starting point is thus a simple beam model.

The governing wave equations for the system are:

$$\frac{\partial^4 y_1}{\partial x^4} - k_1^4 y_1 = - \frac{\xi}{D_1} (y_1 - y_2)$$
$$\frac{\partial^4 y_2}{\partial x^4} - k_2^4 y_2 = - \frac{\xi}{D_2} (y_2 - y_1) \qquad (20)$$

The functions y_1 and y_2 describe the lateral displacement of the bottom and top plates respectively. The parameters k_1 and k_2, and D_1 and D_2 are the corresponding wave numbers and bending stiffnesses. The spring constant for the mineral wool layer is denoted by ξ. The parameters k_1, ξ and D are all complex to account for the losses.

The boundary conditions for the one-dimensional case are:

$$y_1(0) = y_1(l) = 0$$
$$y_2''(0) = y_2''(l) = 0$$
$$y_1''(0) = M \qquad (21)$$
$$y_1''(0) = y_2''(0) = y_2''(l) = 0$$

The function M is consequently proportional to a bending moment at the edge x = 0.

The equations (20) and (21) yield a solution for the one-dimensional case.

Approximate results for the two-dimensional case can be obtained by using a variational technique or the method of superposition.

The velocity level difference ΔL_v between the plates is found to be a function of the plate dimensions l_1 and l_2, the spring constant ξ and the loss factor δ for the mineral wool layer, the coincidence frequencies f_{c1} and f_{c2} for the bottom and top plates respectively and of the mass per unit area μ_2 for the top plate. The results also depend on whether the resonant or forced motion of the top plate dominates i.e. whether the top plate is lightly or heavily damped.

A. *Lightly damped top plate*

i) $f_{c1} > f_{c2}$

$$\Delta L_v = 25 \log f + 10 \log \left[\frac{\delta (2\pi)^3 \mu_2 (f_{c1}^2 - f_{c2}^2)\, l_1\, l_2}{\xi c\, f_{c1}^3\, f_{c2}^{\frac{1}{2}}\, (l_1 + l_2)} \right]$$

ii) $f_{c1} = f_{c2}$

$$\Delta L_v = 30 \log f + 10 \log \left[\frac{8c^2\, \mu_2^2 (2\pi)^2}{\xi^2\, f_{c2}\, l_1\, l_2} \right]$$

iii) $f_{c1} < f_{c2}$

$$\Delta L_v = 25 \log f + 10 \log \left[\frac{\delta (2\pi)^3 \mu_2 (f_{c1}^2 - f_{c2}^2)^2\, l_1\, l_2}{\xi c\, f_{c1}^{3/2}\, f_{c2}^2 (l_1 + l_2)} \right]$$

B. Heavily damped top plate;

$f_{c1} \neq f_{c2}$

$$\Delta L_v = 40 \log f + 20 \log \left[\frac{\mu_2\, 4\pi^2}{\xi} \left| \frac{f_{c1}^2}{f_{c2}^2} - 1 \right| \right]$$

Measurements were made on four floating floor constructions. The constructions were;

A) Steel deck plus a layer of 60 mm Rockwool (density 100 kg/qu.m.) and a 22 mm chipboard panel.

B) Steel deck plus 50 mm Rockwool (200 kg/qu.m.) and a 22 mm chipboard panel.

C) Steel deck plus 90 mm Rockwool (100 kg/qu.m.) and a 6 mm steel plate.

D) Steel deck plus 30 mm Rockwool (100 kg/qu.m.) and a 6 mm steel plate.

The plate dimensions l_1 and l_2 were 2.9 and 3.4 m respectively. The results are shown in Figs. 16 and 17.

