ENCLOSURES TO REDUCE NOISE FROM HEAT PUMPS: FOUR CASE STUDIES

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ABSTRACT

Case studies of partial enclosures to reduce noise from residential air-to-air heat pumps are reported. Several enclosures, most quite similar in design, were built round each of four heat pumps to permit evaluation of the effect of design variables rather than produce an optimum enclosure for each site.

SOMMAIRE

Les résultats de quatre études portant sur des enceintes destinées à réduire le bruit engendré par les pompes à chaleur air-air pour les habitations sont présentés. Plusieurs enceintes de conception semblable ont été construites autour de quatre pompes à chaleur dans le but d'évaluer l'incidence des variables de conception plutôt que de réaliser une enceinte à rendement optimum pour chacun des sites.

INTRODUCTION

In response to a succession of enquiries concerning noise from heat pump units installed in suburban areas DBR/NRCC has carried out a series of case studies. In some instances annoyance was due to noise interference in adjacent outdoor space and in others to noise intrusion in nearby homes. Relocating the outdoor units can often lessen noise problems, but the cost is appreciable and available space may be insufficient to resolve the problem fully. In such cases it seems reasonable to install a barrier or enclosure to reduce noise impact. The use of an enclosure is particularly appealing if the unit is installed between two houses where reflections from building walls would make a simple barrier ineffective.

ENCLOSURE DESIGN

To satisfy the owner of a heat pump, an enclosure should be inexpensive, aesthetically unobtrusive (preferably attractive), without adverse effects on heat pump operation, and effective in reducing noise. Obviously a practical enclosure must involve some compromise when there are so many objectives. Aesthetics could be safely ignored in this study because the enclosures were temporary structures, but the other concerns were taken into account to maximize the general utility of the results.

An enclosure can reduce noise impact by absorbing part of the sound energy and redirecting some of the remainder to minimize that reaching noise-sensitive locations. To achieve this, the walls should be of an impervious material sufficiently heavy to provide negligible sound transmission and the interior surfaces should be lined with an acoustically absorptive material.

In addition, the enclosure must permit essentially unimpeded air flow through the fan and heat exchanger of the heat pump. The general features of the enclosure design are illustrated in Fig. 1 for a unit drawing air in through the sides and blowing it out at the top. This is the most common configuration; conversion of the concept for other air flow patterns is obvious. Air enters the enclosure through one or more inlets and exits through the opening at the top. A separator panel divides the enclosure into inlet and outlet sections to minimize recycling of outlet air back through the heat pump. The inlet and outlet openings may include baffles to block the path from the noise source to the exterior, but diffraction and reflection of the sound waves limit the effectiveness of such baffles. It is wise to cover the inlet and outlet openings with a material such as wire mesh to prevent birds and small animals from entering the enclosure.

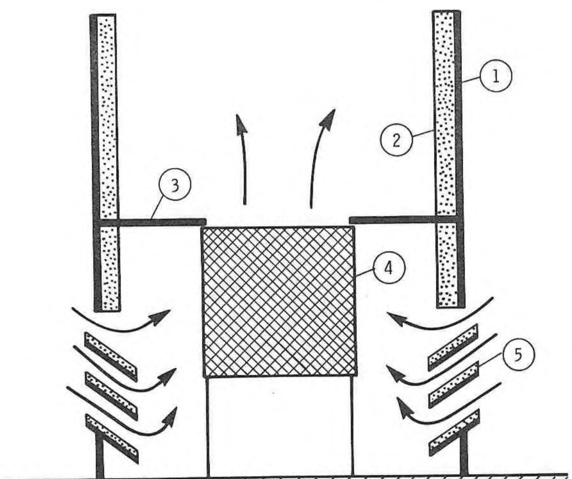

FIG. 1 Schematic representation of acoustic enclosure elements:
(1) impervious wall
(2) absorptive lining
(3) separator panel
(4) heat pump on stand
(5) baffle panel with absorptive surface

