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OSHA Noise Levels and the Marine Industry

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OSHA NOISE LEVELS AND THE MARINE INDUSTRY

Abstract

The paper presents various considerations for upgrading the marine industries' approach to engine room noise, including: OSHA's Noise Exposure Limits; suggestions for better planning; and typical values of noise reduction utilizing existing techniques.

BACKGROUND

Back in 1969, when Congress was considering various legislative proposals to deal with occupational safety and health, it was claimed that "in the preceding 25 years nearly 50 million workers had suffered disabling injuries on the job." That was three times as many as the number wounded in *all* the wars this country had fought.

In April of 1971, the Occupational Safety and Health Administration became a reality. OSHA's initial efforts were directed toward accident prevention — hard hats, glasses, safety shoes, guard rails, etc. Of late, they are putting more emphasis on health related subjects such as toxic or cancer-causing chemicals and noise.

Eula Bingham, OSHA Administrator, signaled this trend when in July 1977 she said "We're going to get tough on the health hazards in the work place that cause irreversible injury — cancer, nerve damage, leukemia, lung disease, hearing loss and so on."

The American Petroleum Institute, in 1973, adopted the OSHA *Permissible Noise Exposures* for refinery service machinery (API Standard 615).

The engineering consulting firms that prepare the machinery specifications for the utility industry, consistently require noise levels to meet OSHA.

Even the Navy, whose involvement in noise goes considerably beyond that of industrial plants, is affected by OSHA. In the three year period between 1973-1976, payments to civilian shipyard workers for loss of hearing awards increased fourfold; i.e., from roughly 10 million to 40 million dollars a year. Industry-wide averages place an award for complete loss of hearing at $100,000 — with partial loss awards ranging from $10,000 to $30,000.

This increased interest and activity in, and general awareness of, industrial noise is primarily the result of OSHA's involvement.

OSHA Noise Levels

The OSHA approach to occupational noise is based on permissible exposure; i.e., a specified length of time for a given sound level. Table I illustrates these relationships.

TABLE I

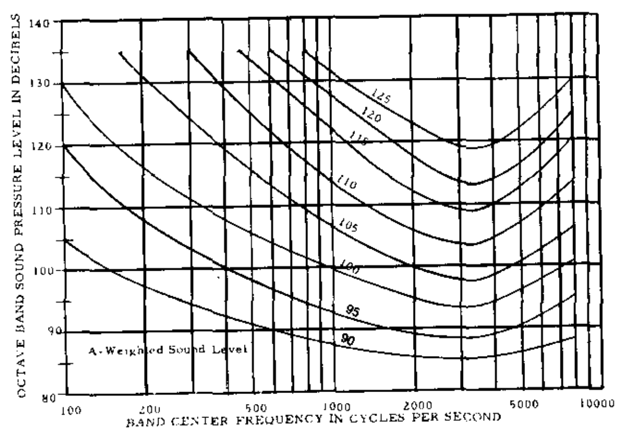
PERMISSIBLE NOISE EXPOSURE

Duration per day, hours	Sound Level dBa
8	90
6	92
4	95
3	97
2	100
1-1/2	102
1	105
1/2	110
1/4 or less	115

It will be noted that these limits are expressed as dBA. This means that it is an overall noise level measurement made on the A-scale (A weighing network) which has the effect of attenuating both high and low frequencies. Said another way, the A-scale permits higher levels at high and low frequencies since these noises are less damaging to the ear.

These shaped or contoured sound levels are shown in Figure I.

FIGURE 1



Equivalent sound level contours

This type of presentation is useful when making spectral analyses and is therefore popular in specification writing. It can also be used for determining a dBA overall from octave band levels, e.g.

> "Octave band sound pressure levels may be converted to the equivalent A-weighted sound level by plotting them on this graph and noting the A-weighted sound level corresponding to the point of highest penetration into the sound level contours. This equivalent A-weighted sound level, which may differ from the actual A-weighted sound level of the noise, is used to determine exposure limits from Table I."

