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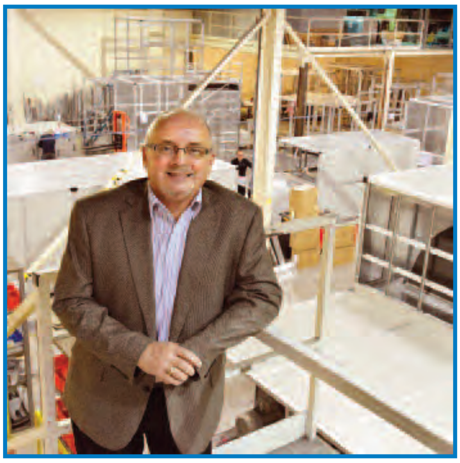


Sound Advice for Ventilation Plant in Schools

By David Pinchbeck

air HANDLERS

INDEX OF CONTENTS

REQUIREMENTS FOR BB93 COMPLIANCE

To comply with Building Bulletin 93 Acoustic Design of Schools and Places of Learning we have compiled the following information as a Design Guide to meet both the Building Regulations and BB93 for Schools and Places of Learning.

Specification of Acoustic Performance

Type of Room	Room classification for the purpose of airborne sound insulation in Table 1.2		Upper limit for the indoor ambient noise level
	Activity noise (Source Room)	Noise tolerance (Receiving Room)	LAeq. 30min (dB)
Nursery school playrooms Nursery school quiet rooms	High Low	Low Low	35 35
Primary school: classrooms, class bases, general Teaching areas, small group rooms	Average	Low	35
Secondary school: classroom, general teaching areas, seminar rooms, tutorial rooms, language laboratories	Average	Low	35
Open Plan: Teaching areas Resource areas	 Average Average	 Medium Medium	 40 40
Music: Music classroom Small practice/group room Ensemble room Performance/recital room Recording studio Control room for recording	 Very high Very high Very high Very high Very high High	 Low Low Very low Very low Very low Low	 35 35 30 30 30 35
Lecture Rooms: Small (fewer than 50 people) Large (more than 50 people)	 Average Average	 Low Very low	 35 30
Classrooms designed specifically for use by hearing impaired students (including speech therapy rooms)	Average	Very low	30
Study room (individual study, withdrawal, remedial work, teacher preparation)	Low	Low	35
Libraries: Quiet study areas Resource areas	 Low Average	 Low Medium	 35 40
Science laboratories	Average	Medium	40
Drama studios	High	Very low	30
Design and Technology • Resistant Materials, CADCAM areas • Electronics/control, textiles, food, graphics, design/resource areas	 High Average	 High Medium	 40 40
Arts rooms	Average	Medium	40
Assembly halls, multi-purpose halls, (drama, PE, audio/visual presentations, assembly, occasional music)	High	Low	35
Audio-visual, video conference rooms	Average	Low	35
Atria, circulation spaces used by students	Average	Medium	45
Indoor sports hall	High	Medium	40
Dance studio	High	Medium	40
Gymnasium	High	Medium	40
Swimming pool	High	High	50
Interviewing/counselling rooms, medical rooms	Low	Low	35
Dining rooms	High	High	45
Ancillary spaces: Kitchens Offices, staff rooms Corridors, stairwells Coats and changing areas Toilets	 High Average Average-High High Average	 High Medium High High High	 50 40 45 45 50

The Acoustic Performance Specification indoor noise levels are based on an ambient noise level LAeq.30min. This applies to all teaching spaces and is intended only for guidance in administration and ancillary spaces.

AIRBORNE SOUND INSULATION BETWEEN SPACES

The objective is to attenuate airborne sound transmission between spaces, through walls, floors and ceilings.

SOUND INSULATION OF MECHANICAL VENTILATION PLANT

Mechanical Ventilation plant is located in various areas within the school, and can be one of the following areas:

A. **ROOF** – Rooftop Ventilation plant will be of weatherproof construction, and will normally take its fresh air intake and discharge its exhaust air to atmosphere.

Noise transmission from these plants can be passed both back into the building spaces through the roof/windows etc or it can be transmitted to local residencies beyond the school boundaries.

There are various ways of suppressing and attenuating rooftop plant noise and the table below highlights this.

ATTENUATION & SUPPRESSION OF ROOF PLANT NOISE

TABLE 1

AREAS OF CONCERN	METHOD OF NOISE CONTROL
Direct Noise Transmission Through the Roof	**This can be from both airborne noise and vibration transmission.** **1. Airborne noise can be eliminated with the correct sound reduction index of the plant casework.** **2. Vibration transmission can be controlled with the correct selection of anti-vibration mountings with a flexible connector on the fan or plant.**
Sound Transmission in the Building from Roof Plant	**1. Airborne sound emission from the plant fresh air and exhaust air louvres can be reduced with attenuators correctly sized to prevent plant noise entering the school and local residences via windows and building facia.** **2. Roof plant noise transmission from the plant casework can also transmit into the building and local residences.** **The correct selection of plant casework will eliminate this.**
Sound Transmission via Ductwork into the Building and Room Spaces	**Airborne sound emission via the supply and return ductwork. Correctly designed attenuators will reduce this noise into the school spaces.**

B. **PLANTROOM –** Plant room located ventilation plant can be designed to sit on the plant room floor or be suspended from the ceiling, and will normally take its fresh air and air discharge from ductwork and louvres from within the plant room external walls.

Noise transmission from these plants can be passed back into the building spaces through the emission from the louvres through external windows and facia or can be transmitted to local residences beyond the school boundaries.

Other forms of noise transmission are, duct airborne noise from the plant via the supply & return air ductwork, plant breakout through adjoining walls and structures, vibration through the floor or ceiling.

The table below highlights the methods of suppressing or attenuating plant room noise emission.

ATTENUATION & SUPPRESSION OF PLANTROOM NOISE

TABLE 2

AREAS OF CONCERN	METHOD OF NOISE CONTROL
Noise to atmosphere via plant room louvres.	**Noise to atmosphere will be mechanical noise transmitted from the plant via ductwork discharging exhaust air or bringing fresh air through louvres on the facia of the building.** **This emission can be controlled with the use of attenuators positioned to prevent flanking of plant room noise back into the duct after the attenuator.**
Sound Transmission through the plant room walls/floor/ceiling	**This can be from both airborne noise and vibration transmission.** **1. Airborne noise can be eliminated with the correct sound reduction index of the plant casework.** **2. Vibration transmitted into walls, floor and ceiling of plant rooms can be dampened with correctly sized anti-vibration isolators.**
Duct borne noise transmitted by the plant through ductwork into the room spaces	**Duct borne noise can be reduced by introducing attenuators into the ductwork. The selection of the attenuators must be selected to achieve the room noise requirement. Care must be taken to ensure noise flanking of the attenuators does not take place.** **A recommended way to avoid flanking is to acoustic lag all ductwork from the attenuator to the plant room wall. A second method is to position the attenuator directly before or through the plant room wall.**

CEILING MOUNTED SUPPLY & EXTRACT VENTILATION PLANT

TABLE 3

AREAS OF CONCERN	METHOD OF NOISE CONTROL
Direct noise transmission from the plant into the room	**This can be from both airborne noise and vibration transmission. The mechanical ventilation plant is installed within the ceiling space of the room being served or in a room adjacent to the room being served.** **1. Vibration transmitted noise can be transmitted into the room & structure. This should be dampened by correctly sized anti vibration isolators and flexible connectors.** **2. Noise breakout can radiate from the plant casework in to the room space.** **High impact plant casework is required offering enhanced sound reduction of the fan sound power level.** **False ceiling or plant sound treated spaces will improve noise control of in room mechanical ventilation plant.**

Duct borne noise transmitted by the plant through ductwork into the room spaces	**Duct borne noise can be reduced by introducing attenuators into the ductwork. The selection of the attenuators must be selected to achieve the room noise requirement.** **Care must be taken to give consideration to the room acoustic criteria which will vary with room size and absorption.** **Further consideration must be given to noise "Flanking" of attenuators.** **"Flanking" means noise which enters the ductwork after the attenuator from the plant room which increases the noise in the duct.** **Recommended ways to avoid Flanking are to acoustic lag all ducts in the room after the attenuator or position the attenuator directly before or through room walls.**

C. **<u>ROOM MOUNTED</u> –** Mechanical ventilation plant mounted within classrooms or in adjacent rooms are the most noise sensitive to acoustically meet low noise level room requirements for teaching spaces. The fact that the fan(s) are in the same location as the listener makes consideration of the noise breakout and room acoustic quality to absorb and decay noise of the highest importance.

The room(s) will have mainly two fans emitting sound; these will have the function of supply & extract. It is recommended that the supply and extract air are distributed into the room via ductwork.

Atmospheric side and system side attenuators must be used to reduce the airborne noise to the required level.

It is recommended that a false ceiling is used with no penetrations in it, allowing no "Flanking" to occur between the plant casework and the room.

We would require the false ceiling to be both impact decay of minimum 10dbA SRI and have a good absorption element of a class "A" ceiling, and a 100% room coverage, a recommended typical ceiling would be:-

> 2 X 12.5mm plasterboard skins (10kg/m^2) with 50mm mineral wool in the cavity suspended on resilient hangars, lighting must be acoustic type if integral in the ceiling.

All penetrations within the celling must be sealed.

ATTENUATION & SUPPRESSION OF ROOM MOUNTED PLANT

TABLE 4

AREAS OF CONCERN	METHOD OF NOISE CONTROL
Noise to atmosphere via louvres	**Noise to atmosphere will be mechanical noise transmitted from the plant via ductwork both discharging exhaust air and bringing fresh air through louvres on the building facia. This emission can be controlled with the use of attenuators positioned to prevent flanking. Note if the ductwork is not acoustic lagged from the plant to the attenuator noise may emit from the duct with a higher breakout level than the plant.**
Sound transmission into the ceiling and room	**This can be from both airborne noise & vibration transmission.** **1. Airborne noise can be eliminated with the correct sound reduction index of plant casework.** **2. Vibration transmitted into the ceiling can be dampened with correctly sized anti-vibration isolators.**
Duct borne noise transmitted by the plant through ductwork into the room spaces	**1. Duct borne noise can be reduced by introducing attenuators into the duct work. The selection of the attenuator must be selected to achieve the room noise requirement.** **Further consideration must be given to noise "Flanking" of attenuators.** **"Flanking" means noise which enters the ductwork after the attenuator from the plant area which increases the noise in the duct. This is normally for ventilation plant located in an acoustically sealed false celling with high breakout noise from the plant.** **Recommended ways to avoid Flanking are to acoustic lag the ductwork from the plant to the attenuators directly before or after the mechanical ventilation plant.**
Plant breakout into the room	**2. A very common issue is noise breakout of mechanical ventilation plant located in the room being served or a noise sensitive area.** **This application is often ceiling mounted in the room and will require serious acoustic treatment to the plant casework to reduce noise breakout.** **The sound power level of fan(s) in mechanical ventilation plant will be determined by the air volume and total static system pressure of each fan.** **Performance of mechanical ventilation plant is determined by the occupancy of the rooms and the fresh air requirement also CO_2 increasing above the recommended levels.** **Areas such as kitchens, toilets, fume cupboards in chemistry labs, will require fresh air and exhaust air to be separated.** **Often in the use of acoustically cased mechanically ventilated plant the casework radiated noise cannot be reduced by the plant casework alone, which means one of the following options is available.**

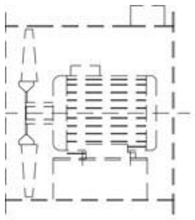
	A. An acoustic enclosure around the mechanical plant, which must be sealed where penetration is made to prevent sound leakage. **This solution is not normally practical, due to access to the plant, also adding height and length, which is usually a premium.** **B. The installation of a false ceiling with a sound reduction of at least 10 dbA. This ceiling must not have any penetrations which are not sealed to prevent sound leakage, particular point to take note are, lights which are integral in the ceiling could have a sound leakage more than the ceiling.** **Also the false ceiling must not be used as a return air plenum as sound leakage will occur through the return air terminals.** **A further point with cased mechanical plant installed in classrooms and noise sensitive areas, is that often the room is "live" with little sound absorption, whilst it points out clearly in Building Bulletin 93, reverberation must be low Tmf secs in the mid band octave frequencies, which represents a reverberation time of 0.8Tmf sec. This is rarely achieved by the building designers, adding to the room noise level.** **For further information on the impact of reverberation time on internal areas, visit the section "Reverberation Time and its Effect on Rooms".**

TYPES OF MECHANICAL VENTILATION PLANT

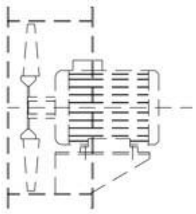
EXTRACT PLANT

AXIAL OR MIXED FLOW FANS:- Fans of this nature are mainly used for kitchen extract from canopies covering the cooking equipment. These fans are cased with metal single skin casing, with little sound breakout reduction, axial flow fans are often bifurcated with the motor out of airstream.

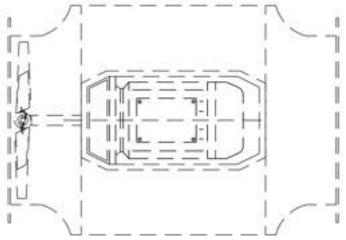
With bifurcated fans care must be taken as noise will be emitted directly from the motor to atmosphere if mounted externally or into the plant area if internally mounted.