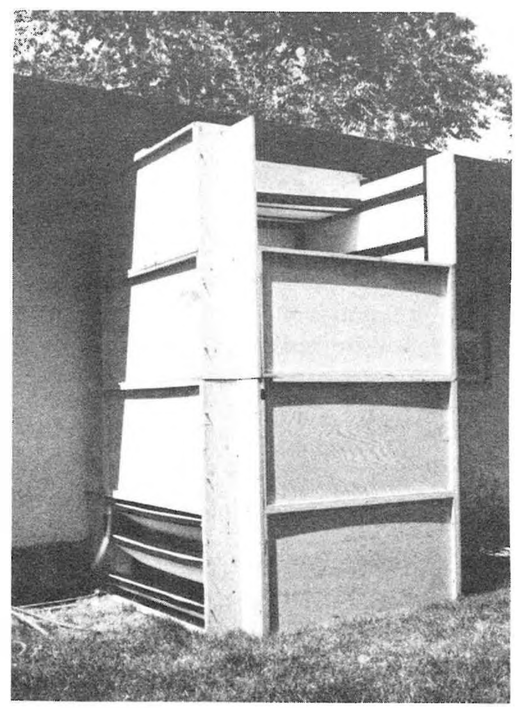

FIG. 2 Enclosure design 1.4

The enclosures for this project were assembled from 0.6 x 1.2 m modules that could be bolted together to form a complete enclosure (Fig. 2). Modular construction offered several advantages: the modules could be readily transported from site to site, enclosure designs could be easily modified, and the sound transmission and absorption properties were the same for all enclosures (and could be tested in the laboratory). There were two types of unit: air flow units (which could include acoustical baffles) and basic panels.

The outer frame of the air flow units was made of 19-mm plywood (Fig. 3). The "baffles," which could be mounted in this frame, were three 1.2 x 0.25-m panels consisting of a central septum of 6-mm thick plywood covered on both sides with 15-mm thick glass-fibre absorptive panels. Inserting the baffles in the frame reduced the cross-section of the openings for air flow by about 40%. The enclosures all had effective inlet and outlet cross-sections of at least 1 m^2. It should be noted that no assessment was made of possible reduction of thermal efficiency or increased strain on the fan due to air flow restriction from the enclosures.

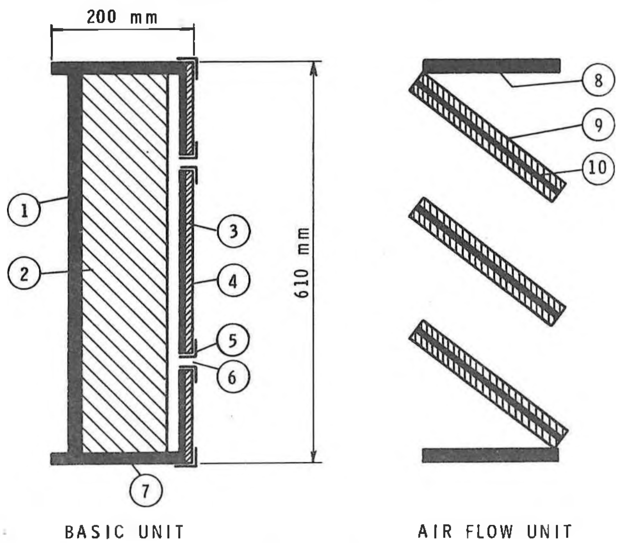

FIG. 3 Schematic cross-section of modular units showing components: (1) 19-mm plywood outer face; (2) 150-mm glass fibre batt; (3) 6-mm plywood; (4) 12-mm glass fibre; (5) protective plastic corner; (6) slit (10 mm wide x 1.2 m); (7) 19-mm plywood frame; (8) 19-mm plywood frame; (9) 15-mm thick glass fibre; (10) 6-mm plywood. These module types are represented symbolically in Fig. 6 by:

Basic panel
Air flow opening
Opening with baffles (outer edge high)
Opening with baffles (inner edge high)

The basic panel units were constructed with an outer frame of 19-mm plywood to which the outer and inner surfaces were fastened, as shown in the cross-section at the left of Fig. 3. The outer surface of 19-mm thick plywood provides the main barrier to sound transmission through the enclosure wall. The inner face of 6-mm thick plywood supports a 12-mm thick glass fibre absorptive layer and provides the other boundary for a resonant cavity to give low frequency absorption. This cavity was filled with low-density glass fibre and the 10-mm width of the two slots on the inner face was selected to tune the absorption resonance to approximately 125 Hz. Plastic corner strips gave clearly defined edges for the slits.

Sound transmission loss for the basic panels was measured according to ASTM E90-81 for a 2.4 x 3.05-m assembly (Fig. 4). The absorption coefficient for the basic panels was measured according to ASTM C423-81 (see solid curve, Fig. 5). A secondary result (with the slits blocked to suppress the resonant absorption around 100 Hz) is shown by the dashed curve in Fig. 5.