Table II shows the octave band levels equivalent to 90 dBA in the more traditional manner.

TABLE II

Octave Band Center Frequency, Hz

	31.5	63	125	250	500	1000	2000	4000	8000
dB	___	107	102	97	92	88	86	85	87

Note that the octave bands are identified by center-band frequency, rather than the obsolete "Commercial Octave Band" designation of frequency range. This is consistent with the filter specifications used in up-to-date instruments.

Marine Industry

Even though OSHA's noise levels have been widely accepted by American industry for machinery specifications, there has been little, if any, sign of their adoption in the marine field. There are probably several reasons for this. Marine people, in general, tend to be very conservative. The economic climate has been less than robust over the past several years, which also tends to retard change. Jurisdictional disputes may play a role. And, finally, Marad's "Standard Specifications for Ship Construction" — which have a great influence in these matters, makes no recognition of OSHA. With all due respect to the Maritime Administration, their current specifications (December, 1972) contain sections that could benefit by an update.

Any change in this situation will probably be slow in coming, but there's nothing to prevent us from considering how it might best be accomplished.

The most important thing needed is better planning. This must begin with the owner and design agent and include:

1. A better understanding of the many complexities of acoustics, including measurement and suppression.
2. Acknowledgment of practical minimums in terms of noise generation at the source.
3. Recognition of the probable need for barriers and/or other noise reduction techniques.

Complexities of Acoustics

To expand on the above — consider first, some of the peculiarities of noise. Since airborne acoustical measurements are without contact between the instrument and the source of sound, the environmental influence can become critical. The compression waves generated by the sound source interact with any and all surfaces and obstruction. The resulting energy loss or reflection can change the measured sound pressure level many orders of magnitude. It is altogether possible to get changes of 3,000 per cent by varying the environment alone. Such a variation expressed in acoustical notation would be over 14dB.

In addition, sources do not radiate sound equally in all directions. This results in large variations caused by such things as source geometry, radiation efficiency, maximum energy areas and others. Variations due to source directionality are frequently in the order of ± 5 to 20 dB.

The instrumentation used is poor at best, when one considers typical laboratory accuracies in other disciplines. A Class II sound level meter, which is the recognized basis for compliance with U.S. federal noise level standards has, at best, an accuracy of ± 3 dB. This means that it reads the airborne pressure changes to a tolerance of plus 100 per cent, minus 50 per cent. Class I, or precision sound level meters, are good to ± 1.5 dB.

To complicate the problem further, certain other conditions arise in, for example, reverberant spaces. The creation of standing waves can change the level at any given spot in the area 15 to 20 dB as compared to other locations.

What does all this mean in terms of better planning? It means, for example, that consideration should be given to the environmental influence. No one is likely to suggest that an engine room be designed as an anechoic chamber. But, on the other hand, a look around actual installations gives the impression that no one ever considered the use of non-reflective or energy absorbing surfaces. There is considerable potential for noise reduction using such materials.

Regarding instrumentation, specification sheets should address themselves to the type; the calibration of; and the techniques in using sound measuring equipment. ANSI standards S1.1, S1.13, S1.4 and S1.6 provide good references for this purpose.

On the subject of standing waves (patterns formed by two waves of the same length, i.e., frequency, traveling in opposite directions due to reflection off a hard surface, which therefore represent a resonance caused by the room) — they should be avoided for microphone location. It was interesting to read the instructions for locating microphones in a recent machinery specification for a large merchant service locked train gear. They amounted to seeking out standing waves.

Noise At The Source

Another step toward better planning is an honest assessment of noise generation at the source. (Most of the comments in this paper apply to any machine element, but they are basically directed to gears — both main propulsion and SSTG sets.)

The simple truth is — main propulsion gears designed in accordance with modern day practice and load intensities will not meet OSHA's noise level in the vicinity of the gear unit. This is also true of the gears in SSTG sets, especially when they are designed to withstand torque multipliers of 10 to 12 due to short circuits.