Drawing of Typical Long Cased Axial Fan



Drawing of Typical Short Cased Axial Fan



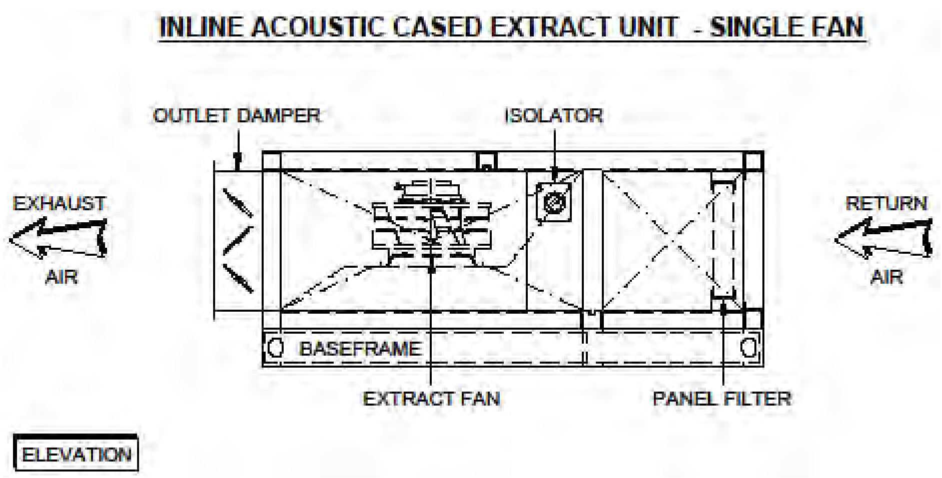
Drawing of Typical Bifurcated Axial Flow Fan

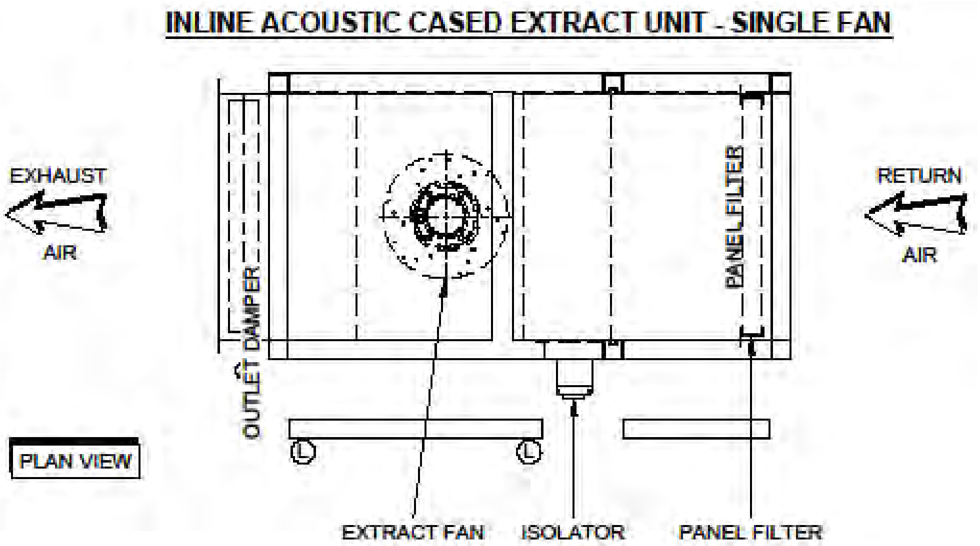
INLINE ACOUSTIC CASED EXTRACT UNITS

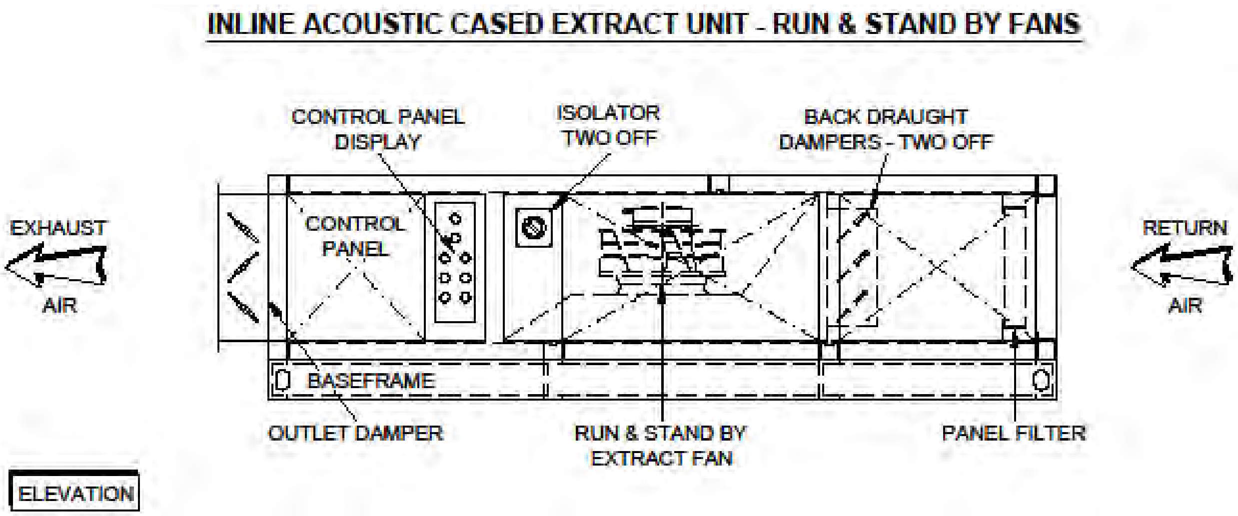
These extract fan units which are cased in double skin insulate casework will have a degree of sound breakout reduction. This will depend on the sound reduction properties of the casework.

Typical uses of this type of extract fan unit are, toilet extraction where duty and standby fans are used with auto change over mechanisms to give standby operation as dictated in building regulations.

Sound attenuation may be required to the system and atmospheric side, also an additional enclosure or false ceiling may be required if the plant casework sound reduction properties need further impact decay to achieve the room noise level.




INLINE ACOUSTIC CASED EXTRACT UNIT - SINGLE FAN
EXHAUST
AIR
RETURN
AIR
OUTLET DAMPER
PANEL FILTER
PLAN VIEW
EXTRACT FAN
ISOLATOR
PANEL FILTER

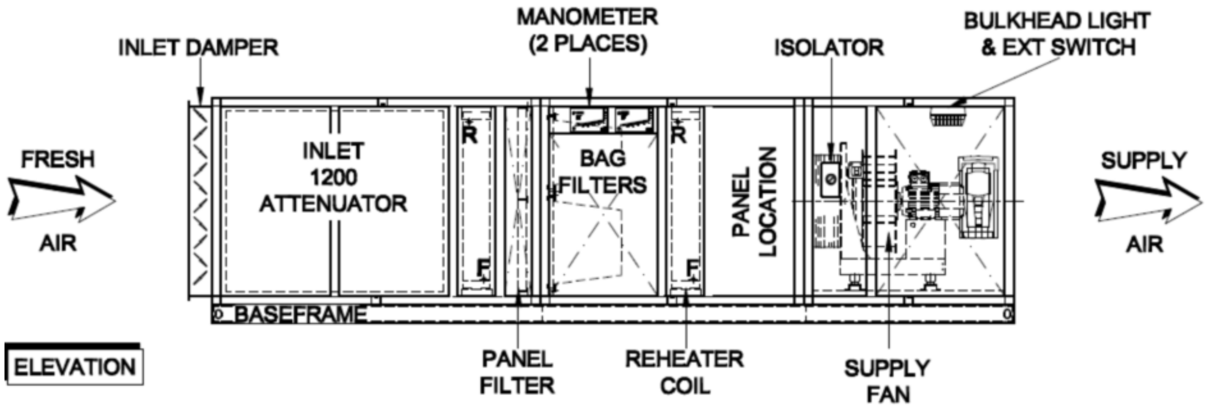

INLINE ACOUSTIC CASED EXTRACT UNIT - RUN & STAND BY FANS
CONTROL PANEL
DISPLAY
ISOLATOR
TWO OFF
BACK DRAUGHT
DAMPERS - TWO OFF
EXHAUST
AIR
CONTROL
PANEL
RETURN
AIR
BASEFRAME
OUTLET DAMPER
RUN & STAND BY
EXTRACT FAN
PANEL FILTER
ELEVATION

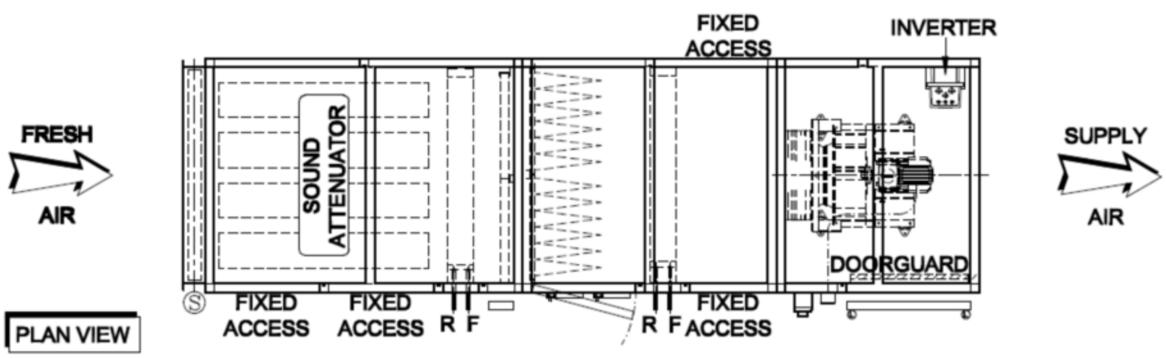
TYPICAL FRESH AIR SUPPLY AIR HANDLING UNITS

These are double skin insulated with thermal and a degree of acoustic insulation. Enhanced performance acoustic composite sound insulated casework is available to meet the requirements of noise sensitive areas.

Typical uses of supply only Air Handling Plant are found, supplying fresh air to kitchen, toilets, fume cupboards, where recirculated air, or heat recovery is not possible.

Sound attenuation may be required to the system and atmospheric side, if the plant is located internally within a noise sensitive area, then an enclosure or false ceiling may be required to reduce the noise breakout.





TYPICAL HEAT RECLAIM & ENVIROFRESH HEAT PUMP AIR HANDLING UNITS

Most Air Handling Units or this type are roof mounted, and have integrated supply and extract fans, which are separated within the Air Handling Unit.

These systems will have energy recovery devices, filtration systems, re-heat and/or pre-heat coils, shut off dampers for static isolation of airflows.

In some instances cooling other than “free cooling” will apply, this will be via either chilled water cooling coils or direct expansion refrigeration coils.

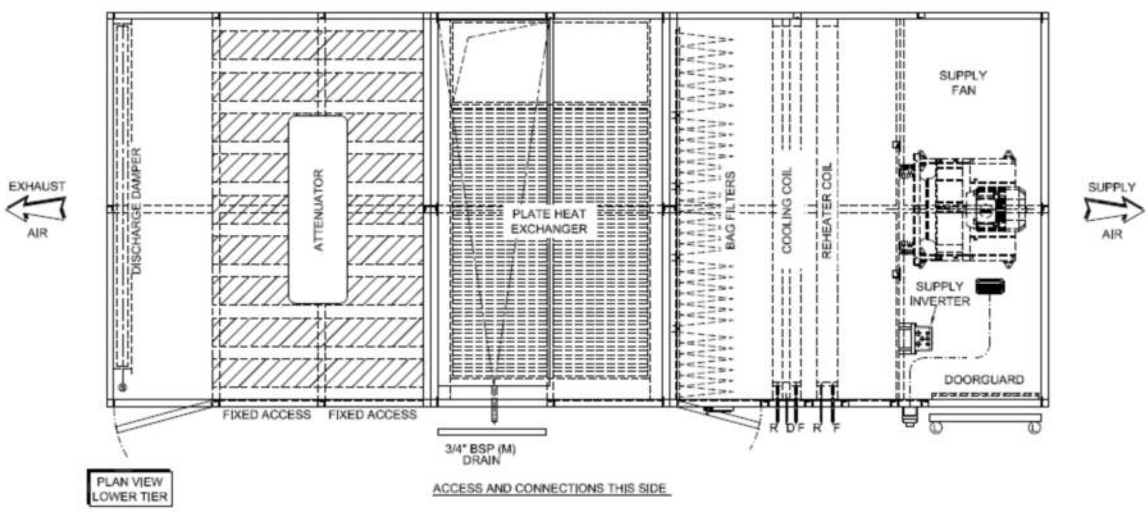
A more advanced method of heating and cooling within an Air Handling Unit, is using Envirofresh Heat Pump Air Handling Unit which contains 70% minimum energy recovery high COP energy efficient digital compressors fully piped within the plant, and an intelligent trend or similar control system fully pre-wired with frequency inverter.

Attenuation of atmospheric and system side noise from the Air Handling Unit may be required, also enhance sound reduction casework may also apply.

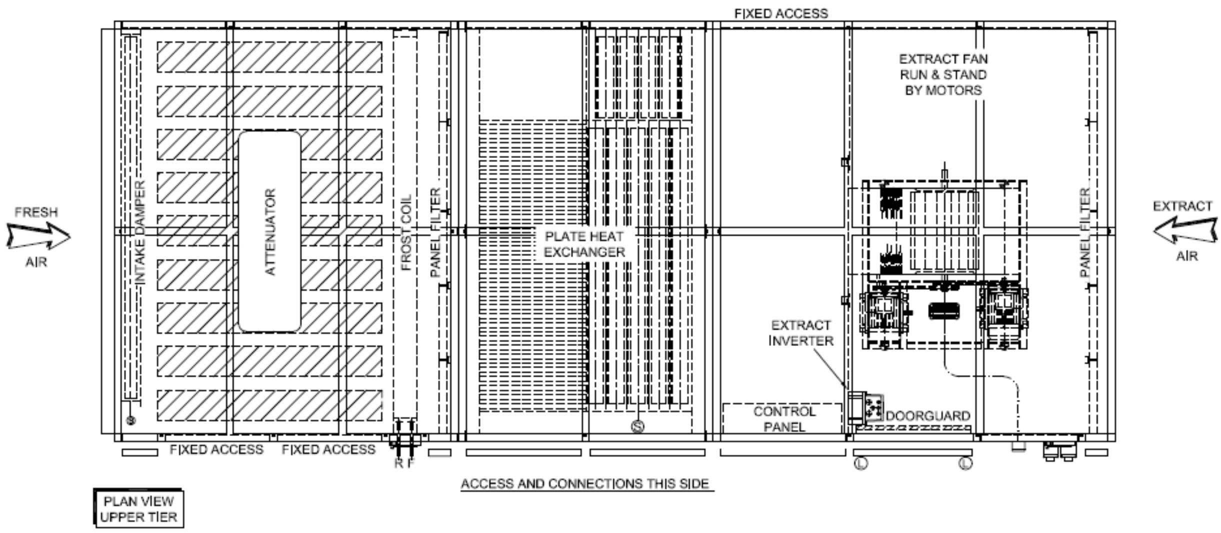
In the case of the Envirofresh Heat Pump Air Handling Unit, on the atmospheric side on external plant, the compressor section may require acoustic treatment with compressor acoustic shells, inertia base and high deflection anti-vibration isolation, also flexible refrigeration connectors. Also compressor chamber absorber panels will reduce internal sound pressure levels.

