THE EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS ON THE AIRBORNE SOUND INSULATION PROPERTIES OF MILD STEEL SHEET

D R Robinson

Northampton Borough Council

Introduction

However desirable it may be to engineer out potential noise and vibration problems at the machinery design stage, it may not always be achievable. Economic or technical considerations may prescribe against such a solution but often it is a lack of due consideration to such problems or want of acoustic expertise which manifests itself.

Assuming design changes are not viable, the amplitude of received airborne sound may be reduced by, for example, the screening or enclosing of the noise source. The transmission of vibration may be controlled by suitable anti-vibration mountings and the radiation of structure-borne sound may be reduced by the application of panel damping treatments.

The method of reduction will vary from case to case and it is not uncommon for a combination of different control measures to be adopted in order that a satisfactory solution be achieved.

As the effects of barriers, enclosures and anti-vibration mountings are well researched, the consequence of their applications are predictable with a reasonable degree of accuracy. However, the effects of panel damping treatments are much more difficult to predict, especially if the engineering characteristics of the chosen damping layer have not been investigated and defined.

Although it is beyond the scope of this paper to provide an exposition of the damping and acoustic insulation of structures in general, its intention is to examine and evaluate the effect of NON-PROPRIETARY damping materials on the airborne sound insulation properties of a mild steel panel. The types of material under test are to be found in most high street hardware shops and have been manufactured for use in general do-it-yourself applications unrelated to panel damping.

Airborne Sound Insulation of Panels

The relative importance of the different mechanisms of sound transmission through a plate varies across the audio frequency spectrum. As a steel plate possesses the qualities of both stiffness and mass, it can thus exhibit both resonance and mode effects. At low frequencies, transmission depends mainly on the stiffness of the plate material. In this STIFFNESS CONTROLLED REGION damping and mass are relatively unimportant.

At slightly higher frequencies, the resonances of the plate control its behaviour. In the RESONANCE CONTROLLED REGION, although mass exerts its influence, damping effects assume a dominant importance.

EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

Above a frequency in the order of twice the fundamental resonant frequency, the plate will tend to behave as an assembly of small masses. This MASS CONTROLLED REGION is little affected by damping effects and extends up to a critical frequency, above which plate transmission losses are in the COINCIDENCE CONTROLLED REGION and as well as mass, damping once again assumes an important role, as does plate stiffness.

As observable damping effects are normally restricted to the resonance controlled and coincidence controlled regions, any further discussion will be similarly restricted.

Panel Resonance and Coincidence Effect

The stiffness controlled and mass controlled regions are separated by the resonant frequencies of the panel in bending. For a test panel with all edges clamped, the fundamental resonant frequency is calculated [12] to be 117.6 Hz.

At frequencies above the lowest resonant frequency, additional bending resonances occur in increasing numbers. These bending resonances are the result of an in-phase combination of waves travelling toward and away from the edges of the panel. This situation is analogous to the production of eigenmode frequencies in a reverberant space.

The lowest frequency at which coincidence can occur, with the acoustic wave front at grazing incidence to the panel, is called the CRITICAL or COINCIDENCE frequency and is derived from the classical Cremer formula which, for the mild steel test panels under consideration, computes [5] to 10019.6 Hz.

Equivalent and various approximations have been derived for the calculation of the coincidence frequency of a homogeneous panel. Several other predicted coincidence frequencies [2,10] for the mild steel test panels are 10096.4 Hz, 10469.5 Hz and 10549.8 Hz. Additionally, the product of surface density and coincidence frequency for steel, in metric units, is given [1] the value of 97500. For the test plates under consideration, this computes a coincidence frequency of 10337.2 Hz.

Structural Damping

Most engineering materials like steel contain little inherent damping. They also behave in a practically ideal elastic manner when subjected to loading. As such an elastic material cannot convert applied vibrational energy to heat, imposed vibrations will propagate through the material with ease and can result in noise and vibration problems.