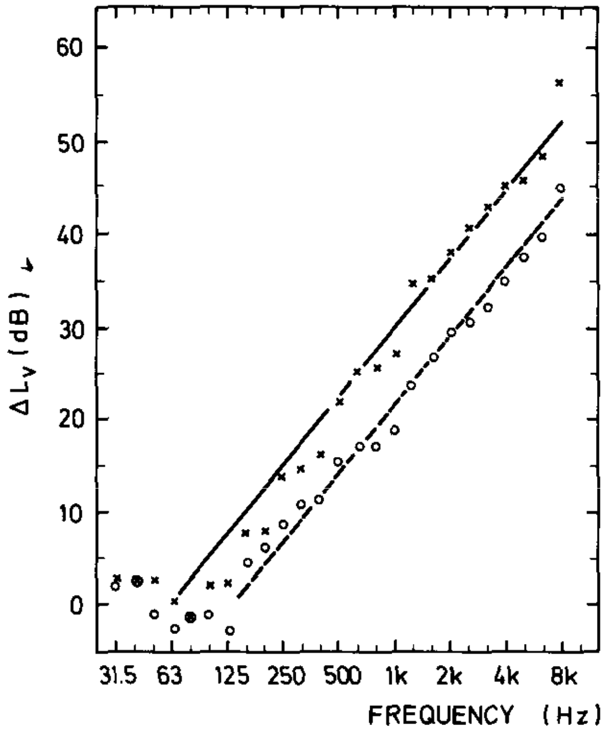

Fig. 16. Velocity level differences between the bottom and top plates. For type A floor solid line is calculated result and x line is the measured result. The dashed line and O line are the corresponding results for the B construction.

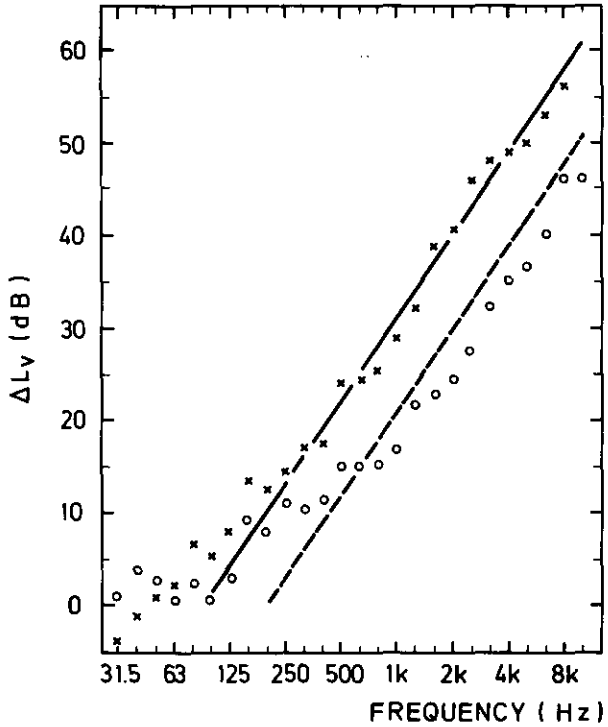

Fig. 17. Velocity level differences between the bottom and top plates. For type C floor solid line is calculated result and x line is the measured result. The dashed line an O line are the corresponding results for the D construction.

For heavily damped top plates ΔL_V increases by 40 dB/ decade. This is in accordance with the impact sound insulation obtained for locally reacting floating floors. For lightly damped top plates ΔL_V can vary between 25 and 30 dB/decade depending on the coupling between the wave fields in the plates. A high level difference is obtained if the top floor is heavy and the elastic interlayer is soft i.e. a thick layer of low density mineral wool. Radiation ratios and sound reduction indices were also measured for the four constructions. Measured values were in accordance with existing theories.

Floating floors are further discussed in ref. /33/. Measurements on some alternative constructions are presented in ref. /34/.

The effectiveness of a floating accommodation system is indicated in Fig. 18 (from ref. /35/). Bulkheads and ceiling were first firmly mounted to the deck and steel structure and the resulting noise level in the cabin was measured. The accommodation system was thereafter mounted on top of a floating floor. Elastic hangers were used for the ceiling. The noise level in the cabin was again measured - for the same running condition as before. The noise reduction due to the alterations is shown in Fig. 18. The total level was decreased by 20 dB(A). For the case discussed above there was no porthole. Openings for windows can reduce the effectiveness of a floating accommodation system considerably.