All Heat Recovery Air Handling Units will need to be checked for noise emission to both the atmospheric side and system side of the supply and exhaust.

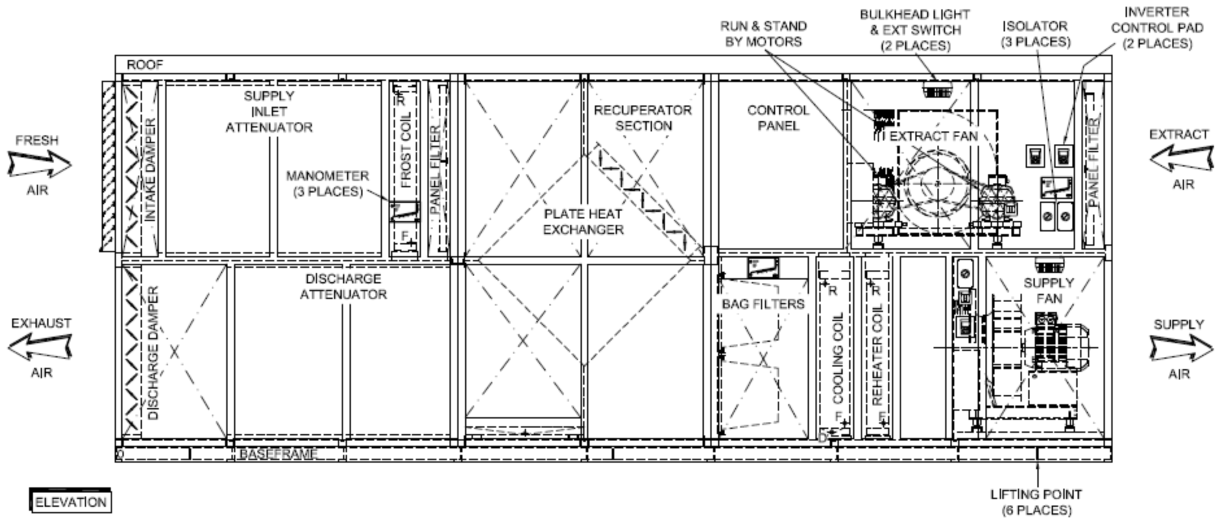
HEAT RECLAIM AIR HANDLING UNITS - PLATE HEAT EXCHANGER



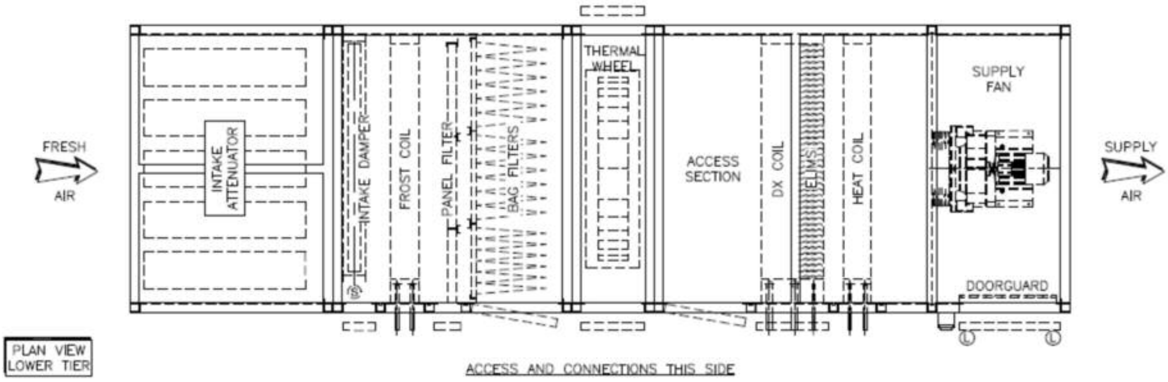
HEAT RECLAIM AIR HANDLING UNITS - PLATE HEAT EXCHANGER



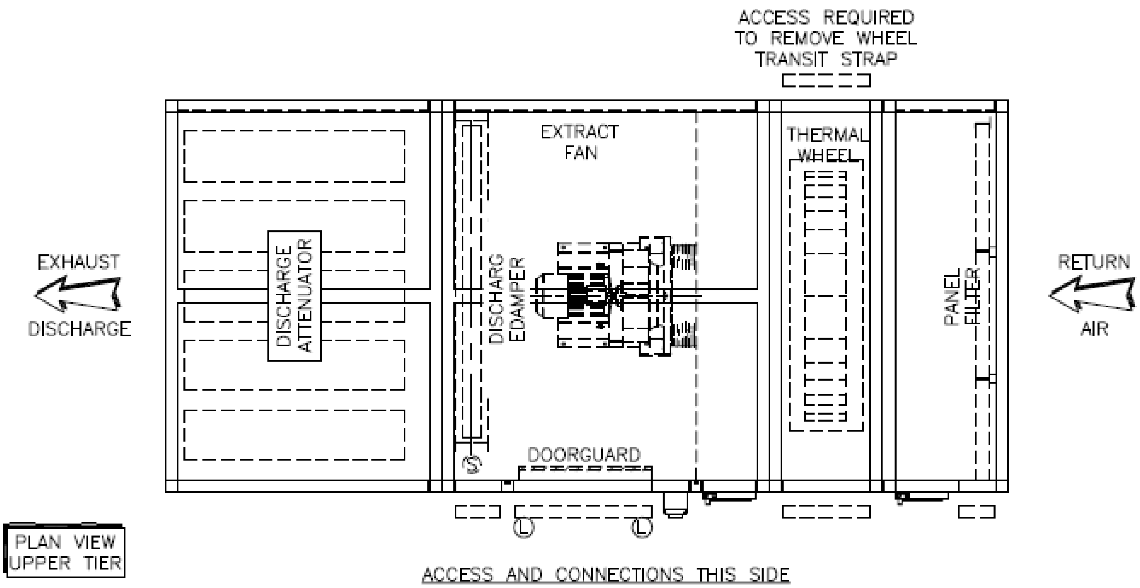
HEAT RECLAIM AIR HANDLING UNITS - PLATE HEAT EXCHANGER



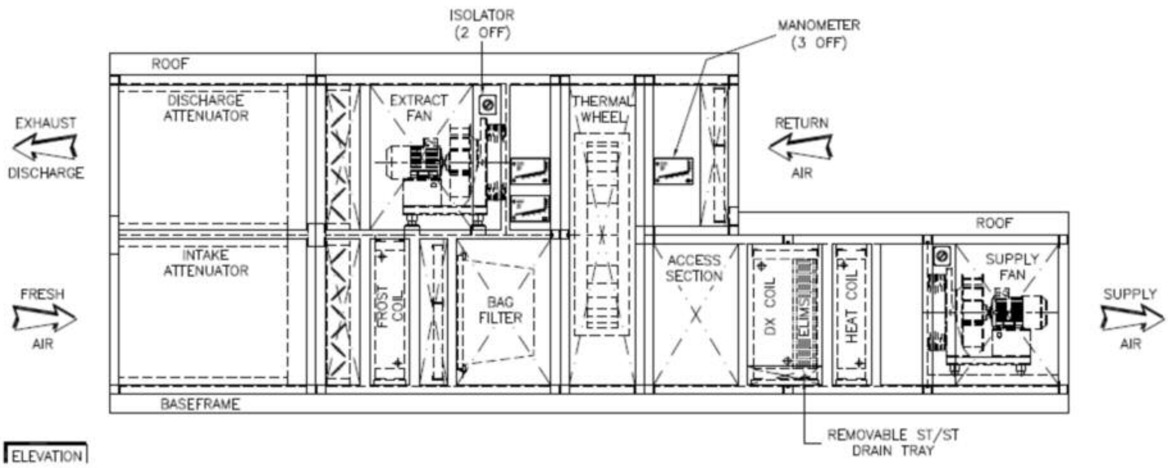
HEAT RECLAIM AIR HANDLING UNIT – ROTARY HEAT EXCHANGER



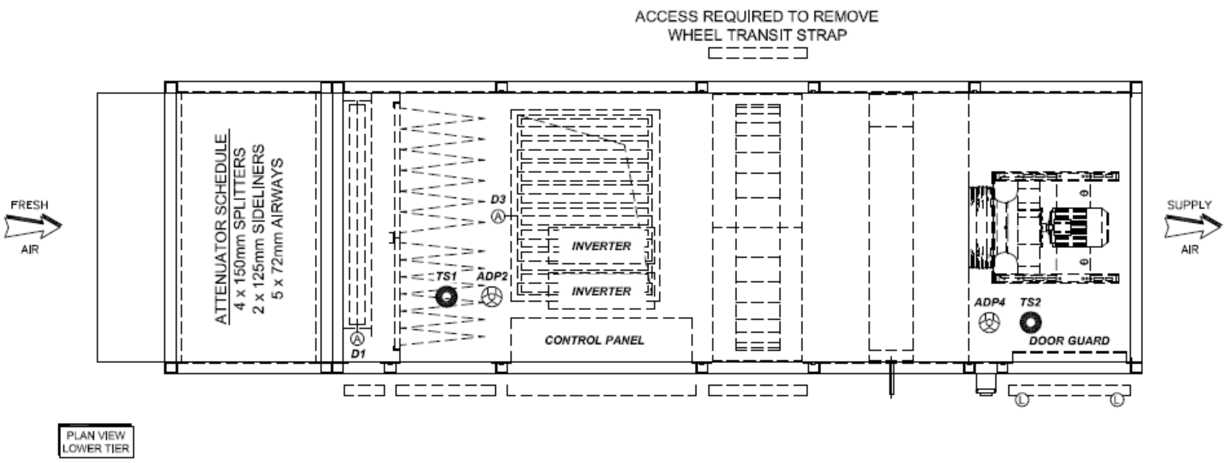
HEAT RECLAIM AIR HANDLING UNIT – ROTARY HEAT EXCHANGER



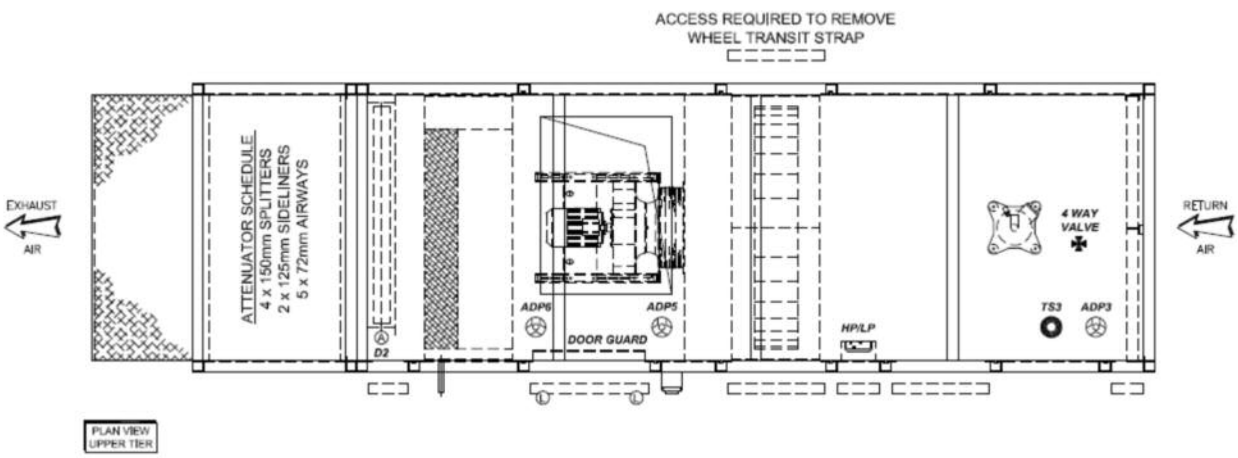
HEAT RECLAIM AIR HANDLING UNIT – ROTARY HEAT EXCHANGER



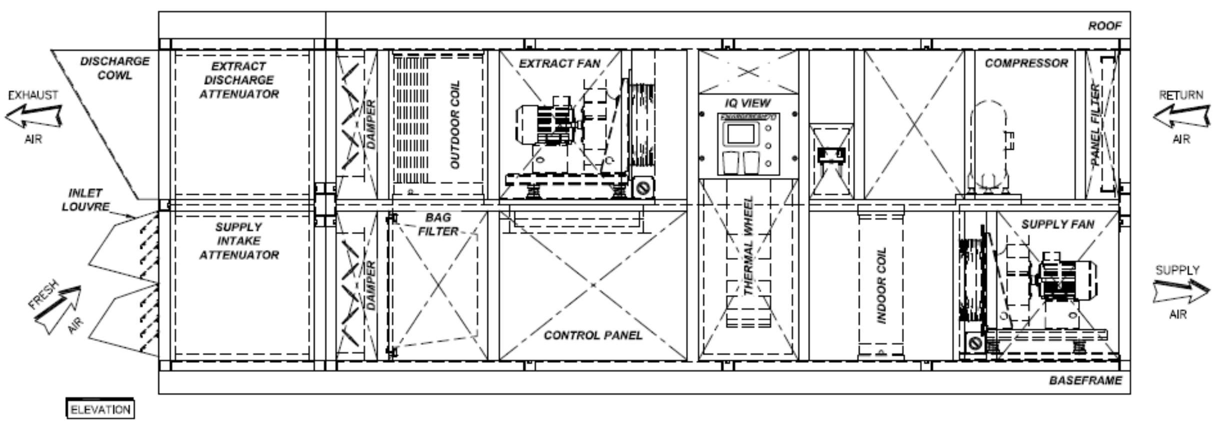
ENVIROFRESH 70



ENVIROFRESH 70



ENVIROFRESH 70



HEAT RECLAIM CLASSVENT (HRU) AIR HANDLING UNITS

These are generally located in classrooms, at high level, in noise sensitive areas. They comprise of Supply and Extract fans with Heat Reclaim via a plate heat exchanger for fresh air operation, and have mixed air or recirculation for fast warm up of the areas.