If any resonant vibrations do become problematical, they must be reduced by the application of external control. In the case of steel plates, use is sometimes made of some sort of stiffening arrangements. The resonances are not damped but merely shifted towards higher frequencies. If, as a consequence, the resonances can be shifted to frequencies which will not be excited by the imposed mechanical or acoustical energy then this solution may prove acceptable

EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

If such stiffening arrangements prove impracticable then the solution to the problem is to apply some form of external damping treatment to the panel which can both store the mechanical energy and convert it to heat, thus reducing the amplitude of vibrations in a construction. If the treatment is applied to dampen structure-borne sound then, in the majority of cases, such damping will also result in a reduction of airborne sound radiation.

The effectiveness of a damping treatment for noise reduction is dependent on two factors. Firstly, panel vibrations must occur at RESONANT FREQUENCIES. Damping having relatively little effect on off-resonant vibrations. Secondly, panel vibrations must be capable of GENERATING SOUND WAVES.

In the case of airborne sound excitation, bending waves contribute little to acoustic radiation at frequencies below the critical frequency. Therefore, a surface damping treatment will have little effect between about twice the fundamental panel resonant frequency and the critical frequency other than to increase the panel mass somewhat. Nevertheless, application of a damping treatment to panel edges, corners and other attachment points may be effective in reducing radiated noise in the mass controlled region, depending on mounting conditions.

At frequencies greater than the critical frequency, a damping treatment applied to the panel surface at locations of greatest displacement is said to be a useful noise control procedure.

Of the various types of damping treatments in common use, only two types are to be considered here, namely FREE-LAYER or HOMOGENEOUS damping, and CONSTRAINED-LAYER damping.

The system loss coefficients provided by either type are approximately equal for treatment surface weights in the order of 10-20% of the panel surface weight. For treatment weights less than 10% of the panel surface weight, constrained-layer damping is probably more effective than homogenous damping. For treatment weights greater than 20% of panel surface weight, the converse is probably true.

Test Room

The room within which the experiments were undertaken is SUBTERRANEAN with no direct access to the external air. As a consequence, the ambient noise level within the room is low. Its major dimensions being 5.41 m long, 4.79 m wide and 2.93 m high. Therefore, with a gross volume of 75.9 m^{-3}, it is relatively small.

The floor and ceiling surfaces are constructed of concrete and a large concrete supporting beam spans the ceiling. The walls are constructed of brick and a large proportion of their surface is covered with fibreboard fixed on battens with a 25 mm air space. A small blackboard, again with a 25 mm air space, is affixed to one wall and two of the walls possess doors.

EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

Room reverberation times were measured at the test microphone position, the results of which were as follows

	1/1 OCTAVE BAND CENTRE FREQUENCY : Hz					
	125	250	500	1K	2K	4K
MEASURED T_{60} sec^{-1}	2.0	1.3	1.6	1.9	2.0	1.8

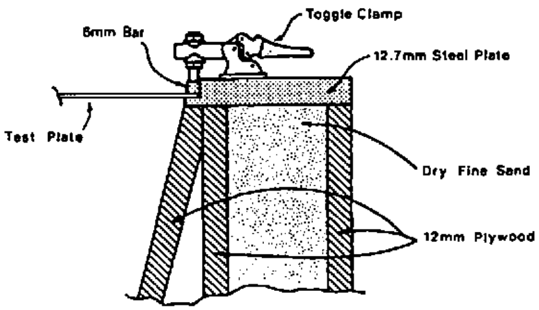
It can be seen that, although relatively small, the room is fairly 'live'.