The industry has gotten all the mileage that can be had from platitudes like . . . . . "tooth proportions, pressure angle and helix angle shall be selected to minimize noise" . . . . . and the ever popular "increased accuracy" approach has reached a leveling off point in terms of noise reduction.

If the gear contribution to engine room noise is to be reduced, barriers and/or acoustic treatment will have to be utilized.

Sound Barriers and Acoustic Lagging

The simplest and least expensive form of barrier is the one of the so-called "personnel hearing protective devices." There are several types of ear plugs available that provide average attenuation of 6 to 10 dB. Ear muffs generally provide somewhat greater protection. A few operating engineers, however, are hesitant to use this type device; they consider their ears important diagnostic tools in monitoring the health of their equipment. This may, or may not, be a valid objection. It warrants investigation.

Another type of barrier is the acoustic enclosure. It is by far the most effective tool for noise reduction. Properly designed enclosures — with single walls, 100 mm (4 in) thick — can provide attenuation of mid-range frequencies in excess of 40 dB. They can be used in one of two ways: i.e., enclose the people or enclose the machinery. Enclosing people is usually easier; the box is smaller, it interferes less with maintenance, and lends itself ideally for air conditioning.

On the other hand, enclosing machinery, including reduction gears, is certainly feasible. Gas turbine propulsion plants are prime examples.

Acoustic lagging or sound coatings come in many different forms. My company has tried several of them and, with one exception, was disappointed with the results. One treatment that did yield measurable attenuation involved the use of lead.

We were aware of an "acoustic lagging" technique the Navy had used on main propulsion gears, wherein the entire gear case was covered with a lead fabric. The lead fabric was supported by a layer of foam rubber which, in turn, was supported by the gear case. The Navy approach was to cover 100 per cent of exposed surface — including all bolts, flanges and joints. They reported reductions in noise of 15-20 dB. We were curious to see how much noise reduction could be obtained with a similar treatment, but one in which the bolts, flanges, and joints would be left exposed for convenience during inspections and maintenance.

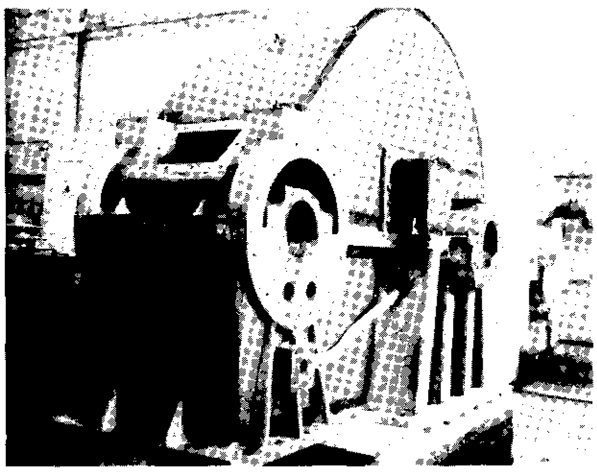
There were a series of 2500 KW SSTG sets going through our shops. The multiplicity of parts gave us an opportunity to conduct a test involving a minimum of variables, while adhering to a fixed time frame for testing.

Specifically, we applied the acoustic treatment to a gear housing — case and cover. We then conducted a full load test on that turbine generator set and recorded noise data. The acoustically treated housing was then removed and an untreated case and cover were installed while maintaining the same rotating parts; i.e., turbine, coupling, pinion, gear, and generator. The only variables, therefore, were the gear casing and time. A second full load test was then run and similar noise data recorded.

The material used was a fiberglass backed, lead/vinyl fabric weighing 7.3 Kg/m² (1-1/2 Lbs/Ft²). It was supported by a 25 mm (1 in.) thick closed cell, fire resistant foam rubber.