A summer by pass offers free cooling to the area via a face and by pass damper arrangement actuator operated, with shut off dampers if the plant shuts down.

The Classvent Unit can have a recirculated volume/temperature night setback/or be operated via PIR room sensors. CO_2 detection is also available which will ramp up the extract fan extraction volume.

Due to the noise sensitive location of this type of plant and the reverberant conditions found in classrooms which do not meet the required reverberation time (Tmf (secs)) of building bulletin 93, serious consideration must be given to the noise control of this type of Air Handling Plant.

Duct work external resistances must be kept to a minimum; we would recommend 150 pascals maximum.

Supply and extract fans would be sized to give the lowest sound power level (LW db) available. Enhanced acoustically treated Casework will be required to meet the 35 dbA classroom requirements for schools or NR29 its equivalent in noise rating. This will be in a low reverberant room condition of 0.8 Tmf (secs) in schools and 0.6 Tmf (secs) in Primary Schools.

If the HRU casework does not have enhanced casework sound reduction to decay the fan developed sound pressure level inside the AHU Casework to the room sound pressure level of 35 dbA requirement, additional sound reduction will be required from the following.

A. Additional AHU or room absorption.

B. False ceiling with the required decay. Please take care that the ceiling is sealed with no sound leakage (i.e. using the ceiling as a return air plenum, or lighting which leaks sound).

C. Additional acoustic enclosure to cover the HRU which must be sealed for sound leakage. This solution will increase the overall height of the plant in the ceiling, and will give access difficulty for maintenance of the HRU.

Atmospheric and system side attenuation is normally integral with the HRU to prevent "Flanking". Attenuator, ductwork and air terminal airflow velocities are to be low, to prevent any regenerated noise.

NOISE CONSIDERATION FOR THE DESIGNER TO ACHIEVE BUILDING BULLETIN 93

Atmospheric Noise

Atmospheric noise has normally in schools two considerations for the designer:

1. Noise break back into the building.

2. Plant noise travelling to site boundaries or residencies with restricted day time and night time noise levels.

Atmospheric Sound Decay

Atmospheric sound pressure decay's through air by the well-known inverse square law by which sound pressure level decreases by 6dB per doubling of distance from the source.

Source sound pressure level R_o is SPL_o sound pressure level at point of measurement R_1 is SPL_1 then $SPL_o - SPL_1 = 20LogR_1 – 20LogR_o$ or $SPL_o – SPL_1 = 20Log\ R_1/Ro$ db

The following illustration shows an unimpeded sound source from a louvre of a mechanical ventilation plant and the boundary of the site which will require a much reduced pressure level, see *Illustration 1*.

DIRECT SOUND PRESSURE SOURCE WITH BOUNDARY IMPACT

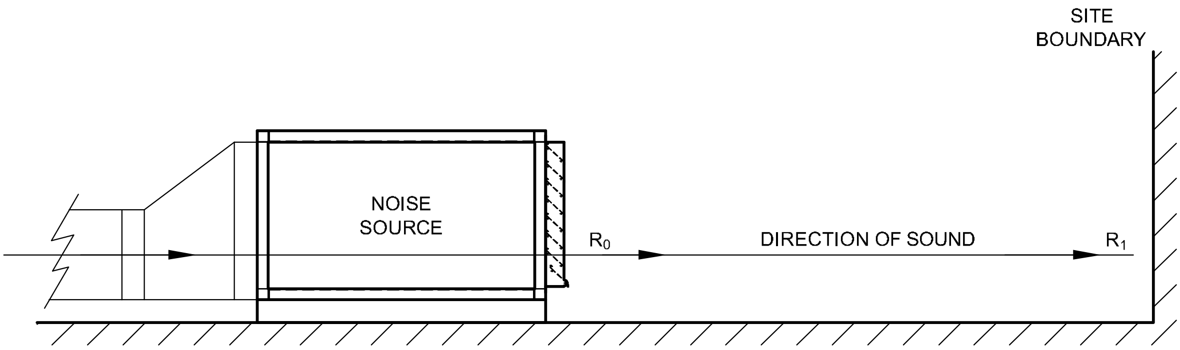


ILLUSTRATION 1

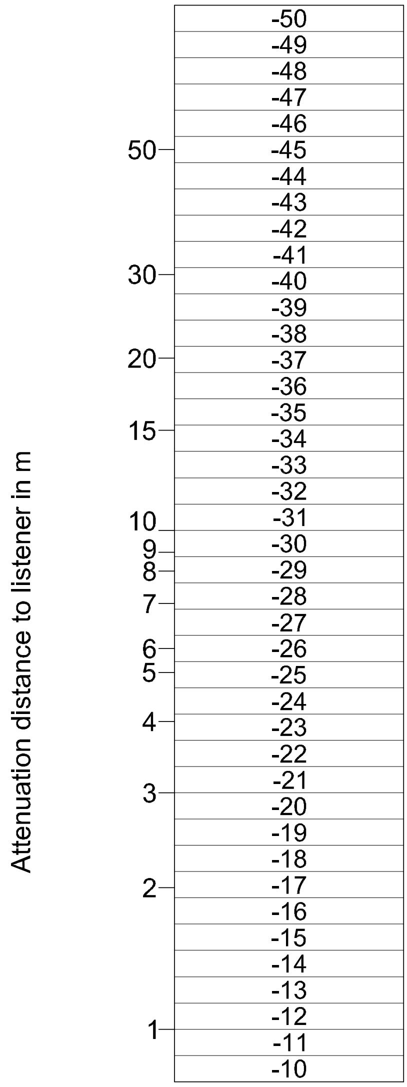
Direct Sound Pressure Source with Boundary Impact

Atmospheric sound can be influenced over distance by the following:

1. Wind Speed and Direction
2. Rain or Barometric Conditions
3. Direction of Sound Path with Regard to the Point Impact R_1
4. Obstructions in the way of the sound path (i.e. Trees, Planting, Walls, Structure, Screens, etc)

Illustration 2 gives attenuation at distance with no external influence from wind, rain, directivity or obstruction.

IMPACT DECAY ON ATMOSPHERIC SOUND PATH



Atmospheric attenuation with distance for a point source of sound.

ILLUSTRATION 2

Impact Decay on Atmospheric Sound Path

As previously indicated, wind, rain, and barometric conditions will influence the attenuation over distance for a point source of sound.

Whilst the effect may be considerable, it will have no influence over the implementation of sound pressure destruction when calculating boundary and neighbourhood plant noise emissions.

Impact of Obstruction Decay on Atmospheric Sound Path

Where there is an obstruction which is permanent, which is in the direct sound path from the noise source to the boundary or point of analysis then this will influence the noise impact calculation.

Care must be taken when assessing the impact decay, which can be influenced by the size and shape of the obstruction, also its distance from source and any absorption or reflection it may have.

The typical effect of obstruction to sound pressure direct sound path is shown in *Illustration 3*. Also in *Illustration 4* the effect of an obstruction containing absorption, and reflective properties is shown.

INFLUENCE OF OBSTRUCTIONS

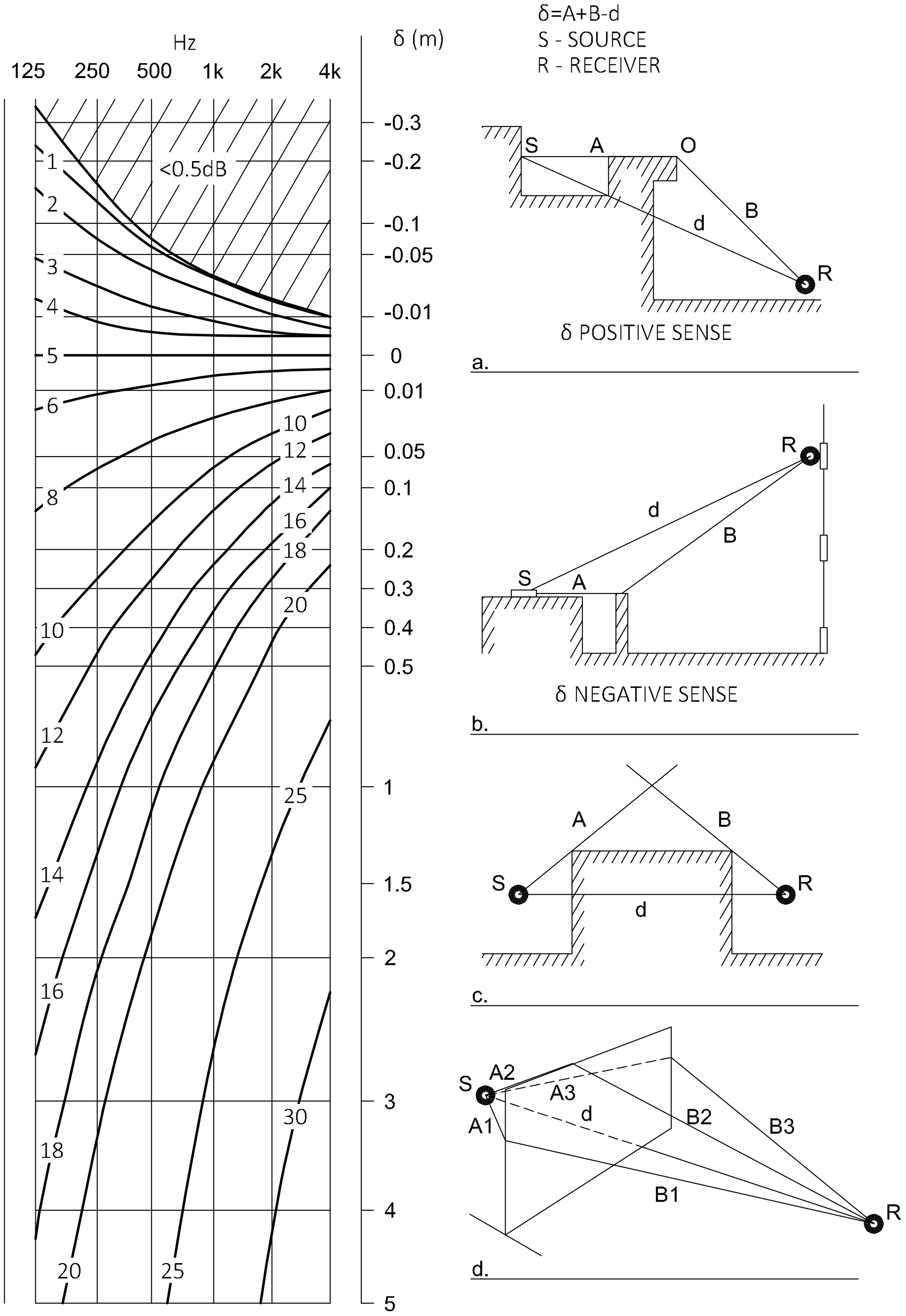


ILLUSTRATION 3

DISTRIBUTION OF ENERGY FROM AIRBORNE SOUND, STRIKING A BARRIER

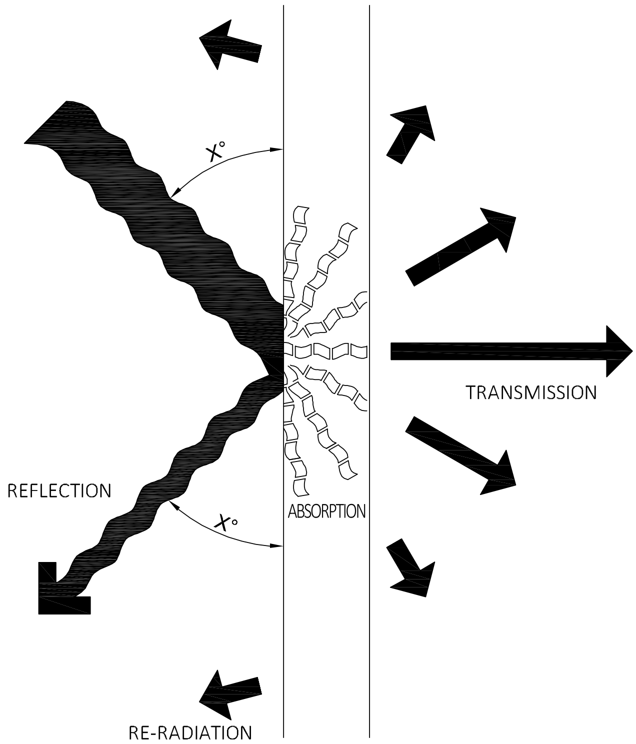


ILLUSTRATION 4

Where screens or acoustic screens are used to reduce sound pressure levels *Illustration 5* shows different arrangements. There is additional absorption decay if acoustic treatment is applied to the screen by the manufacturer which can be deducted off the sound pressure level.

DIFFERENT SCREENING ARRANGEMENTS

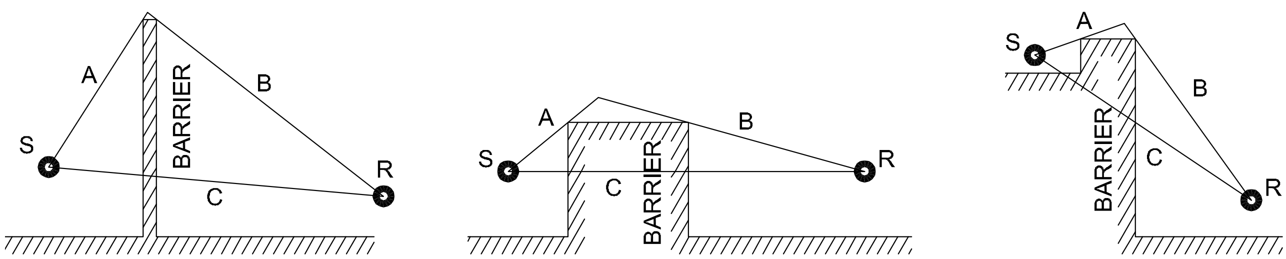


ILLUSTRATION 5

The influence of the screen is given by the results of S = A + B - C in *Illustration 6* (Table). The attenuation value is given in dB across the octave bands 63Hz to 8000Hz.