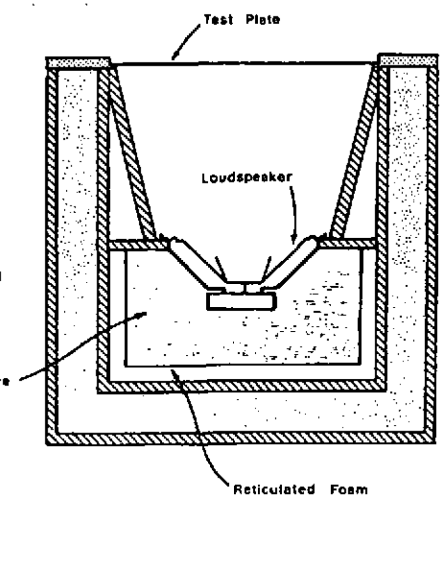
Room eigenmodes were also computed and although it was not considered that room resonances would adversely affect the measurement results, diffusers were positioned in the three main axes of the room in line with the measurement microphone.

Test Rig
As ISO 140 test facilities were not readily available, it was necesary to design and construct a test rig suitable for the purpose.

Mild steel plate is commonly used in the construction of small machine enclosures and casings, ductwork, etc. When used in such a manner, the mild steel plate will normally be subjected to diffuse acoustic pressure. Accordingly, it was decided to use 18 GAUGE mild steel plate for the test panels and to arrange, as far as possible, for a diffuse sound field to impinge on one face of the test panel only.

Taking the various constraints into consideration, the final construction used for the tests is shown below





EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

The rig had not only to be capable of securely holding the panel under test but to provide consistency in mechanical performance between test pieces. It also had to be so constructed that flanking transmission was minimised.

Essentially, it comprises a double box construction with the space between the boxes filled with dry fine sand. The inner box is horizontally divided into two sections, the lower of which acts as an infinite baffle loudspeaker enclosure. The free air resonance of the loudspeaker unit is quoted as 35 Hz and its response on a baffle, against the above recommended enclosure volume, is further quoted as 30 Hz to 16 kHz. Although the frequency range tolerances are not quoted, the range was considered sufficient for the purpose.

The internal dimensions of the inner box at its top correspond to those of the test panel free area. The sides and corners of the upper portion of the inner box, from the outside edge of the loudspeaker unit to the top edges of the box are fitted with tapered wooden infills, principally to minimise the generation of internal standing-wave resonances.

The boxes are capped off using 12.5 mm steel plate. This plate is machined to provide a rebated orifice against which the test plates are clamped. It is securely screwed down onto the outer box together with rubber compression seals on both inner and outer box top edges. The square central orifice has sides of 316.23 mm (12.45 in) giving a free area of 0.1 m^{-2}.

In order to minimise the possibility of flanking transmission by way of air leakage around the plate edges they, after some experimenting with non-hardening mastic, were finally bedded in a thin film of Duckhams 'Keenol' grease along the rebate face.

Each of the three coated test panels and the untreated control panel were mounted and tested separately in the rig, the pink noise test sound field being established before the series of tests and maintained at a constant level throughout the whole series of measurements from panel to panel.

Test Results

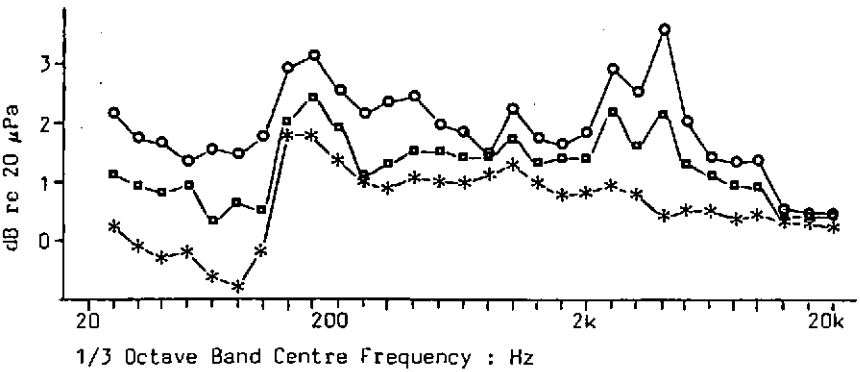
The first plate was treated with a synthetic rubber compound called ISOFLEX LIQUID RUBBER. It is a urethane elastomer and is primarily intended for the waterproofing and repair of flat roofs. The test plate was first painted with Isoflex Special Primer before applying two coats of Isoflex Liquid Rubber in accordance with the manufacturer's recommendations. All coats were applied to the plate by means of a paint brush, the final thickness being in the order of 0.5 mm. The weight of compound applied to the plate was 0.055 Kg.