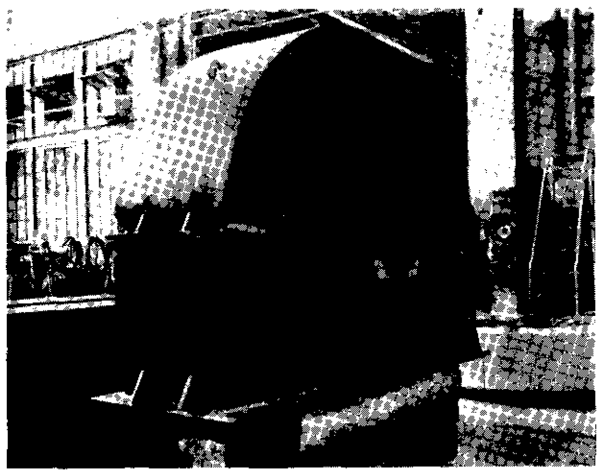
Figures 2 and 3 show the gear case covered with foam rubber before the lead vinyl was attached.

FIGURE 2



Gear Casing — Turbine Side

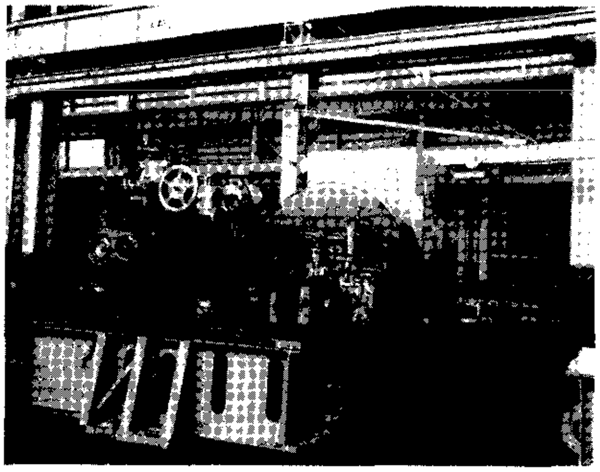
FIGURE 3



Gear Casing — Generator Side

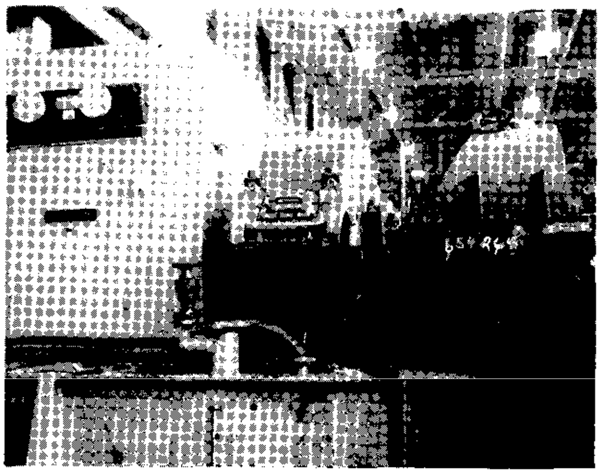
Figures 4, 5 and 6 show the gear unit in place on the bedplate with the finished "acoustic lagging."

FIGURE 4



Ships Service Turbine Generator Set

FIGURE 5



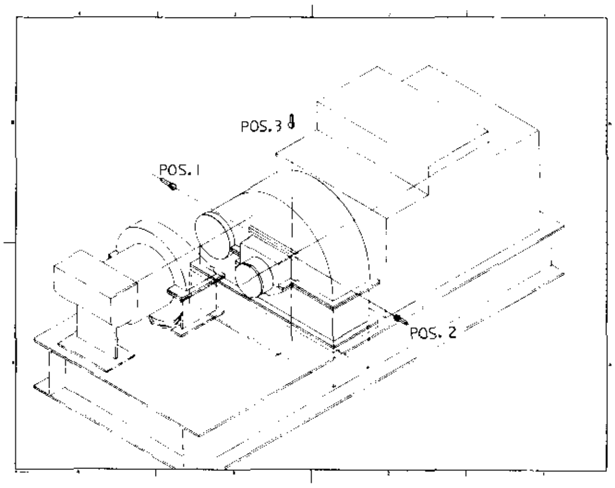
SSTG Set — Side View

FIGURE 6

Close Up of Gear Case

Figure 7 shows a general arrangement of the turbine, gear, generator, and bedplate.