<u>Illustration 6</u>

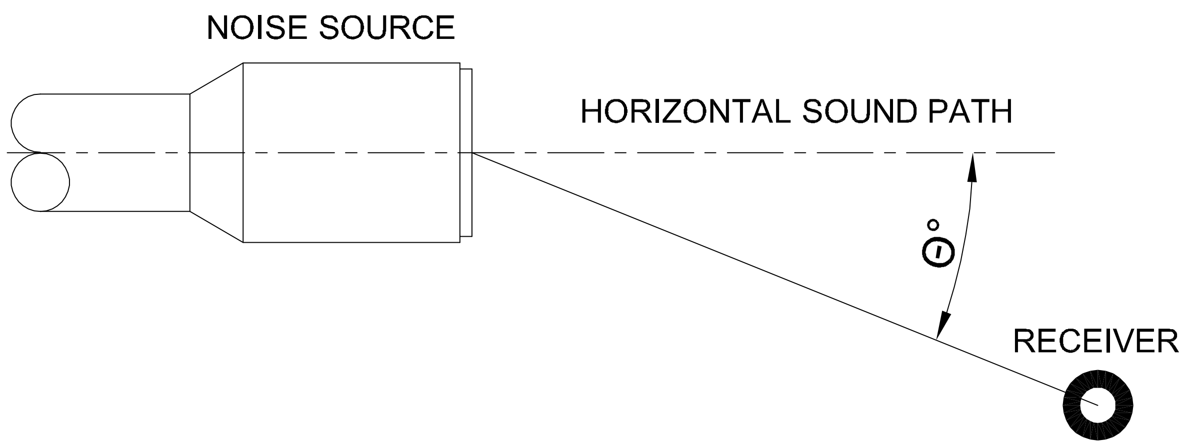
Additional Attenuation due to Screens: S = A + B – C (Metres)								
S(m)	**Octave Centre frequency, fm in Hz**							
	63	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
-0.3	1	0	0	0	0	0	0	0
-0.2	2	1	0	0	0	0	0	0
-0.1	3	2	1	0	0	0	0	0
-0.05	3	3	2	1	0	0	0	0
-0.01	4	4	4	3	3	2	1	0
0	5	5	5	5	5	5	5	5
0.01	5	6	6	6	7	8	8	9
0.05	7	7	8	9	10	12	13	15
0.1	7	8	9	10	11	14	16	18
0.2	8	9	10	11	14	16	19	20
0.3	8	9	10	13	16	18	20	22
0.4	9	10	12	14	17	20	22	24
0.5	9	10	12	15	18	20	23	25
1.0	11	12	14	18	20	23	25	27
1.5	13	14	16	19	22	25	27	30
2.0	14	15	18	20	24	27	29	31
3.0	15	17	20	22	25	28	30	32
4.0	16	18	20	24	26	30	31	33
5.0	16	18	21	25	27	30	32	34

<u>The Effect of Sound at the Receiver by Directional Change</u>

A further influence on sound pressure decay to the listener or point of sound is directivity.

The angle the noise source is directed at the point of sound has a directivity effect on the decay. *Illustration 7* shows a noise source in respect to the receiver.

<u>DIRECTIVITY</u>



<u>ILLUSTRATION 7</u>

Directivity

As the angle Θ increases from the horizontal sound path, there will be an influence in the final sound pressure level at the receiver.

This reduction is given in *Illustration 8,* directivity index, which gives the relationship of intake/outlet area in square metres to octave band frequency from 63Hz to 8000Hz.

Directivity Index

Frequency Hz	Inlet/Outlet Area (m^2)													
	0.25	**1**	**2.5**	**5**	**8**	**14**	**20**	**30**	**40**	**60**	**90**	**120**	**160**	**250**
63	4	5	6	6	7	7	8	8	8	8	8.5	8.5	9	9
125	5	6	7	7	8	8	8	8.5	8.5	9	9	9	9	9
250	6	7	8	8	8.5	9	9	9	9	9	9	9	9	9
500	7	8	8.5	9	9	9	9	9	9	9	9	9	9	9
1k	8	8.5	9	9	9	9	9	9	9	9	9	9	9	9
2k	8.5	9	9	9	9	9	9	9	9	9	9	9	9	9
4k	8.5	9	9	9	9	9	9	9	9	9	9	9	9	9
8k	9	9	9	9	9	9	9	9	9	9	9	9	9	9

Illustration 8

The directivity factor *Illustration 9* takes the directivity index from *Illustration 8* for each octave band frequency and converts the degree of directivity angle (Q)

Directivity Factor, dB

Index	Directivity Angle Q									
	0^o	**20^o**	**40^o**	**60^o**	**70^o**	**80^o**	**90^o**	**100^o**	**120^o**	**140^o**
4	+4	+4	+3	+3	+2	+2	+2	+2	+1	0
5	+5	+5	+4	+3	+2	+1	+1	+1	0	-2
6	+6	+6	+4	+3	0	-1	-1	-1	-3	-6
7	+7	+6	+5	+2	0	-1	-2	-4	-9	-15
8	+8	+7	+5	+2	-1	-4	-7	-13	-30	-45
8.5	+8.5	+7	+5	+1	-8	-16	-20	-28	-40	-50
9	+9	+8	+6	0	-15	-25	-35	-45	-50	-60

This table may also be used with stacks, up to a 120^o angle from the axis

Illustration 9

Calculating Receiving Sound Pressure Levels

To calculate the sound pressure level at a boundary, receiver or neighbourhood window etc, Illustration 9 gives the directivity Factor dB, which is used in the overall sound pressure level calculation.

Directivity Parameters

The directivity of a source, i.e. the fact that more sound is radiated in one direction than another, depends on the size of the source and the frequency of the sound. As the frequency increases, the directivity of a source of a particular size increases.

Similarly, for constant frequency the directivity increases as source size increases. The directivity is different in the horizontal and vertical directions for a rectangular source such as a louvre, cowl or open window because of the different effective size of source (width and height).

End Reflection

End reflection occurs on louvres and cowls both supply and exhaust. The inlets/outlets can be fascia mounted on walls of the building where the noise source is within the building, or on mechanical ventilation plant mounted externally.

The fact that the wavelength of the sound may be comparable with the dimensions of the inlet/outlet terminal radiating the sound means that not all the sound is radiated, particularly at low frequency.

This end correction loss must be allowed for and a convenient way of estimating this is given in *Illustration 10* for each of the lower frequency octave bands.

End reflection loss values shown in *Illustration 10,* are based on louvres/cowls positioned in walls of the building. If the louvre/cowl is at the atmospheric terminal end of a mechanical ventilation plant then approximately 10% of the end reflection loss will be lost due to the fascia decrease.

END REFLECTION LOSS

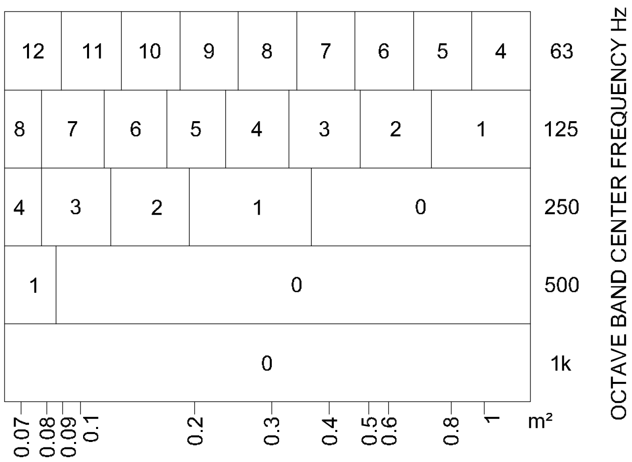


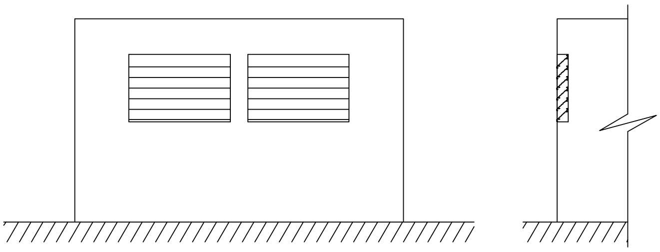
ILLUSTRATION 10

Influence of Multiple Inlets & Outlets

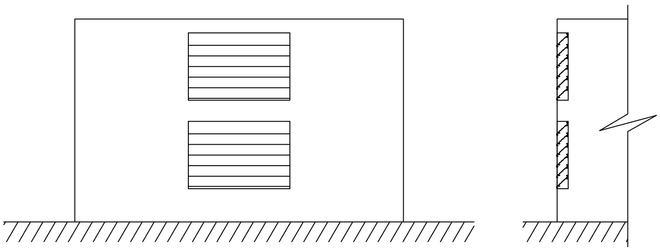
When there are several inlets and outlets in close proximity of each other, then this may influence the overall sound pressure level of the total sound emission.

The inlets and outlets can be side by side or double stacked, and possibly both. *Illustration 11* shows various types of mechanical ventilation plant examples.

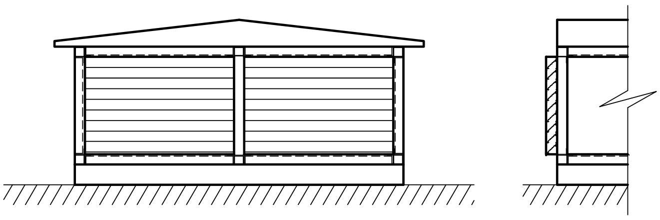
MULTIPLE INLETS & OUTLETS



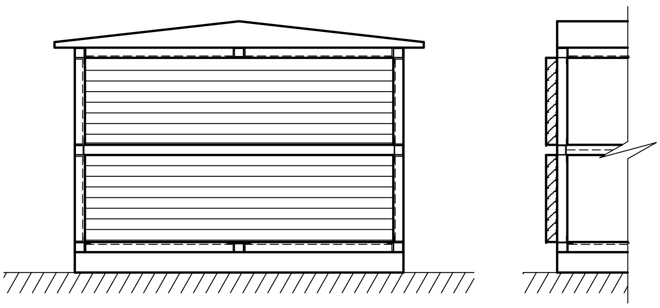
EXAMPLE 1 - TWO LOUVRES, FLUSH MOUNTED ON A BUILDING WITH A SIDE BY SIDE ASPECT



EXAMPLE 2 - TWO LOUVRES, FLUSH MOUNTED ON A BUILDING WITH A DOUBLE STACKED ASPECT



EXAMPLE 3 - TWO LOUVRES MECHANICAL VENT PLAN, MOUNTED WITH SIDE BY SIDE ASPECT



EXAMPLE 4 - TWO LOUVRES MECHANICAL VENT PLAN, MOUNTED WITH STACKED ASPECT

ILLUSTRATION 11

Two Louvres Mechanical Vent Plant Mounted with Double Stacked Aspect

To calculate the two sound sources operating together then the following formula gives the resultant Sound Pressure Level (SPL).

$$Total\ \ SPL = 10\log\frac{{P_1}^2 + {P_2}^2}{{P_{ref}}^2}$$

SPL_1 is the Sound Pressure Level P
SPL_2 is the Sound Pressure Level 2P

The following chart *Illustration 12* gives the addition of two Sound Pressure or Sound Power Levels.

Illustration 12

Difference between Two Sound Levels dB	Add to Higher Level dB
0	3
1	2.5
2	2.1
3	1.8
4	1.5
5	1.2
6	1
7	0.8
8	0.6
9	0.5
10 or more	0

‘A’ Weighting SPL

Local authorities will impose noise level constraints for planning requirements on neighbourhood boundaries for daytime and night time. This will be dbA (Leq) rating which is a ‘A’ weighted scale over a period of time.

The sound pressure level will be dBA, which will require the octave band levels converting to one ‘A’ weighted combined overall eight octave band dbA level.

Illustration 13 outlines ‘A’ weighting relative response (db) which is used in *Illustration 14* to determine dBA.

Illustration 13

Centre Frequency Hz	63	125	250	500	1k	2k	4k	8k
SPL db	x	x	x	x	x	x	x	x
‘A’ Weighting	-26	-16	-9	-3	0	+1	+1	-1
Corrected SPL(dBA)	(x-26)	(x-16)	(x-9)	(x-3)	(x)	(x+1)	(x+1)	(x-1)

Example

If we take a typical Sound Level from a louvre after end correction as below then an example of correction to dbA is shown in Example 1.

OVERALL WEIGHTED "A" SPL CALCULATION

CENTRE FREQUENCY Hz	63	125	250	500	1000	2000	4000	8000
SOUND PRESSURE LEVEL (SPL), dB	85	88	89	84	81	77	72	67
"A" WEIGHTINGS	-26	-16	-9	-3	0	1	1	-1
CORRECTED SPL, dB(A)	59	72	80	81	81	78	73	66

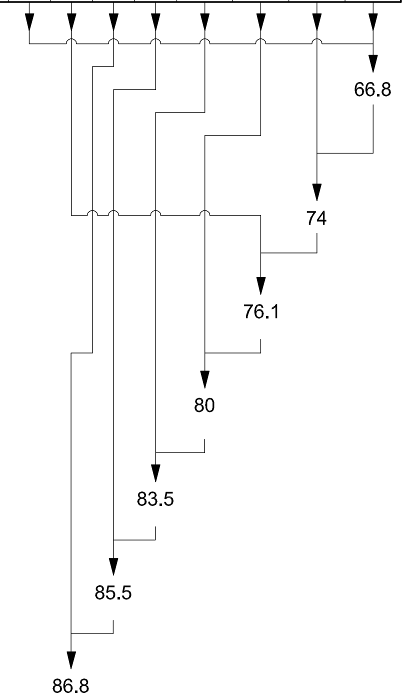


ILLUSTRATION 14

Overall Weighted SPL 86.8dbA

Example 1 – Corrected 'A' Weighting Calculation

1. **66 – 59 = 7(0.8) = 66 + 0.8** **66.8**
2. **73 – 66.8 = 6.2(1.0) = 73 + 1.0** **74**
3. **74 – 72 = 2(2.1) = 74 + 2.1** **76.1**
4. **78 – 76.1 = 1.9(2.0) = 78 + 2.0** **80**
5. **81 – 80 = 1(2.5) = 81 +2.5** **83.5**
6. **83.5 – 81 = 2.5(2.0) = 83.5 +2.0** **85.5**
7. **85.5 – 80 = 4.5(1.3) = 85.5 + 1.3** **86.8dBA**

Calculation of SPL at the Receiver

The calculation sheet *(Illustration 15*) deducts the attenuation from the SWL leaving the source to the receiver; this will give the sound pressure in octave band db levels.