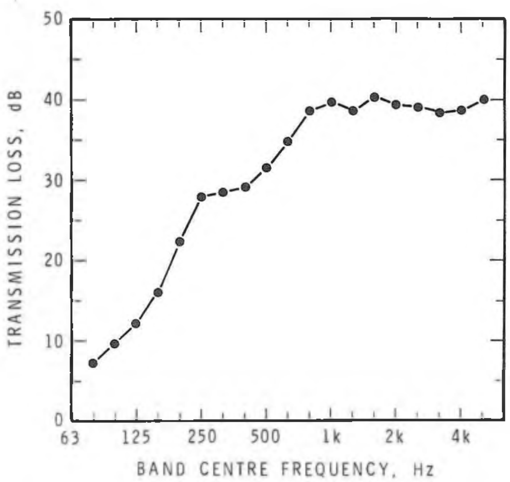

FIG. 4 Transmission loss of a wall assembled from 10 basic modular units with joints sealed

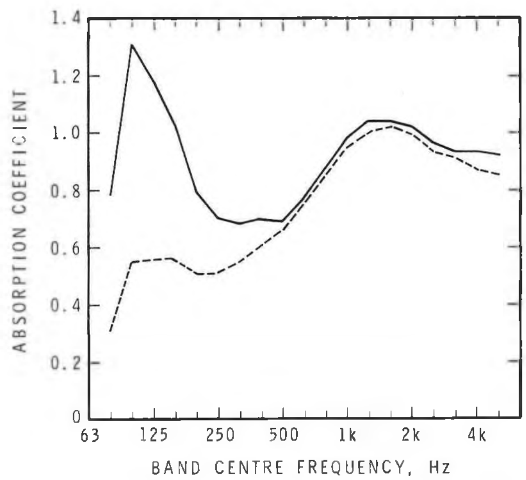

FIG. 5 Absorption coefficient for an assembly of 10 basic modular units: (---) slits blocked, (——) slits open

MEASUREMENT OF NOISE REDUCTION

The basic procedure was to measure the sound from the heat pump without enclosure and then to re-measure the sound levels with each enclosure in place. The difference is the noise reduction provided by the enclosure, assuming no change in the sound power output from the heat pump. The measuring environments are shown by site maps in Fig. 6(a); microphone positions are indicated by the circled numbers on the latter. Schematic drawings of the enclosure designs are in Fig. 6(b).

All the data were measured by means of an integrating sound level meter (B&K Type 2218) with an attached octave band filter set. In general, an integration time of at least 16 s was used and measurements were repeated several times to minimize the risk of error.

For several reasons (asymmetry of the enclosures, directionality of noise emission from the heat pump, and interference between the sound waves propagating directly from source to receiver and those reflected from building surfaces and the ground) it was necessary to take measurements at several positions round each unit in order to assess the overall change in sound levels. These effects caused significant variations in measured sound levels from one position to another even without an enclosure, as illustrated by the data for no enclosure at Site 1 (see the solid curves in Fig. 7). As shown by the site map, the three measuring positions are at similar distances from the heat pump, but the measured spectra are quite different. To minimize distortion of the results by spatial variation of sound levels the microphone was carefully returned to the original positions for each series of tests.

Background sound levels (with the heat pump switched off) were also measured from time to time to assess possible contamination of the data by other noise sources, including playing children, occasional aircraft, local traffic, and rustling leaves

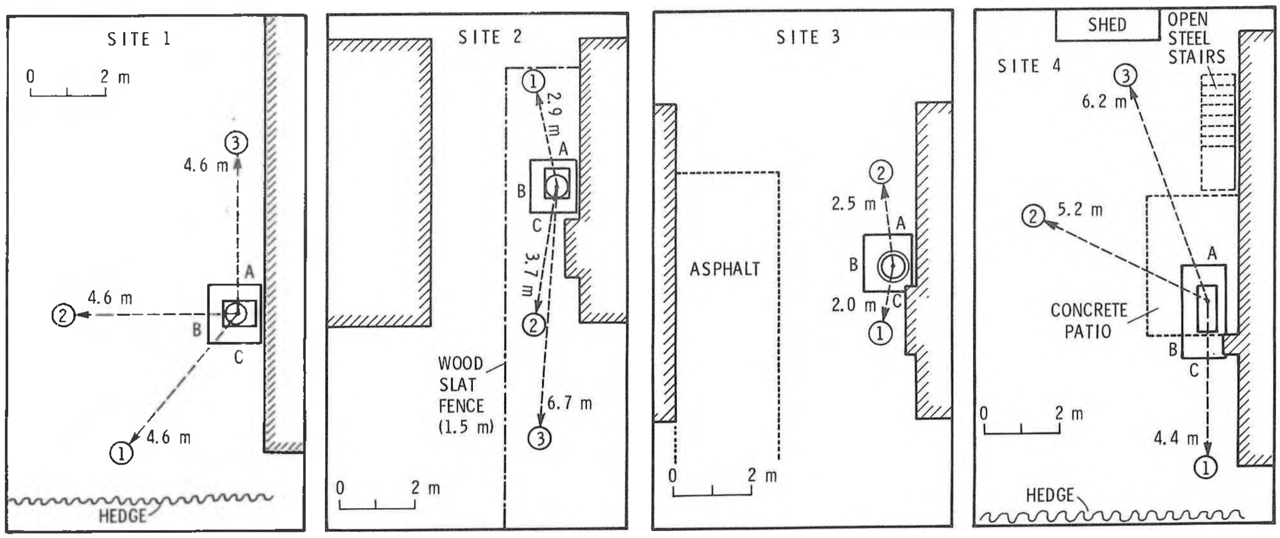

FIG. 6(a) Four test sites showing measurement positions (circled numbers) and site details.