The second plate was treated with an adhesive bandage called SYLGLAS ALUMINIUM TAPE. The tape is constructed as an aluminium backing layer covered with a coating of adhesive. It is manufactured as a waterproofing and repair tape for glazing and sealing. It is available in various widths and is self-adhesive. The 75 mm wide tape was used and laid on the plate in abutting strips. The thicknesss of the tape was in the order of 1.0 mm and the weight of the tape applied to the plate was 0.068 Kg.

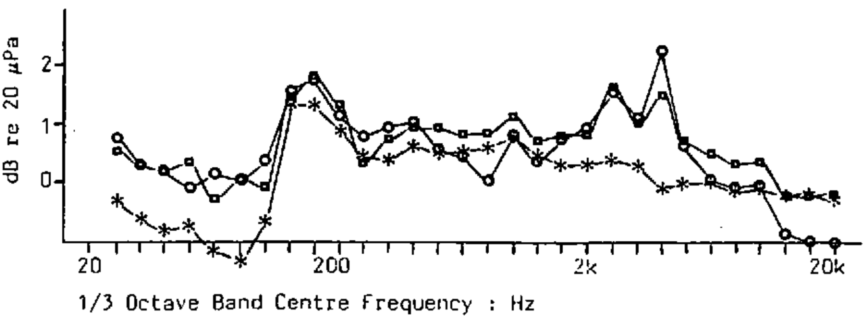
EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

The third plate was treated with a bitumenised bandage called EVO-STIK FLASHBAND FLASHING. The tape is constructed as an aluminium backing with a self-adhesive bitumen coating. It is also available in various widths and is intended for use as roof flashing or for the repair of guttering, soil and rainwater pipes, etc. As with the previous plate, the 75 mm tape was used and was applied in the same manner. The thickness of this tape was in the order of 1.5 mm and the weight of tape applied to the plate was 0.16 Kg.

Radiated sound pressure levels were measured using a microphone which, after several trial and error positions, was placed 0.5 m above each plate. The measured levels for each treated plate were compared to an untreated plate which was used as a control. The 1/3 octave band level differences are shown graphically below, both corrected and uncorrected for mass controlled effect.



UNCORRECTED LEVEL DIFFERENCES OF DAMPED v. UNDAMPED PANELS



LEVEL DIFFERENCES OF DAMPED v. UNDAMPED PANELS CORRECTED FOR MASS EFFECT

—— ISOFLEX LIQUID RUBBER
▫——▫ SYLGLAS ALUMINIUM TAPE
o——o EVO-STIK FLASHBAND FLASHING

EFFECT OF NON-PROPRIETARY SURFACE DAMPING TREATMENTS

Finally, taking published airborne sound insulation values for 18 gauge steel plate as a standard, the effect of applying each damping treatment is predicted, as follows

	1/1 O B CENTRE FREQUENCY : Hz					
	125	250	500	1K	2K	4K
Published Sound Transmission Losses [12]	15	19	31	32	35	48
Additional losses due to ISOFLEX LIQUID RUBBER	1	1	1	1	1	1
Predicted Sound Transmission Losses for Damped Panel	16	20	32	33	36	49
Additional losses due to SYLGLAS ALUMINIUM TAPE	2	1	2	2	2	2
Predicted Sound Transmission Losses for Damped Panel	17	20	33	34	37	50
Additional losses due to EVO-STIK FLASHBAND FLASHING	3	2	2	2	2	3
Predicted Sound Transmission Losses for Damped Panel	18	21	33	34	37	51

Commentary

It would be very easy to simply dismiss the effect of non-proprietary damping treatments, or any damping treatment for that matter, on the airborne sound insulation efficiency of mild steel sheet as of only marginal significance. Although this conclusion may be accurate, it would be based on the results for only the three samples studied.