The noise level at each frequency from 63Hz to 8000Hz can be compared with noise rating curves (NR) or converted by using the charts on *Illustrations 13* and *14* into dBA.

Both NR curves and dBA levels are used to determine limits at the receiver, *Illustration 16* shows NR curves which has a 1000Hz threshold.

Illustration 15

Atmospheric Side Calculation Sheet

Project

Fan Duty

System Supply/Extract

Frequency Hz	**63**	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
A. SWL leaving system								
B. Addition of 2 sources								
C. Directivity								
D. Barrier/Screen Loss etc								
E. Distance from Source								

A - [B + C + D + E]

The outcome of A - [B + C + D + E] will give the SPL at the receiver, and can be compared with the corresponding noise rating curve.

If the noise level at the receiver is required in dbA then the sound pressure levels will need to be converted to 'A' weighting – see *Illustration 14*.

NR Curves (Illustrations 16 & 17)
The dBA and NR curves are subjective units which give us a representation of how the ear assesses noise. We then have to fix limiting values to them for different environments and people.

NOISE RATING NUMBER NR

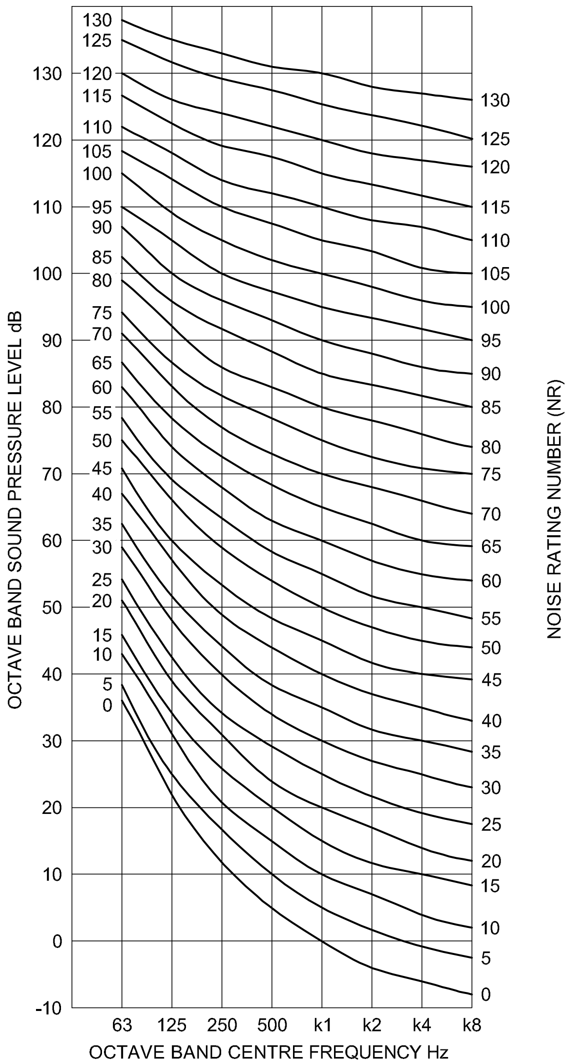


ILLUSTRATION 16

Maximum Sound Pressure Level (dB)									
Noise Rating -NR-	Octave band mid-frequency (Hz)								
Curve	31.5	62.5	125	250	500	1000	2000	4000	8000
NR 0	55	36	22	12	5	0	-4	-6	-8
NR 10	62	43	31	21	15	10	7	4	2
NR20	69	51	39	31	24	20	17	14	13
NR 30	76	59	48	40	34	30	27	25	23
NR 40	83	67	57	49	44	40	37	35	33
NR 50	89	75	66	59	54	50	47	45	44
NR 60	96	83	74	68	63	60	57	55	54
NR 70	103	91	83	77	73	70	68	66	64
NR 80	110	99	92	86	83	80	78	76	74
NR 90	117	107	100	96	93	90	88	86	85
NR 100	124	115	109	105	102	100	98	96	95
NR 110	130	122	118	114	112	110	108	107	105
NR 120	137	130	126	124	122	120	118	117	116
NR 130	144	138	135	133	131	130	128	127	126

Illustration 17

Tonal Influence

When the noise at the receiver has a predominant peak level similar to a pure tone or predominance over a small frequency range, this will cause annoyance at the receiver, also if the peak level is periodic, this increases the annoyance.

If this occurs or is likely to occur then +5db must be added to the sound pressure level or correspondingly -5db reduced off the target noise level.

Internal Room Acoustics

The objective is to provide suitable indoor ambient noise levels, as stated in Building Bulletin 93 (1.1.1).

a. For clear communication of speech between teacher and student, and between students.

b. For study activities.

The ambient noise level contributes from:

1. External sources outside the school premises including road, rail and air traffic noise.
2. Building Services noise generated by mechanical ventilation plant which may be located outdoor or indoor.
3. In room noise consisting of computer server noise, or ventilation plant noise (Classvent HRU Unit), and other sources listed in clause 1.1.1.

Room Requirements

A. An adequate amount of sound must reach all parts of the room. Most attention in this respect needs to be given to those seated furthest from the source.

B. An even distribution of sound throughout the room irrespective of distance from the source.

C. Other noise sources which could mask the required sound level have to be reduced to a level as to not increase the sound pressure level above the room classification (ie: classroom 35dBLeq (30 mins).

D. The rate of decay of sound within the room (reverberation time (Tmf)) must be the optimum for the required room use (ie: classroom (0.8 Tmf secs). This is to ensure clarity for speech or fullness for music.

Behaviour of Sound

Sound can be reflected in a similar way to light, the angle of incidence being equal to the angle of reflection. However, it must be remembered that for this to be true, the reflecting object must be at least the same size as the wave length concerned. It can be very useful to carry out a limited geometrical analysis. This can prevent the problem of long delayed reflection and focusing effects. It is impractical to take a geometrical analysis beyond the first or second reflections but it can prevent design errors.

An example of errors is the focusing effect of concave shapes which may produce places with loud sounds or dead spots. (See *Illustration 18*).

SOUND EFFECT ON NON PLANAR SURFACES

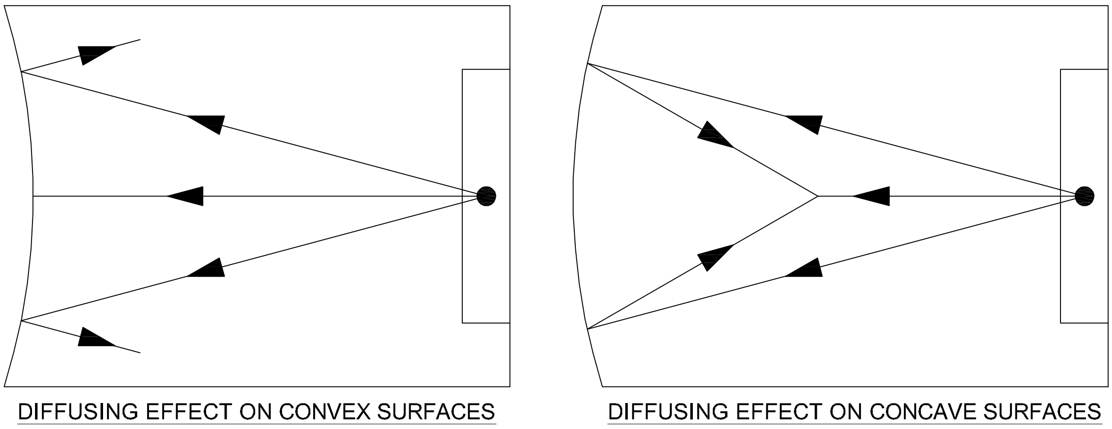


ILLUSTRATION 18

Average speech is at the approximate rate of 15 to 20 syllables per second or roughly 70 to 50 ms respectively. This corresponds to a delay of approx. 17 metres.

A member of a class sitting at 8.5 metres from a reflective rear wall will find it difficult to understand speech.

How to Achieve Compliant Room Acoustic Condition

To achieve the room acoustic requirement laid down in BB93, the design of the room must contain adequate absorption to deliver the reverberant condition in the room as laid down in the Acoustic Performance Requirement of Building Bulletin 93, for the room acoustic performance of the various rooms and areas.

Type of Room	**Tmf (seconds)**
Nursery school playrooms Nursery school quiet rooms	<0.6 <0.6
Primary school: classrooms, class bases, general teaching areas, Small group rooms	<0.6
Secondary school: classrooms, general teaching areas, seminar Rooms, tutorial rooms, language laboratories	<0.8
Open-plan Teaching areas Resource areas	 <0.8 <1.0
Music Music classroom Small practice/group room Ensemble room Performance/recital room Recording studio Control room for recording	 <1.0 <0.8 0.6 – 1.2 1.0 – 1.5 0.6 – 1.2 <0.5
Lecture Rooms Small (fewer than 50 people) Large (more than 50 people)	 <0.8 <1.0
Classrooms designed specifically for use by hearing impaired Students (including speech therapy rooms)	<0.4
Study room (individual study, withdrawal, remedial work, Teacher preparation)	<0.8
Libraries	<1.0
Science laboratories	<0.8
Drama studio	<1.0
Design and Technology • Resistant materials, CADCAM areas • Electronics/control, textiles, food, graphics, design/resource areas	 <0.8 <0.8
Art rooms	<0.8
Assembly halls, multi-purpose halls (drama, PE, audio/Visual presentations, assembly, occasional music)	0.8 – 1.2
Audio-visual, video conference rooms	<0.8
Atria, circulation spaces used by students	<1.5
Indoor sports hall	<1.5
Gymnasium	<1.5
Dance studio	<1.2
Swimming pool	<2.0
Interviewing/counselling rooms, medical rooms	<0.8
Dining rooms	<1.0
Ancillary spaces Kitchens Offices, staff rooms Coats and changing areas Toilets	 <1.5 <1.0 <1.5 <1.5

Reverberation Time and its Effect on Rooms

After being emitted from a source, sound waves are repeatedly reflected from the room surfaces and, as a result of absorption, gradually reduce or decay in strength. The reverberation time (T) of a space is a measure of the rate at which the sound decay's. It is defined as "the time taken for the reverberant sound energy to decay to one millionth of its original intensity". (Corresponding to a 60dB Decay in sound level).

Reverberation time is certainly one of the most important criteria. Sound does not die away the instant it is produced but will continue to be heard for some time because of reflections from walls, ceiling, floors etc. It will mix with later direct sound which is known as reverberant sound.

The other elements that effect reverberation time are the room volume and absorption of materials within the room which absorb sound.

To calculate reverberation time in a given room, Sabine Formula is used.

Sabine Formula

$$T = \frac{0.16V}{A}$$

Where t = Reverberation time in seconds
V = Room volume in M^3
A = Absorption in M^2

Note: Absorption (A) can be determined as Mean Absorption Coefficient.

The general performance of a room in relation to reverberation is given as:- Live/Average/Dead. *Illustration 19/20* shows the relation between reverberation time, and correction for room volume.

CORRECTION FOR REVERBERATION TIME

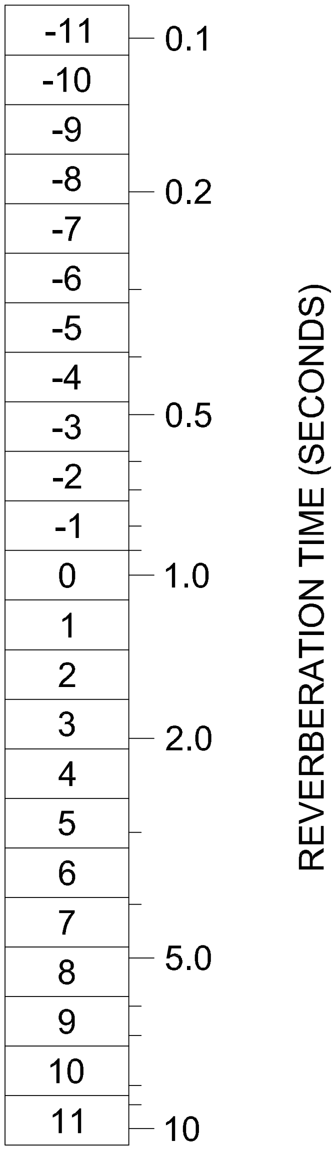


ILLUSTRATION 19

CORRECTION FOR ROOM VOLUME

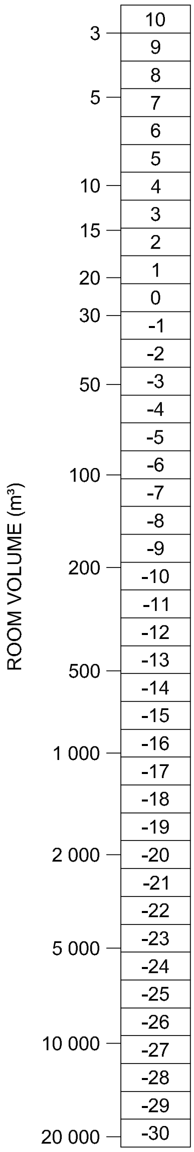


ILLUSTRATION 20

Example

Based on a classroom having overall dimension of 15m Long x 10m Wide x 3.3m High to false ceiling with a room reverberation time of 0.8 seconds Tmf, the noise source is a Heat Reclaim Mechanical Ventilation Unit situated behind the false ceiling and emitting 40dB @ 250Hz from the ceiling.

Room Volume	**15 x 10 x 3.3 = 495m^3**
Reverberation Time	**0.8 Seconds (Classroom)**
Plant Emission @ Source	**40dB @ 250Hz**
Sound Pressure Level	**40db**
Correction for Room Volume (Illustration 20) 495m^3	**-13db**

Correction for Reverberation Time (Illustration 19) 0.8 Seconds	**-1db**
Reverberant Component	**26db**

Absorption in Rooms

All rooms will have a degree of absorption; most construction materials used to construct a room in a school will absorb sound energy.

Rooms in schools are constructed not only to stop noise transferring from higher noise activity areas, but to absorb sound in the room itself.