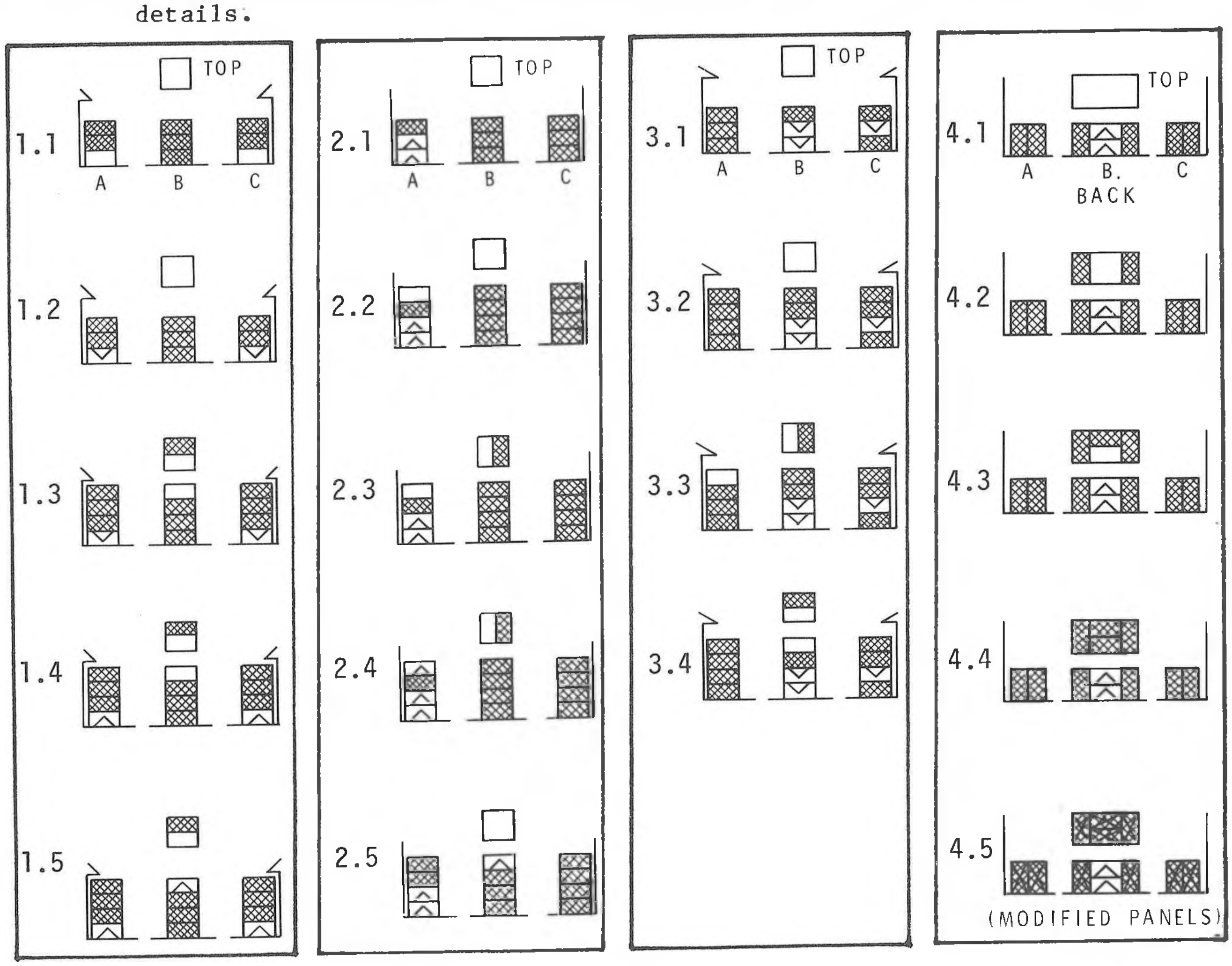

FIG. 6(b) Schematic drawings of enclosures. Module representation is explained in caption of Fig. 3.

(often the dominant source at 4 kHz). Whenever specific events were noticed measurements were suspended and all suspect data were discarded. There remained, however, the variable contribution from numerous minor sources like those listed above. The highest ambient levels recorded at each position in each frequency band are presented in the figures as a cautionary indicator of background sound. These background levels are given by the dotted lines in Figs. 7-11. In many cases the enclosures reduced the sound from the heat pumps to levels comparable with and occasionally below these nominal background values. In such cases the ambient noise presumably limits the apparent noise reduction provided by the enclosures and the true noise reduction is greater than the uncorrected data would suggest.