PREDICTION

The noise prediction programs existing today should be considered as design guides rather than methods to calculate the actual noise level in an accommodation space. The accuracy of a noise prediction could never be expected to be better than that of noise measurements. Measurements made on four sister ships indicate that the resulting dB(A) levels in comparable spaces are within 5 dB(A).

The data or type of information necessary for making a noise prediction

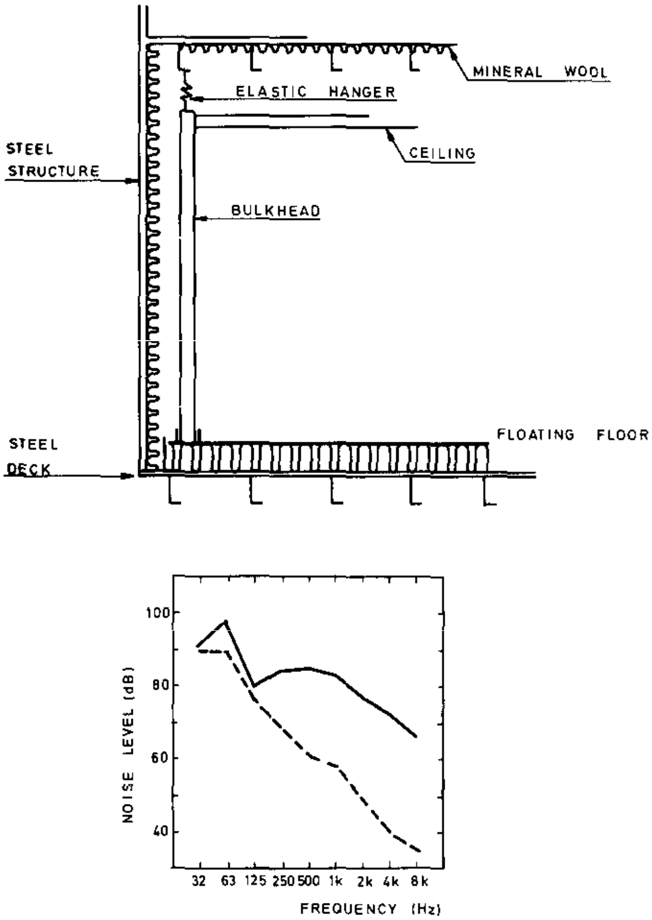

Fig. 18. Measured noise levels in a cabin above the propeller. The solid line is the level when bulkheads and ceiling are mounted firmly to the steel decks. The dashed line is the result for the floating accommodation system shown above.

are:

i) type of engines
ii) type of propeller
iii) description of other sources
iv) steel drawings
v) general arrangement
vi) type of accommodation system and mounting
vii) description of any sound reducing measures

It is assumed that the necessary acoustical parameters for engines, propellers and accommodation systems can be obtained from a data bank or else be estimated. However, predictions based on measurements are always preferred.

The type of information which can be obtained from this type of input data is shown in Fig. 19. (Noise Prediction Program NV590). The result indicates the most dominating source and also the structure in the cabin which radiates the most acoustical power. Based on this type of information proper sound reducing measures can be installed.

CONCLUSIONS

Methods to predict and prevent noise on ships have been improved considerably during the last decade. However, a number of problems still remain unsolved. Some of the most important topics to be investigated are;

i) coupling between the main sources and the steel structure,
ii) prediction models for the description of the power induced by engines and propeller,
iii) description of the propagation of structure-borne sound in the double bottom,
iv) the total attenuation in a ship structure as function of the losses in the vertical and horizontal directions.

Most of these problems are now being investigated within a research program started in Scandinavia during 1977.

Further, more acoustical data on typical sources should be collected and stored in data banks. Complete noise predictions should be compared to full scale measurements.

Despite certain shortcomings the noise prediction methods existing today must be considered as indispensable design tools.

ACKNOWLEDGEMENT

Parts of the projects discussed above have been financially supported by the Norwegian Council for Technical and Scientific Research (NTNF).