Most construction materials fall into the porous absorber category, which include carpets, draperies, aerated plaster, fibrous mineral wool and glass fibres, open cell foam and porous ceiling tiles. All of these materials allow air to flow into a cellular structure where sound energy is converted into heat.

To experimentally take sound absorption measurements using a plane wave impedance tube, a plane sound wave is put on an acoustic absorber. Some energy is absorbed and some energy is reflected, if the pressure (P_1) is the incident wave is:

$$P_1 = \mathrm{A}\ \cos(2\pi f t)$$

The Reflective Wave (P_2)

$$P_2 = B\cos(2\pi f t)\left(t - 2\frac{x}{c}\right)$$

The total sound pressure in the tube can be measured by a microphone and is given by the sum of the two. Here f is the frequency (Hz), t is the time, x is the distance from the sample surface (m), c is the velocity of sound (m/s) and A and B are the amplitudes.

The proportion of sound absorbed by the surface is called the sound absorption coefficient. The sound absorption coefficient is represented by **a .**

$$\alpha = 1 - \left(\frac{B}{A}\right)^2$$

The coefficient can be viewed as a percentage of sound being absorbed. Where 1.00 is complete absorption (100%) and 0.01 is minimal absorption (1%).

Sound Absorption by Materials

Sound waves are reflected when they hit hard surfaces. Providing an absorbent surface can reduce some of the reflected sound. In a “Live” room porous materials will reduce noise by reducing the reflected sound. Only reflected sound can be treated, while direct sound will not be affected.

The correct selection of porous materials with varying density and composition is critical in the choice of absorber and maintain the best absorption values. The air channels should all be open to the surface so that sound waves can propagate into the material.

If pores are sealed, as in closed cell foam or painted surfaces, if material covering is applied as “protective shielding” for the absorber material then it must be perforated or porous.

Absorption Coefficient of Various Materials

The following listings for absorption coefficients are a guide as other factors will influence actual absorption.

a. Method of Mounting
b. Thickness
c. Decorative Surface Treatment

(See *Illustration 21*)

All materials have absorption coefficient that vary at different frequencies and absorption coefficients in manufactures published literature at 125, 250, 500, 1k, 2k, 4k Hz.

Often when discussing the general performance of a material and its absorption coefficient the performance at 500Hz is used as a benchmark.

The mean absorption coefficient of a room at a given frequency is defined as:-

$$\frac{\sum S_1 \propto + \sum S_2 \propto_2 + \sum S_3 \propto_3}{\sum S}$$

Illustration 21

Material	Absorption Coefficients					
	125	250	500	1K	2K	4K
Fibre Board						
Solid Backing	**0.05**	**0.1**	**0.15**	**0.25**	**0.3**	**0.3**
With 25mm airspace	**0.3**	**0.3**	**0.3**	**0.3**	**0.3**	**0.3**
Glass						
3mm Thick	**0.2**	**0.15**	**0.1**	**0.07**	**0.05**	**0.02**
6mm Thick	**0.1**	**0.07**	**0.04**	**0.03**	**0.02**	**0.02**
Cork 25mm	**0.05**	**0.1**	**0.2**	**0.55**	**0.6**	**0.55**
***Plaster* Gypsum**						
Solid Backing	**0.03**	**0.3**	**0.02**	**0.03**	**0.04**	**0.05**
In studded wall with airspace	**0.3**	**0.15**	**0.1**	**0.05**	**0.04**	**0.05**
Wood/Lino/Rubber	**0.02**	**0.04**	**0.05**	**0.05**	**0.1**	**0.05**
Flooring						
Wood Panelling 15mm on 25mm battons	**0.31**	**0.33**	**0.14**	**0.1**	**0.1**	**0.12**
Carpets						
Corded on felt	**0.1**	**0.15**	**0.25**	**0.3**	**0.3**	**0.3**
Pile and thick felt	**0.07**	**0.25**	**0.5**	**0.5**	**0.6**	**0.65**
Floor tiles (Rubber)	**0.05**	**0.05**	**0.1**	**0.1**	**0.05**	**0.05**
Curtains						
Medium in folds	**0.05**	**0.15**	**0.35**	**0.55**	**0.65**	**0.65**
Medium straight with backing	**0.05**	**0.1**	**0.15**	**0.2**	**0.25**	**0.3**

Ceilings **Proprietary ceiling tile 20mm thick** **Fixed to solid backing**						
Mineral Wool/Fibre	**0.1**	**0.25**	**0.7**	**0.85**	**0.7**	**0.6**
Wood Fibre	**0.2**	**0.3**	**0.65**	**0.6**	**0.6**	**0.6**
Perforated metal tile 30mm thick absorbent infil	**0.2**	**0.55**	**0.8**	**0.8**	**0.8**	**0.75**
Suspended Ceilings						
Mineral Wool/Fibre	**0.5**	**0.6**	**0.65**	**0.75**	**0.8**	**0.75**
Wood Fibre	**0.4**	**0.5**	**0.55**	**0.65**	**0.75**	**0.7**
Perforated metal tile 30mm thick absorbent infil	**0.25**	**0.55**	**0.85**	**0.85**	**0.75**	**0.75**

Room Affect

All rooms are affected by the sound pressure waves that reach the listener along two paths.

A. Direct Sound Pressure Level reducing as the $(\text{Distance})^2$ from the noise source and is the direct transmission of sound power from the source to the observer.
B. Multiple reflections off the room surfaces and room absorption contents, which depend upon the size of the room and the reverberation time, this is known as the Reverberant Sound Pressure Level.

   This is set up by the total sound power fed into the room by the ventilation plant room sources, both case work radiated and terminal inlets and outlets.

   A further parameter is the extent to which this sound power is reflected around the room or absorbed.

Assessing Room Absorption

By assessing the mean absorption coefficient $\overline{\alpha}$ (*Illustration* 22) and the effect of the area of the floor, walls and ceiling (Illustration 23) the reverberant sound pressure can be obtained by the addition or subtraction of the correction values.

Consideration must be given to the distance in the room from the sound source to the listener, which is normally 1.5 meters unless stated otherwise in the specification.

The area of the noise source will also have a bearing on the room condition; larger noise source areas give more predominance’s than smaller noise source areas.

MEAN SOUND ABSORPTION COEFFICIENT

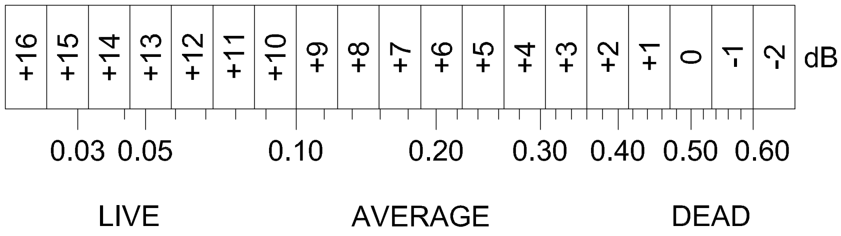


ILLUSTRATION 22

EFFECT OF THE AREA OF FLOOR, WALLS, AND CEILING

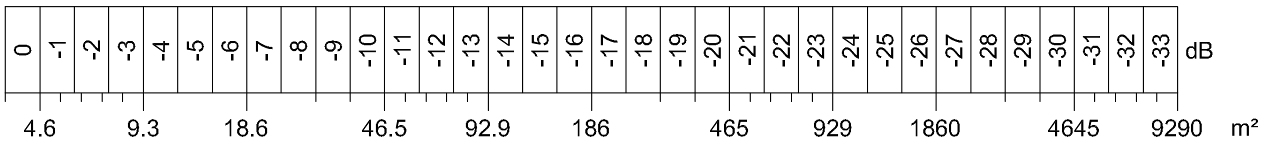


ILLUSTRATION 23

The sound pressure level can be determined in a room based on the directional characteristics of the source and the degree of sound absorption provided by the room surfaces. Q the directivity factor of the source in the direction of interest, is defined as the ratio of sound power output in that specific direction to the mean output over all directions. R the room constant is a measure of the room's total capacity for absorbing sound. **a** is a symbol for the random incidence absorption coefficient of a material.

This is defined as the proportion of incident energy arriving from all directions which is not reflected back into the room by the material. Thus **a** = (1.0) represents total absorption and **a** = (0.0) total reflection.

$$SPL = SWL + 10\log_{10}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right)$$

Where **Q = Directivity Factor of Source**
r = Distance to Receiver (m)
R = Room Constant = $\dfrac{S\bar{\alpha}}{1-\bar{\alpha}}$
S = Total Surface Area of Room
$\bar{\alpha}$ **= Average Absorption Coefficient of Room**

Reverberation Improvement to a Room

To reduce the reverberation time in a room additional absorption must be added to the room surfaces. We have detailed below porous and panel absorbers which will be easy to locate on wall and ceiling within teaching areas. We have described their particular benefit for absorbing sound.

Porous Absorber Panels

These consist of such materials as fibreboard, mineral wools, insulation blankets etc and all have one thing in common, (their network of interlocking pores). Typical characteristics are shown in *Illustration* 24, sound absorption is far more efficient at high frequency than low frequency bands.

ABSORPTION CHARACTERISTICS FOR POROUS MATERIALS

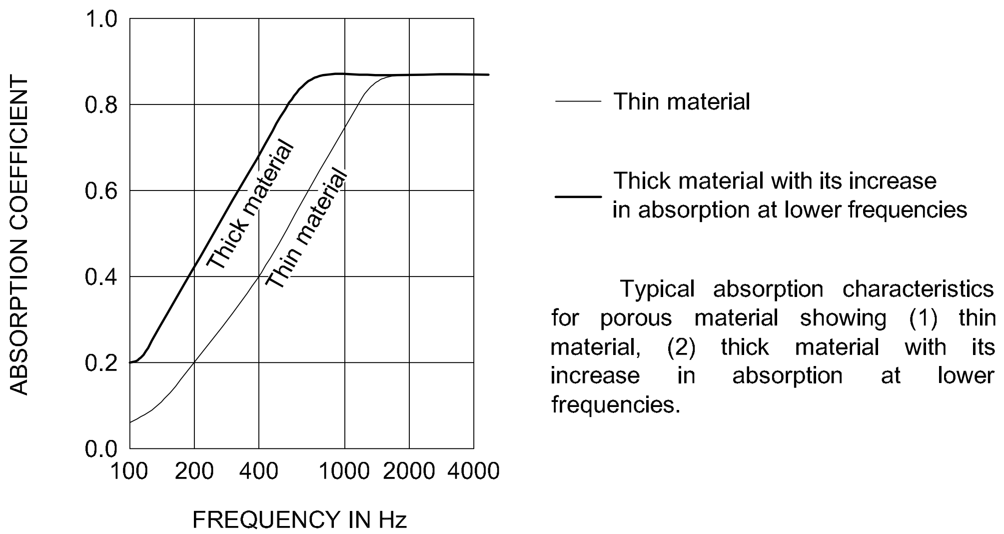


ILLUSTRATION 24

Membrane or Panel Absorbers

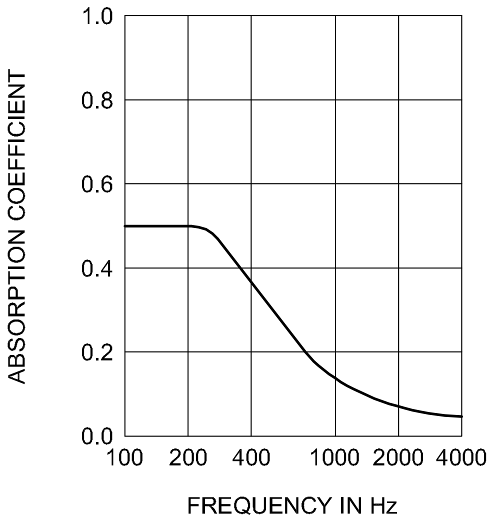
This kind of absorber is useful due to its good absorption characteristics in the low frequency range. Its absorption is highly dependent upon frequency and is normally in the range 63 to 500Hz – see *Illustration* 24.

The approximate resonant frequency can be calculated from:

Where **f = frequency Hz**
m = mass per unit area (KG)m²
d = the depth of air space in m

$$f = \frac{60}{\sqrt{(md)}}$$

TYPICAL ABSORPTION CHARACTERISTICS OF A MEMBRANE ABSORBER



Typical absorption characteristics for a membrane absorber showing increase efficiency at lower frequencies.

ILLUSTRATION 25

Sound Reduction Index

The sound reduction index (R) of a wall, panel, skin of metal or glass, slats of glass fibre, rock wool, polymer mass barrier, or sound board is a measure of its ability to prevent sound energy passing through it.

Sound reduction index is defined as:- $\mathbf{R = 10LOG\left(\frac{1}{\mathit{T}}\right)\mathit{db}}$

Where (T) is the transmission coefficient

The transmission coefficient is defined as the ratio of the acoustic energy transmitted through the obstruction to the total energy incident upon it (see *Illustration* 25*)*

ENERGY FLOW IN AN ACOUSTICALLY EXCITED PANEL

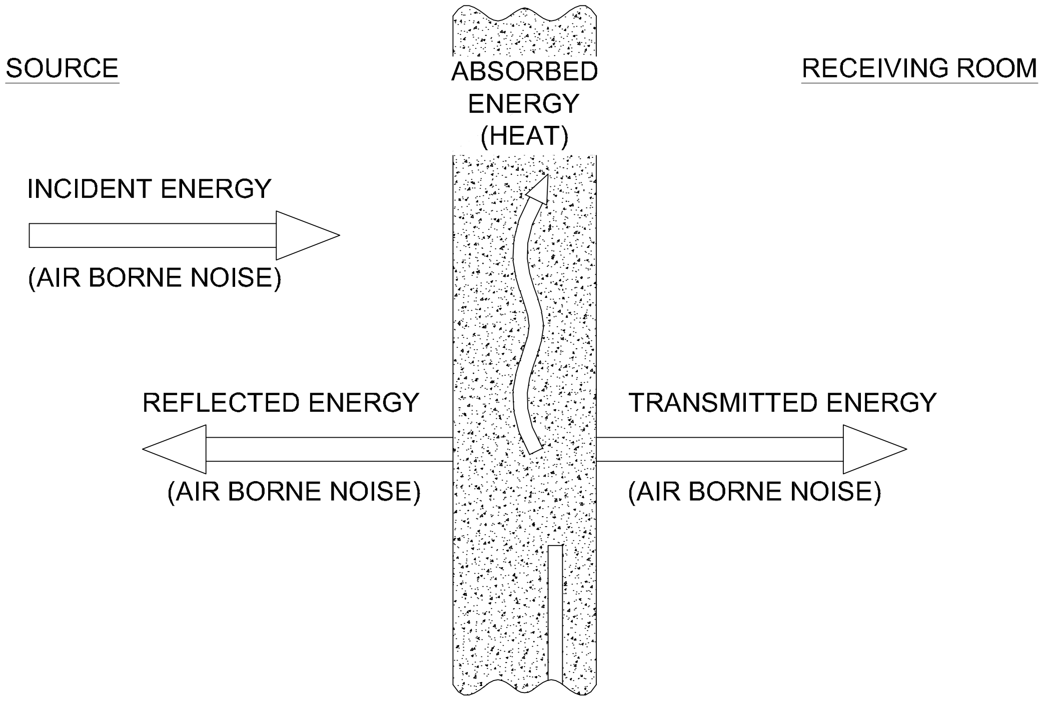


ILLUSTRATION 26

The effect of mass and frequency show the qualities of a panel which determines its resistance to movement. One of the most important is mass.