Because the ambient noise fluctuated appreciably, no attempt was made to "correct" the data numerically for the contribution from background sound. To obtain more reliable results, measurements with the heat pump on were systematically repeated. When appreciable variations were observed, the lowest repeatable value was used on the assumption that the higher values were due to extraneous sources; multiple repetitions when background sources seemed minimal were frequently required. The consistency from site to site of the dependence on variables such as enclosure height suggests that these procedures minimized the effect of background sound on the noise reduction data.

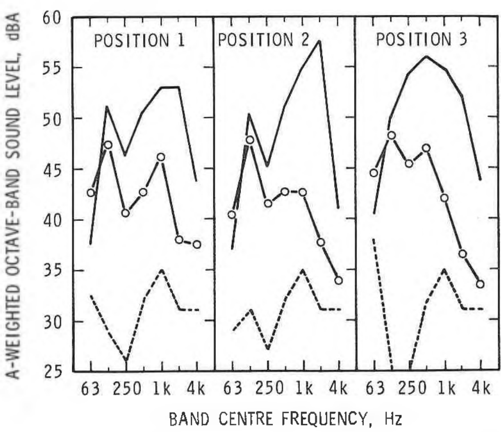

FIG. 7 Measured sound levels at Site 1:
(——) no enclosure
(—o—) enclosure 1.4
(•••) ambient sound level, heat pump off

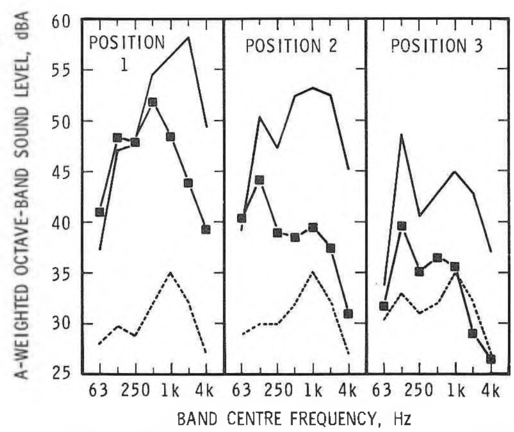

FIG. 8 Measured sound levels at Site 2:
(——) no enclosure
(—■—) enclosure 2.3
(•••) ambient sound level, heat pump off

The circles in Fig. 7 give measured sound levels for enclosure design 1.4 (Fig. 2). At all three measuring positions it is evident that the enclosure provides more noise reduction at high and mid frequencies than at low ones; in fact, a slight increase in sound levels is evident for the 63-Hz band. Similar trends were evident for all enclosures.

Many aspects of enclosure performance can be explained by evaluating the acoustical data for the various measuring positions with reference to the site maps and enclosure designs in Fig. 6. One obvious feature of the data in Fig. 8 for enclosure 2.3 is the large difference in noise reduction for positions 1 and 2. This difference seems reasonable when one considers the enclosure design. The surface facing position 2 consists of four of the standard panels, whereas that facing position 1 is primarily air inlet and outlet openings, which, as expected, provide much less noise reduction. This directionality is evident not only for high frequencies but also for the 63-Hz band, where one might expect more omni-directional performance. A similar though less pronounced effect is discernible in Fig. 7 in the data for enclosure 1.4: the noise reduction at positions 1 and 3 is lower than that at position 2, which is not directly exposed to an air inlet. Comparable effects were observed at the other sites.

In addition to the variation in noise reduction caused by transmission through the air flow openings, there were directional effects associated with diffraction at the top of the enclosure and reflections from adjacent surfaces. The data in Fig. 9 for enclosures 3.3 and 3.4 show the interaction of a partial top on the enclosures with reflections from an overhanging roof soffit. When the top panel is on the side closest to the house wall (squares in Fig. 9), it blocks reflections from the overhanging soffit; there is less noise reduction at position 1 than at position 2 due to transmission through the air inlet facing position 1. When the top panel is shifted to the side nearest position 1 (circles), less sound reaches position 1 over the top of the enclosure, but sound levels increase at position 2 due to increased diffraction over the top edge and reflection from the overhanging surface. The negligible change in the low frequency levels at position 1 suggests that these bands are dominated by sound transmitted through the air inlet. Similar but less clear-cut results at other sites support the belief that reflections from overhanging surfaces can limit noise reduction but that this effect can be reduced by a top that blocks the line from the heat pump to the overhang.