FIGURE 7



Microphone Locations

It also shows three microphone locations. Microphone positions 1 and 2 lay in the plane of the axes of the pinion and gear and were 300 mm (1 ft.) from the gear casing in line with the apex of the two helices. Microphone position 3 was vertically above the bull gear, 300 mm (1 ft.) over the casing and centered on the apex. The 300 mm (1 ft.) distance was used because of space limitations that affected positions 1 and 2. The test unit was on a base located adjacent to and only about 600 mm (2 ft.) from two other bases.

Position 1 was not considered a valid microphone location for several reasons. First — there was a large untreated surface on the top of the bedplate reasonably close to the microphone; secondly, there was an electronic precipitator mounted about 150 mm (6 in.) from the microphone; and third, there was a certain amount of reflected noise from the surfaces of the generator set on the adjacent test stand. We found that moving the position 1 microphone as little as 50-100 mm (2-4 in.) from the prescribed point showed variations of 5 to 10 dBA. There was also some reflection at position 2.

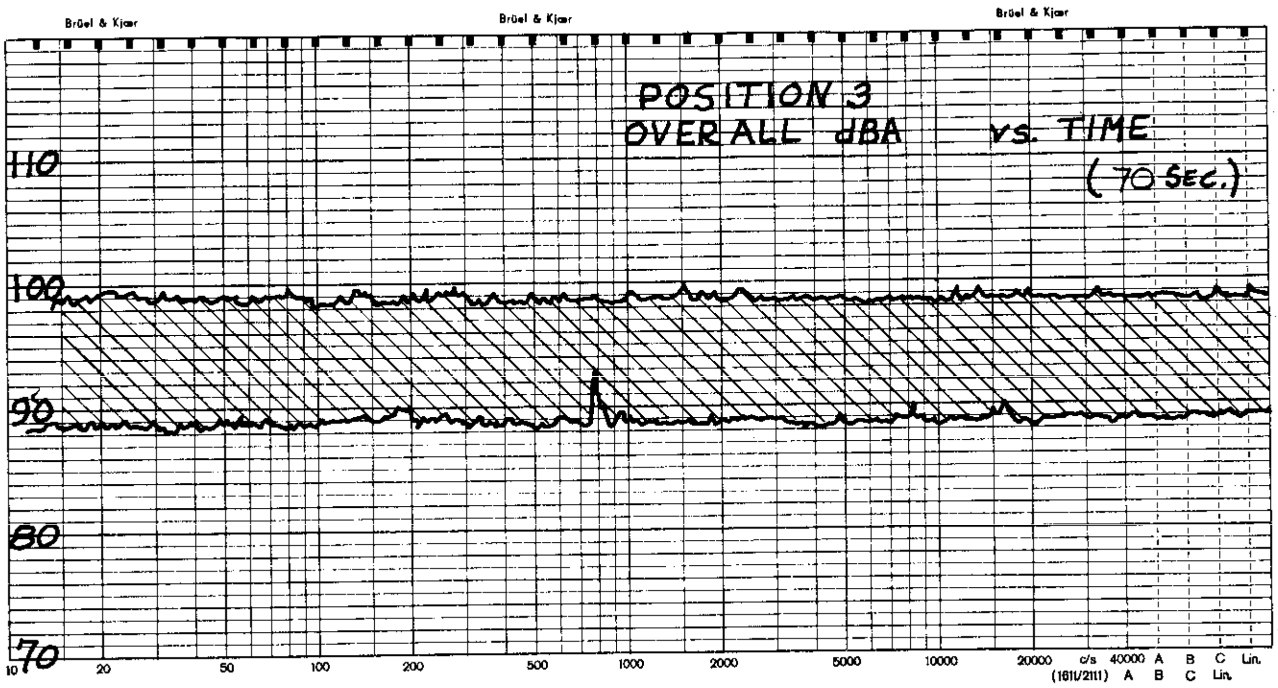
Position 3 was judged to be the best microphone location to measure the change in noise level due to acoustic lagging and, therefore, only those data are presented here.