CBN CREW CBN 36.0 2.0 27.0 2.2 4.2 2.2 4.0 2.0 3.0 .4

CABIN NO. : 36 CREW CBN

CABIN INPUT DATA DECK NO. : 2 FRAME NO. : 27 LENGTH : 2.20 M WIDTH : 4.20 M HEIGHT : 2.20 M

TYPE OF STRUCTURE IN CEILING : NO. 2 C
TYPE OF STRUCTURE IN FLOOR : NO. 4 F2D
TYPE OF STRUCTURE IN BULKHEADS : NO. 3 BH

RESULTS TOTAL NOISE LEVEL : 66. DB(A) MAIN SOURCE : 10 ME : 65. DB(A)

		OCTAVE BAND (DB)									DB(A)
		1	2	3	4	5	6	7	8	9	
TOTAL RADIATION FROM SOURCES		82.	78.	74.	67.	61.	61.	56.	38.	32.	66.
NO. 10	ME	77.	75.	71.	67.	61.	58.	54.	38.	30.	65.
NO. 11	AE	48.	64.	57.	54.	50.	41.	42.	29.	17.	50.
NO. 12	P	79.	75.	67.	61.	57.	53.	49.	33.	28.	61.
NO. 13	FAN	61.	54.	52.	46.	41.	39.	35.	15.	2.	44.
RADIATED FROM CEILING		77.	74.	67.	61.	56.	52.	43.	34.	27.	60.
NO. 10	ME	74.	71.	67.	61.	56.	51.	43.	34.	27.	58.
NO. 11	AE	43.	57.	51.	46.	43.	32.	29.	23.	12.	45.
NO. 12	P	74.	71.	63.	56.	52.	46.	39.	29.	23.	53.
NO. 13	FAN	56.	50.	46.	41.	37.	30.	25.	12.	0.	37.
RADIATED FROM FLOOR		60.	55.	65.	60.	49.	41.	23.	15.	7.	54.
NO. 10	ME	57.	52.	65.	60.	49.	40.	23.	15.	7.	54.
NO. 11	AE	26.	38.	49.	45.	36.	21.	9.	4.	0.	41.
NO. 12	P	57.	52.	61.	55.	45.	35.	19.	10.	3.	48.
NO. 13	FAN	39.	31.	44.	40.	30.	19.	5.	0.	0.	34.
RADIATED FROM BULKHEADS		80.	78.	70.	65.	59.	60.	54.	36.	30.	65.
NO. 10	ME	77.	75.	70.	65.	59.	59.	54.	36.	30.	62.
NO. 11	AE	46.	61.	54.	50.	46.	40.	40.	25.	15.	50.
NO. 12	P	77.	75.	66.	60.	55.	54.	50.	31.	26.	58.
NO. 13	FAN	59.	54.	49.	45.	40.	38.	36.	14.	0.	44.

Fig. 19. Print-out from the DnV Noise Prediction Program.

The sources are denoted:
ME - Main Engine
AE - Auxiliary Engines
P - Propeller

REFERENCES

/1/ L.L. Beranek
Noise and Vibration Control
New York, McGraw-Hill Book Company, 1971.

/2/ M. Heckl and H.A. Müller
Taschenbuch der Technischen Akustik
Berlin, Springer Verlag, 1975.

/3/ Handbook for Shipboard Airborne "Noise Control"
Coast Guard Technical Publication 073-0-100, 1974.

/4/ J.W.E. Pettersen and J. Storm
Noise Control in Ships
Norwegian Council for Technical and Scientific Research, Oslo 1975.

/5/ F. Fahy and E. Lindquist
"Wave propagation in damped, stiffened structures characteristic of ship constructions".
J. Sound and Vibration 1976, 45(1), 115-138.

/6/ A.C. Nilsson
"Wave propagation in simple hull frame structures of ships"
J. Sound and Vibration 1976, 44, 393-406.

/7/ A.C. Nilsson
"Attentuation of structure-borne sound in superstructures on ships"
J. Sound and Vibration 1977, 55(1), 71-91

/8/ B.S. Seland
"Propagation of structure-borne sound in stiffened plates"
DnV-Report 1978 (Det norske Veritas, Oslo)

/9/ J.H. Janssen and J. Buiten
"An acoustical designing in naval architecture"
Proceedings INTERNOISE, 1973.