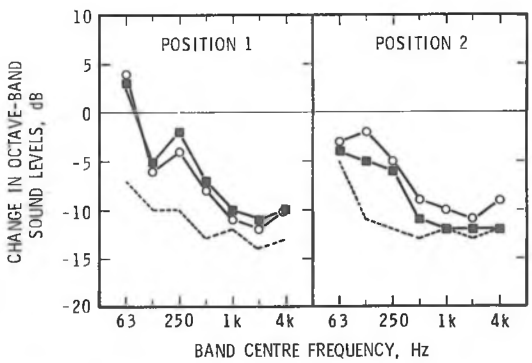

FIG. 9 Change in measured sound levels at Site 3:
(—o—) adding enclosure 3.3
(—■—) adding enclosure 3.4
(• • •) apparent limit due to ambient sound

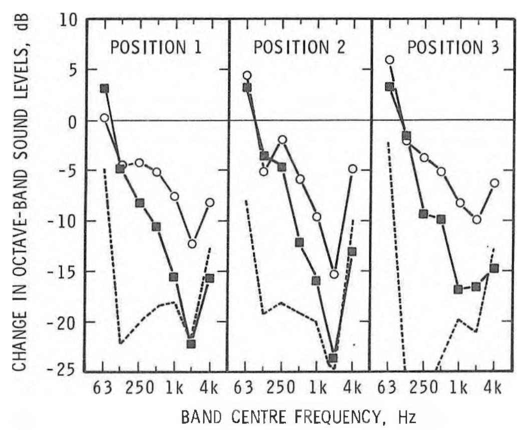

FIG. 10 Change in measured sound levels at Site 1:
(—o—) adding 3-unit high enclosure 1.2
(—■—) adding 4-unit high enclosure 1.5
(• • •) apparent limit due to ambient sound

One of the most obvious variables of the enclosure design was height. Figure 10 shows the reduction in sound levels at Site 1 for enclosure 1.2 (three modules high) and enclosure 1.5 (four modules high). At frequencies above 1 kHz the effect of ambient noise limits the apparent noise reduction for the higher enclosure, but the increased reduction of mid and high frequency sound with increased enclosure height is clearly demonstrated. The increased sound levels in the 63 Hz band were not strongly affected by enclosure height. Similar results were obtained at all sites with this general enclosure design.

Because of horizontal air flow through the heat pump, a rather different enclosure design was used at Site 4. Data are presented in Fig. 11 for enclosures 4.1 (with no top) and 4.4 (complete enclosure) by circles and squares, respectively. As one would expect, addition of a top provided more noise reduction. The complete horizontal enclosures (designs 4.4 and 4.5) provided noise reduction similar to that for other enclosures in these case studies except that they did not amplify the sound in the 63-Hz band.

Enclosure 4.5 was identical to enclosure 4.4 except that the slits on the inner faces of the panels were blocked to eliminate resonant low frequency absorption. Despite the substantial change in the absorption coefficient (as shown by the two curves in Fig. 5), noise reduction by the enclosure changed very little. It increased slightly in some frequency bands and decreased in others. This suggests a reduced sensitivity to wall absorption if wavelength is approximately equal to the enclosure dimensions. Subsequent, more detailed studies with 1:12 scale model enclosures showed a similar effect.

The effect of sound transmission through the air flow openings was noted in the discussion of the directionality of the enclosure performance. To reduce transmission through these openings, the direct path from heat pump to receiver was partially blocked by baffles, as shown in Figs. 1 and 2. The effectiveness of these baffles for the enclosures at Site 1 is illustrated by the data in Fig. 12; circles indicate the change in sound level when baffles were inserted in the inlet openings at the bottom of the three-module high enclosure. There was no significant effect at position 2 (not in line of sight from either inlet opening), but at the other positions the added baffles gave a small reduction of the noise in frequency bands above 500 Hz. This is similar to the reported effect of absorptive flow separators in ducts, where they provide significant attenuation only for separations ∿1/2 wavelength.

The squares in Fig. 12 show the change in sound levels when baffles were added to the outlet opening at the top of the front face of the four-module high enclosure (converting design 1.4 to 1.5). This had little effect at position 3 (not in line of sight of this outlet), but gave considerable additional noise reduction at the other positions. The greater effectiveness of baffles in this case is presumably due to directing sound energy upward, away from the measuring positions; for the inlet openings discussed in the previous paragraph, reflection from the ground and wall surfaces prevented effective re-direction of sound energy.

SUMMARY AND DISCUSSION

Despite large openings to permit air flow through the heat pumps, the enclosures tested provided appreciable noise reduction: typically 7 to 10 dB change in the broadband A-weighted sound level. Comparison of the results for different enclosure designs gives some indication of how design can be optimized to give maximum noise reduction in a given direction. Tabulated data are available from the second author.