The predominant frequency in the noise spectrum of most gear units is that of the tooth mesh. In the majority of high speed industrial gears, in ships service turbine generator sets, and in the high speed trains of marine propulsion gears, the tooth mesh frequencies generally fall in the octave bands centered on 2000 and 4000 Hz.

In the case of the second reduction gears of a marine propulsion unit, the tooth mesh note is frequently found in the octave band centered on 1000 Hz.

Figure 8 shows a time study of the overall noise level in dBA.

FIGURE 8

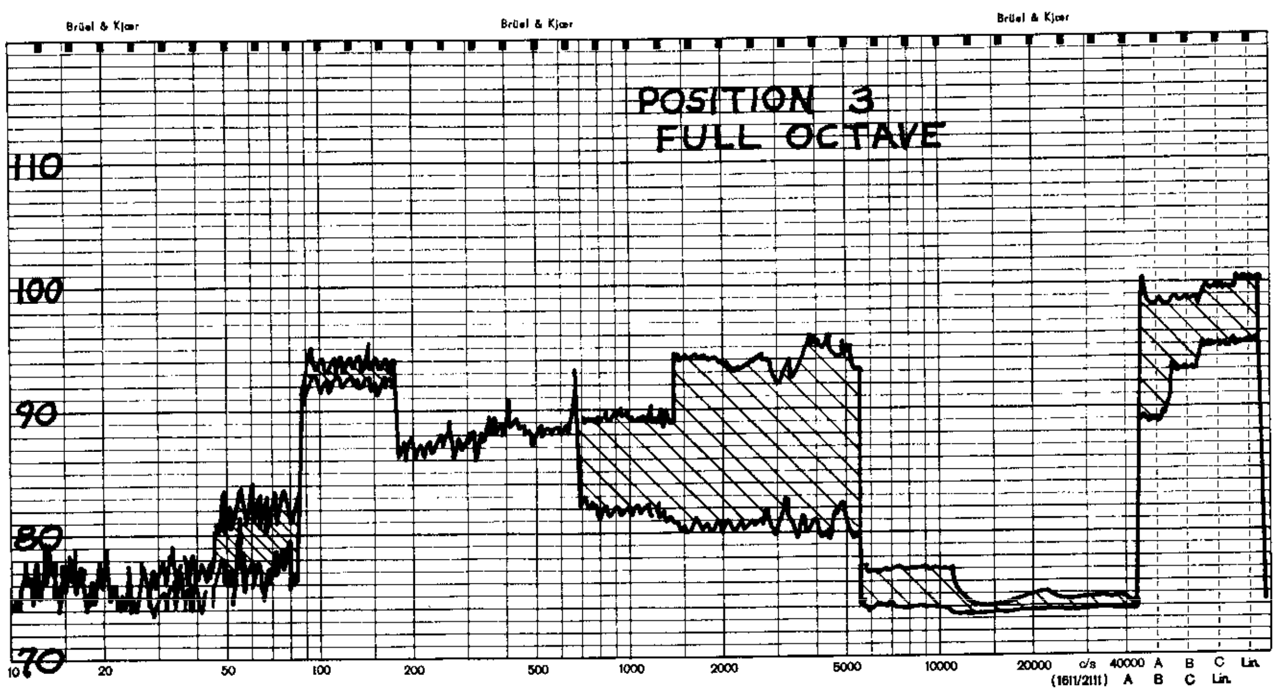


Overall Noise Level

The top line shows the noise level of the untreated housing; the bottom line shows the noise level of the housing with acoustic treatment, and obviously the shaded area in-between represents the reduction in noise attributed to the lead vinyl sheet. The duration of the chart was about 70 seconds and amplitude modulation — at least for that period — was possibly 2 dBA.

Figure 9 is a full octave level recording.