/10/ J. Plunt
"Methods for predicting noise levels in ships" Part 17, Report 77-42, 1977 Chalmers Technical University, Gøteborg.

/11/ J.E. Røren
"Propeller induced noise in ships - A survey"
(Text in Norwegian)
DnV-Report 1977.

/12/ N. Brown
"Cavitation noise problems and solutions"
Proceedings International symposium in Shipboard Acoustics, 1976.

/13/ B.S. Seland
"Noise in ships. Relative importance of machinery and propeller" (Norwegian text)
DnV-Report 1977 (Det norske Veritas, Oslo).

/14/ J. Buiten, J.H. Janssen, H.F. Stenhoek and L.A.S. Hagman
"Prevention of noise and vibration annoyance aboard a sea-going passenger and car ferry equipped with diesel engines".
TNO Report 139S, 140S, 1971.

/15/ K. Geicke
"Practical experiences with propeller noise in small craft"
Proceedings International Symposium on Shipboard Acoustics 1976.

/16/ J. Asztely
The uses of visco-elastic materials for noise reduction on ship.
Noise Control, Vibration and Insulation
1977, 297-299.

/17/ T. Kihlman and J. Plunt
"Prediction of noise levels on ships"
Proceedings International Symposium of Shipboard Acoustics, 1976.

/18/ T. Kihlman and J. Plunt
"Structure-borne sound in ships, a study of different wave types"
Proceedings International Congress on Acoustics (ICA 8), 1974.

/19/ J. Buiten
"Experiments with structure-borne sound transmission in ships",
Proceedings International Symposium on Shipboard Acoustics, 1976.

/20/ J. Suhara
"Analysis and prediction of shipboard noise"
Proceedings International Symposium on Practical Design in Shipbuilding 1977.

/21/ R.H. Lyon
Statistical Energy Analysis of Dynamical Systems. Theory and Applications.
MIT Press, Cambridge 1975.

/22/ J.Ø. Jensen
"Calculation of structure-borne noise transmission in ships using the statistical energy analyses approach"
Proceedings International Sumposium on Shipboard Acoustics 1976.

/23/ R.J. Sawley
"The evaluation of a shipboard noise and vibration problem using statistical energy analysis",
Proceedings Stochastic Processes in Dynamical Problems, ASME, New York, 1969.

/24/ M. Chernjawski and C. Arcidiacano
"Simplified method for the evaluation of structure-borne vibration transmission through complex ship structures"
The Shock and Vibration Bulletin 1972, 42, 235-243.

/25/ B.M.S. Gibbs and C.L.S. Gilford
"The use of power flow methods for the assessment of sound transmission in building structures"
J. Sound and Vibration 1976, 49, 267-286

/26/ M. Heckl
"Investigations on the vibrations of grillages and other simple beam structures"
J. Acoustical Society of America 1964, 36(7), 1335-1343.

/27/ L. Cremer and M. Heckl
Körperschall
1967, Springer Verlag, Berlin.

/28/ A.C. Nilsson
"Reduction of structure-borne sound propagating in simple ship structures. Results of model tests" DnV-Report 1978 (Det norske Veritas, Oslo).

/29/ J. Buiten
"Noise reduction on a Rhine-Cruise Ship due to damping meterial"
J. Sound and Vibration 1972, 21(2), 159-167.

/30/ F.H. van Tol
"A model study on the noise reduction effect of damping layers aboard ships"
TNO-Report 133S, 1970.

/31/ A.E. Turner
"The use of damping materials for noise reduction on a passenger ship"
J. Sound and Vibration 1969, 10(2), 187-197.

/32/ T. Kihlman
"Noise reduction with constrained visco-elastic layers, à case history"
Proceedings International Symposium on Shipboard Acoustics, 1976.

/33/ A.C. Nilsson
"Some acoustical properties of floating floor constructions"
J. Acoustical Society of America 1977, 61(6), 1533-1539.

/34/ J. Buiten
"Acoustical investigations of asphaltic floating floors applied on a steel deck"
TNO-Report 152S, 1971.

/35/ P. Klaveness
"Measurement of insertion loss obtained by a floating accommodation system"
DnV-Report, 1977 (Det norske Veritas, Oslo).