For vertical-discharge enclosures (Sites 1 to 3) it is clear that increasing the height provides significantly more noise reduction. But despite this acoustical benefit it seems unlikely that enclosure heights much greater than 2 m would be considered by most homeowners. The reduction is obtained largely by redirecting sound

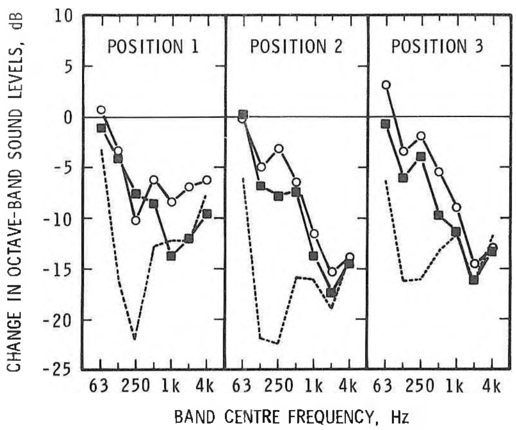

FIG. 11 Change in measured sound levels at Site 4:
(—o—) adding enclosure 4.1, open top
(—■—) adding enclosure 4.4, closed top
(•••) apparent limit due to ambient sound

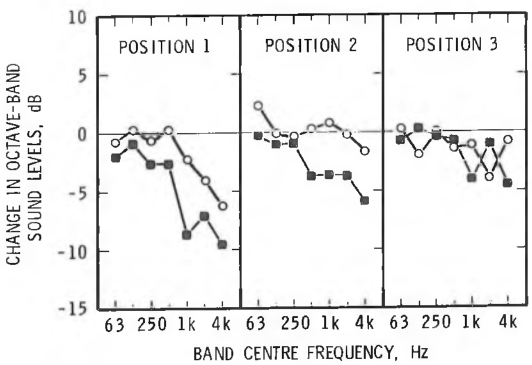

FIG. 12 Effect of baffles in inlet and outlet openings of enclosure at Site 1:
(—o—) baffles added to inlet openings at bottom, changing design 1.1 to 1.2
(—■—) baffles added to outlet at top, changing design 1.4 to 1.5

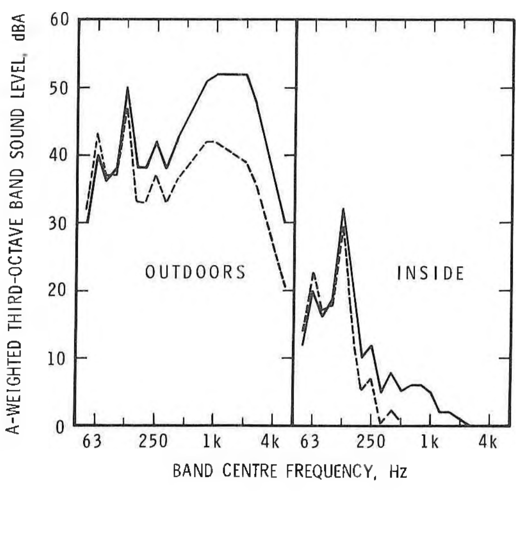

FIG. 13 Combined effect of enclosure and transmission through exterior wall: (——) sound levels without enclosure, (---) sound enclosure added. Outdoor level changes from 60 to 52 dBA. Indoor level changes from 33 to 31 dBA

energy. It is clearly preferable to locate the inlet and outlet openings to direct sound away from the most sensitive points of reception (the most desirable direction being upwards in most cases). Where there is an overhanging reflecting surface, a partial top on the enclosure to block the path from the heat pump can appreciably reduce the sound reflected back towards typical receiver positions.

The insensitivity to low frequency absorption (shown by the small effect of blocking the slits on enclosure 4.4) suggests that a simpler enclosure wall design should be satisfactory. One reasonable possibility is a 38 x 89-mm wood stud frame with 19-mm plywood on the exterior face and 12-mm vinyl-surfaced glass fibre panels on the inner face.

The enclosures obviously provide more noise reduction at mid and high frequencies than at low frequencies; in fact, most of the configurations studied gave an increase in sound level in the 63-Hz band. Subsequent studies with 1:12 scale model enclosures indicate that this effect is due to resonant response of low frequency modes of the enclosure; results from the model tests will be discussed in detail in another paper. This feature of the results is, however, of critical importance in assessing the usefulness of enclosures for reducing noise impact from heat pumps. If noise in adjacent outdoor areas is of primary concern, then reduction in the broadband A-weighted sound level should give a good indication of perceived reduction in loudness and hence the typical change in annoyance. Enclosures like those studied here could in many cases prevent heat pump noise from intruding above the neighbourhood ambient.

Unless windows are open, an enclosure would be much less effective in reducing the noise heard inside a neighbouring building. The exterior walls of a building provide greater noise reduction at high frequencies than at low frequencies. This reduces the contribution of the high frequencies to the perceived indoor loudness of the sound. Figure 13 illustrates the characteristic differences between indoor and outdoor sound levels from a rather noisy heat pump. Addition of an enclosure has little effect on the indoor A-weighted sound levels because it only slightly reduces the low frequencies dominating the indoor sound. Hence, an enclosure is unlikely to reduce significantly annoyance caused by noise heard inside neighbouring houses.