FIGURE 9



Full Octave Levels

There is good agreement between the results on the A weighted networks. It's interesting to note here that most of the attenuation occurred in three octave bands of primary interest; i.e., 1000, 2000 and 4000 Hz.

Figure 10 shows three different real time plots. The first is a third octave on a linear network; the second is a third octave on the A weighted network; and the third is a narrow bank, or discreet frequency chart on a linear network. All of these real time plots represent "peak hold" — meaning that the highest level of any modulating amplitude was recorded.

Examination of these three real time charts gives a pretty good picture of the frequencies that are best attenuated by this particular acoustic treatment. The foam rubber/vinyl lead sheet combination has a natural frequency of about 250 Hz and, interestingly, we see an increase in noise near this frequency.

The primary forcing frequencies of this gear are the two rotationals and the tooth mesh — 20, 150, and 5200 Hz respectively. These can readily be identified on the charts. One half tooth mesh is also identified as a spike.

The results of this experiment can be summarized, based on the data taken at the No. 3 microphone position as follows: the abbreviated coverage of lead/vinyl sheet over foam rubber brought about a reduction in noise at tooth mesh frequency of from 12 to 16 dB, which corresponds to a reduction in overall of from 9 to 10 dBA.

This same treatment — the abbreviated coverage of lead/vinyl sheet — was applied to a 32,000 SHP main propulsion gear. Shipboard measurements, at full power, before and after treatment, revealed the reduction in noise was less than that obtained on the SSTG set. Specifically, the two predominant octave bands, 500 and 1000 Hz, were reduced 3/6 dB and 4/7 dB respectively, which lowered the overall 5/7 dBA.

We believe there are two reasons for the smaller attenuation. First, the predominate frequency in the spectrum of the main propulsion gears — 750 Hz — is much closer to the natural frequency of the lead/rubber spring/mass system than was the case with the SSTG set (5200 Hz). Second, there is a considerably greater structural intimacy between the propulsion gear and hull than is the case with the lighting set, and hence a great deal more radiating surface.

Full Load Tests

There's another aspect of machinery noise that should be reviewed; namely, the time and place for measurement.

SSTG sets are almost always given full load factory tests. With the exception of shipboard environmental amplification — which is beyond the vendor's control anyway — compliance to noise specifications can conveniently be demonstrated on factory test.

Main reduction gears, on the other hand, are rarely factory tested at full load. The required horsepower and test facility (prime mover, load device, foundations and couplings) make factory full load testing not only very expensive, but difficult to find. The back-to-back, or locked-in torque test is at best a poor substitute since half the rotating elements are meshing on their ahead tooth surfaces, half on their astern surfaces, and more than half are running in abnormal locations in their bearings — all of which can affect noise.

The normal factory test at light load — in accordance with SNAME Technical Bulletin 3-8 — is of little value in demonstrating noise level. Realistically, one should plan for levels 15-25 dB higher at shipboard full power.

Who Should Plan

The basic decision on what approach to take in meeting OSHA levels — assuming that's the objective — is one the owner and his design agent should make. Whether it's to be acoustic enclosures, sound absorbing surfaces, suppression coatings, ear plugs, or what have you — it's a decision for the owner, not for the shipbuilder, who is bidding competitively and inviting competitive bids on machinery. It may be necessary for design agents to acquire additional expertise in acoustics to make this work.

The ship's specifications should simply state that noise levels at normally manned duty stations meet OSHA's requirements.

The machinery specification might well require compliance to some noise spectrum that is consistent with the overall approach. In that case, factory noise measurements at full load should be made wherever practical.