The results do, however, support the premise that for an increase in superficial mass of less than 10% then constrained-layer damping is more effective than homogeneous damping.

It is accepted practice that, when considering the airborne sound transmission losses of materials, only the losses for the 1/3 octave bands from 100 Hz to 3.15 kHz are used. Consequently, the average loss will be the arithmetic mean of the losses of these 16 bands.

Of the three treatments tested, EVO-STIK FLASHBAND FLASHING is clearly the most effective overall, but then it does give the greatest increase in mass. The increase in mass alone would show a corresponding increase in insulation efficiency of some 1.4 dB. The observed average increase in loss over the untreated plate of 2.2 dB presumes an average excess loss due to damping effects of 0.8 dB.

The next most effective treatment was the SYLGLAS ALUMINIUM TAPE, which would have shown an increase in insulation efficiency of some 0.6 dB due to mass alone. The observed average increase in loss of 1.5 dB presumes an average excess loss due to damping effects of 0.9 dB.

Both of the above two treatments are of the constrained-layer type and had increased their superficial masses by 17% and 7% respectively.

The least effective treatment was ISOFLEX LIQUID RUBBER with an applied thickness of 0.5 mm. This homogeneous damping treatment had shown an increase in superficial mass of some 6% and would have given rise to an increase in insulation efficiency of some 0.5 dB due to mass effect alone. The observed average increase in loss of 1.0 dB presumes an average excess loss due to damping effects of 0.5 dB.

As expected, all three treatments showed a marked improvement in airborne sound insulation efficiency for the band containing the fundamental panel resonant frequency. However, all three treatments were disappointing for the band containing the critical frequency and above.

Taking only the effects of damping into consideration, both of the constrained-layer treatments showed similar excess losses. Nevertheless, the effect is still marginal and a judicious increase in panel mass would still appear to be more efficacious.

References and Bibliography

[1] L.L.BERANEK (Ed.), Noise Reduction, McGraw-Hill (1960)
[2] L.L.BERANEK (Ed.), Noise and Vibration Control, McGraw-Hill (1971)
[3] R.A.BURGESS et al, The Construction Industry Handbook, Medical and Technical Publishing (1973)
[4] J.T.BROCH, Mechanical Vibration and Shock Measurements, Bruel & Kjaer (1972)
[5] W.FURRER, Room and Building Acoustics and Noise Abatement, Butterworths (1964)
[6] K.B.GINN, Architectural Acoustics, Bruel & Kjaer (1978)
[7] C.M.HARRIS & C.E.CREDE (Eds.), Shock and Vibration Handbook, McGraw-HIll (1976)
[8] D.HARRISON (Ed.), Building Methods and Products, Architectural Press (1976/7)
[9] B.JOHANSSON, Dynamic Vibration Isolation and Absorption, Mechanical Engineering Publications (1982)
[10] L.E.KINSLER et al, Fundamentals of Acoustics, John Wiley & Sons (1982)
[11] H.KUTTRUFF, Room Acoustics, Applied Science (1979)
[12] H.W.LORD et al, Noise Control for Engineers, McGraw-Hill (1980)
[13] E.B.MAGRAB, Environmental Noise Control, Wiley-Interscience (1975)
[14] P.M.MORSE, Vibration and Sound, McGraw-Hill, (1948)
[15] R.M.TENNENT (Ed.), Science Data Book, Oliver & Boyd (1971)