For both indoor and outdoor cases, the addition of an enclosure offers only a limited reduction in noise. A much more effective long-term approach would be to reduce the noise emitted by heat pumps. Locating the compressor unit indoors, reducing cabinet and support resonances, and improving fan designs could significantly reduce noise output, especially the troublesome low frequencies. The combination of better heat pump design, sensible location of the outdoor unit and, in some cases, use of barriers or enclosures could eliminate this type of noise problem.

A repeated cautionary note is required: no assessment was made of possible reduction of thermal efficiency or increased strain on the fan as a result of air flow restriction by the enclosures. Further studies are planned to determine whether these enclosures significantly interfere with proper heat pump performance.

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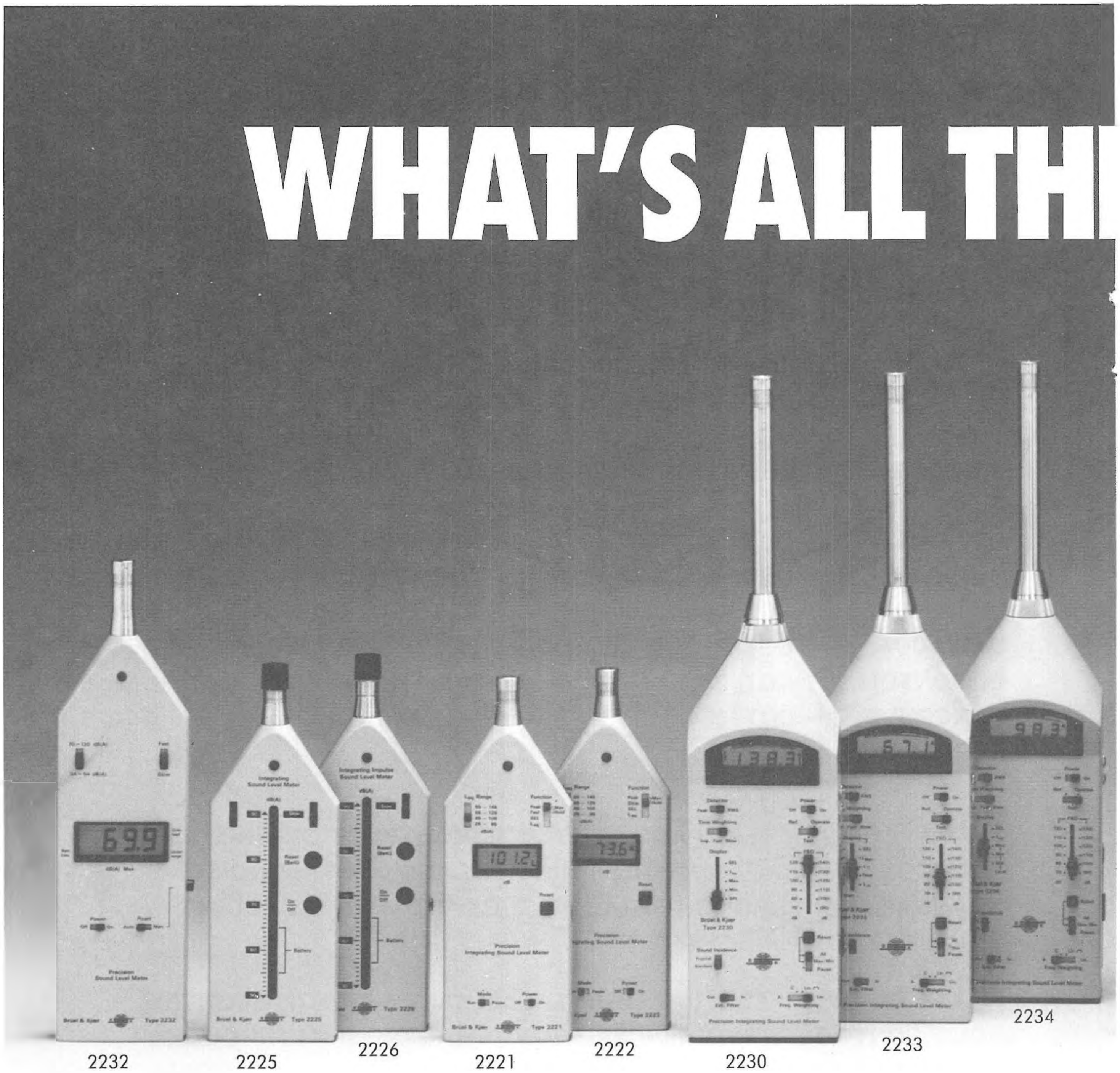

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