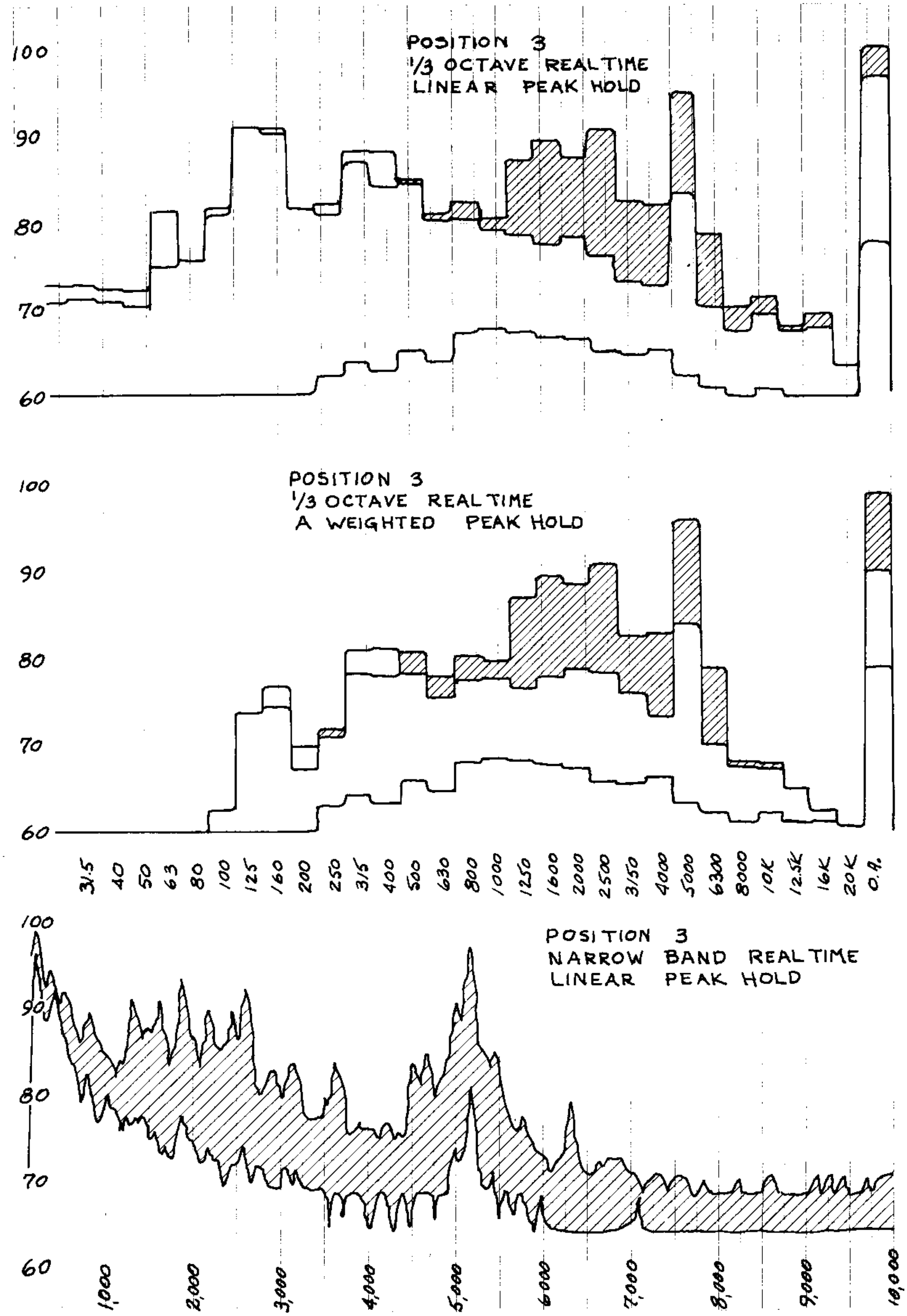
Main propulsion gears are the biggest complication. They're probably best handled with estimated spectrum levels. Measurement during sea trial will help improve estimating accuracy.

Why OSHA, Why Plan?

The best reason to adopt OSHA is protection. Protection of the crew's hearing for one thing and protection of the owner against hearing damage claims for another.

Two reasons to plan. One is — you might achieve what you set out to get. The second is — you might avoid the finger pointing and arguments about responsibility that are almost inevitable without planning.

FIGURE 10



Real Time Spectra