If we try to vibrate the panel rapidly, its resistance to movement is governed by the inertia force which acts in the opposite direction to the applied force. The higher the mass the greater the inertia force resisting movement, hence the higher the sound reduction index.

The frequency of the applied force is also important, if we try to move the mass more times per second, the velocity of movement must be correspondingly higher, as must be the resulting inertia force, hence the sound reduction index will be higher.

The combined effects of mass and frequency can be analysed theoretically for a single skin panel, to what is referred to as “the mass law of sound insulation”.
R = 20LogMf – 43 db

Where (M) is the Weight in Kg/M^2
(f) is the frequency Hz

Note

If the sound plane waves are parallel to the surface, (0^o) incident, and if the source side is a semi reverberant area, the sound will impinge on the panel from all directions.

Under these conditions the sound reduction index will be less than that predicted by the normal mass law.

Provided the range of angles of incidence is less that 80^o on either side of the normal to the panel, we use the field incidence law, which has the same form as the normal incidence mass law, but is approximately 6dB lower.

If there is sound likely to be incident on the panel, at or near, grazing incidence 80^o/90^o, which might occur, ie if the source side is a highly reverberant room, neither the field incidence law nor the normal incidence law, will apply. The level at any value of (M.f) being lower, and also doubling mass or frequency gives only 5dB increase in sound reduction index, compared with 6dB increase predicted by the mass law.

The three curves in *Illustration 27* show the different incidences.

SOUND REDUCTION INDEX FOR DIFFERENT INCIDENCES

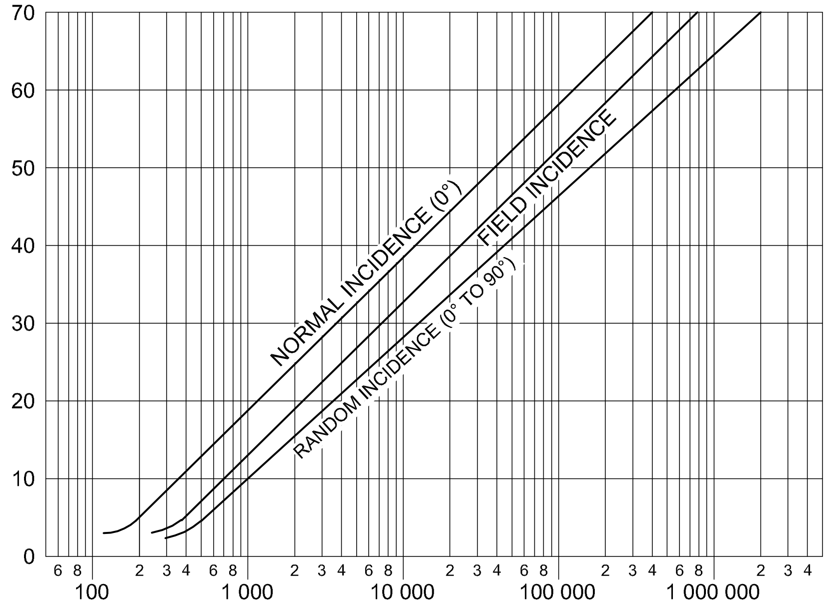


ILLUSTRATION 27

Mechanical Ventilation Plant Casework Sound Reduction

Ventilation plant is constructed normally from a frame with double skin thermally insulated panels of varying thicknesses from 18mm to 50mm.

The type of insulation used is normally fibreglass or rock wool in high density 40<80kg/m^2 density rigid slabs. Also used by some manufacturers are closed polyurethane foam slabs.

While these caseworks will have some sound power loss and decay the plant noise emissions, are generally not designed to reduce the noise of plant to the standards required in building bulletin 93, if located in the room they serve!

To improve noise emissions from mechanical ventilation internal and external mounted plant the following factors in sound barrier and absorption decay must be considered.

Plant Component Sound Power Loss

Sound when passed through absorptive material or restricted none laminar spaces will decay sound. Items found in mechanical ventilation plant that will decay sound passing through them which are called component loss, are as follows:

1. Pre-filters which are normally G3/4 grade filtration efficiency have virtually no absorption decay.

2. Secondary filters which are normally bag type with F6/7 filtration grade will have a slightly higher absorption than the pre-filters, which will be mid to high frequency decay.

3. Coils for heating and cooling which have mechanical bonded fins will have a decay. The magnitude of the decay will be the numbers of rows or the depth of the fins.

   Another factor relating to coil sound decay is the fin spacing, the closer the spacing the higher the sound decay.

SOUND DAMPING IN PLATE HEAT EXCHANGERS

Sound Damping in dB per type of Plate Heat Exchanger
Air Velocity 4.25m/s

Type	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
0	1	2	2	3	3	3	4	4
03	1	2	2	3	3	3	4	4
04	1	2	2	2	3	3	3	3
05	1	2	2	4	4	5	5	5
06	1	4	4	6	7	8	9	9
07	2	5	5	7	8	10	11	11
08	2	5	5	7	9	10	12	12
10	3	6	6	8	10	11	13	13
12	3	6	6	9	10	11	13	13
14	3	6	6	9	10	11	13	13
16	3	7	7	10	11	11	12	15

Recuperator Size

Type	Width	Height	Diagonal
02	200	200	290
03	300	300	430
04	400	400	571
05	500	500	712
06	600	600	854
07	700	700	995
08	800	800	1137
10	1000	1000	1425
12	1200	1200	1708
14	1400	1400	1991
16	1600	1600	2274

The mechanical ventilation plant manufacturer should subtract the component sound power loss's from the fan sound power to give the resultant sound power leaving the plant inlets and outlets.

Inbuilt Mechanical Ventilation Plant Attenuator

Sound passive attenuators or silencers as they are sometimes referred to, attenuate by absorbing sound energy as it passes between two airways with acoustic absorbent material.

Acoustic absorbent material used in commercial attenuators operating in normal temperature conditions will be universal fibreglass slab 60/100kg/m^2 density or rock wool of a similar density.

Note that manufacturers sound insulation materials provide different grades of products, which will have different absorption coefficients at different octave band frequencies. The selection of the acoustic absorbent material will determine the absorption coefficient of the splitter attenuator.

Acoustic absorbent splitters will be constructed from a holding frame which will hold the attenuating material in a stable manner. The acoustic absorbent material will have a material protective face called Lantor Scrim Facing. This facing will prevent particle migration under normal velocities 5.0 m/sec.

If splitter face velocities exceed 5.0 m/sec, it is good practice to use expanded metal with stamped perforations to protect the material face, this can also enhance the insertion loss on certain frequencies.

Take note that most sound absorbent materials if subject to moisture will deteriorate. This deterioration will increase as the wet material dries, leaving the slab in an unstable condition.

Therefore we recommend that on kitchen extract units or fresh air intakes in very exposed areas the use of melinex film is wrapped around the acoustic absorbent slab to prevent any moisture ingress.

Melinex film is manufactured in different thicknesses and this thickness is given in microns. All melinex film products will have a performance reduction on the attenuator, and this will depend on its thickness.

Thickness of melinex generally is selected for its resistance to tear. So often melinex is used with thickness's of 125 micron and above, but there are some melinex films which have a thickness of 19<35 microns which have a high tear resistance.

The performance of an attenuator over a given length is not only dependent on the absorption co-efficient of the product used but a function of its splitter depth, and the distance between splitters.

Distance between splitters will have a big effect on the final sound attenuation. This is because the sound waves are carried with airflow and in its direction; sound waves will be at different wave lengths (ie long waves will have low frequency and shortwaves will be high frequency).

For this reason long wave lengths will have less insertions than short wave lengths over the same distance. This is why it is called insertion loss, for the sound decay of a sound attenuator with a given length, splitter distance or airway gap, splitter depth or thickness of absorbent acoustic material.

Insertion loss is given by the attenuator manufacturer over an octave band range of 63Hz < 8000Hz which is the full octave band range.

Fitting the attenuator onto the mechanical plant inlets or outlets is considered good practice, this will prevent the possibility of "Flanking" the attenuator, also the low velocity of air passing through the plant gives a much larger cross sectional area to position the sound attenuator splitters.

With this increased free area, which will be much less in the ductwork, it is possible to reduce the airways to enhance the insertion loss of the attenuator.

TYPICAL DUCT ATTENUATOR

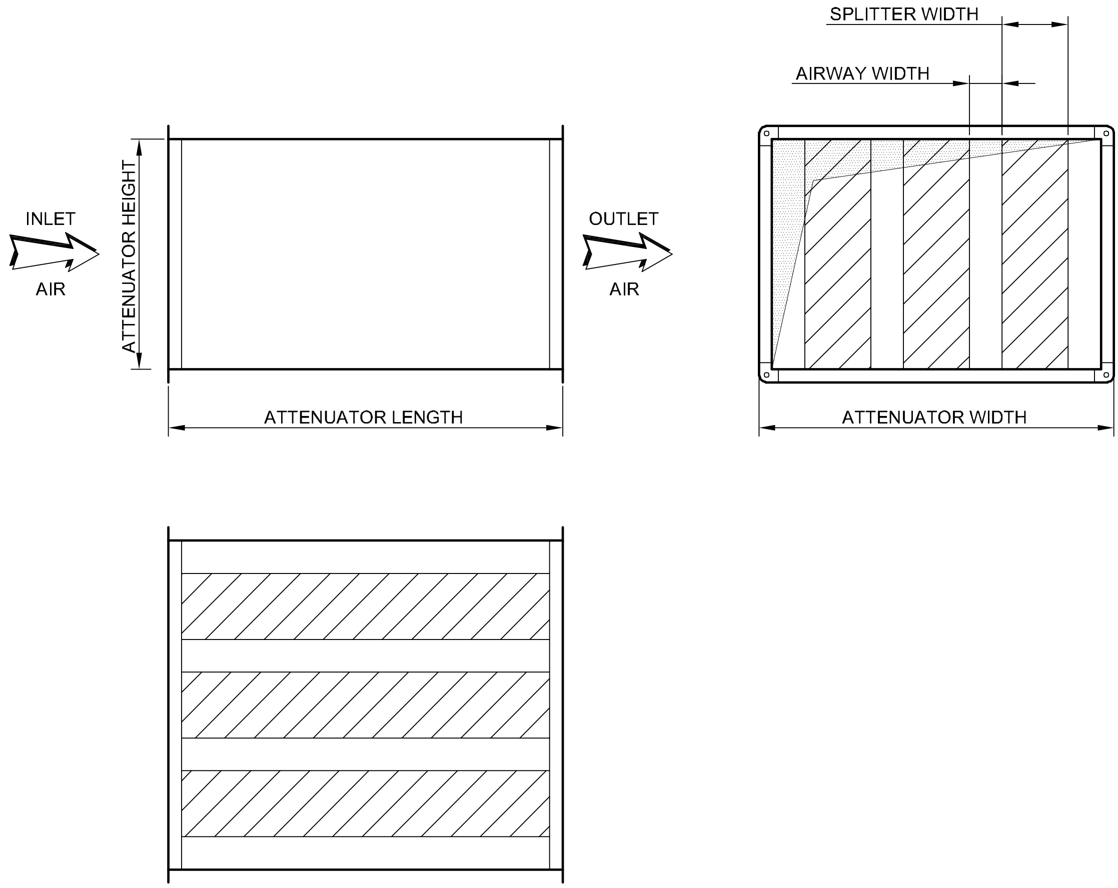
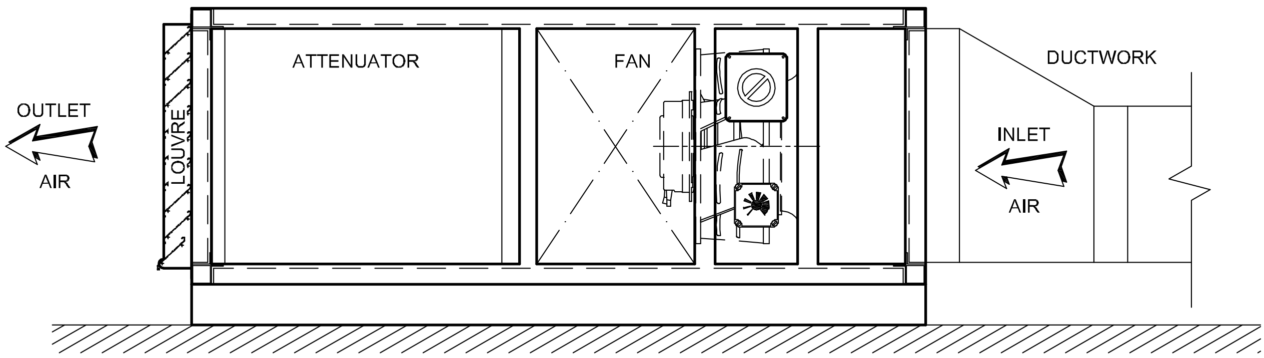


ILLUSTRATION 28

TYPICAL MECHANICAL VENTILATION PLANT IN BUILT ATTENUATOR



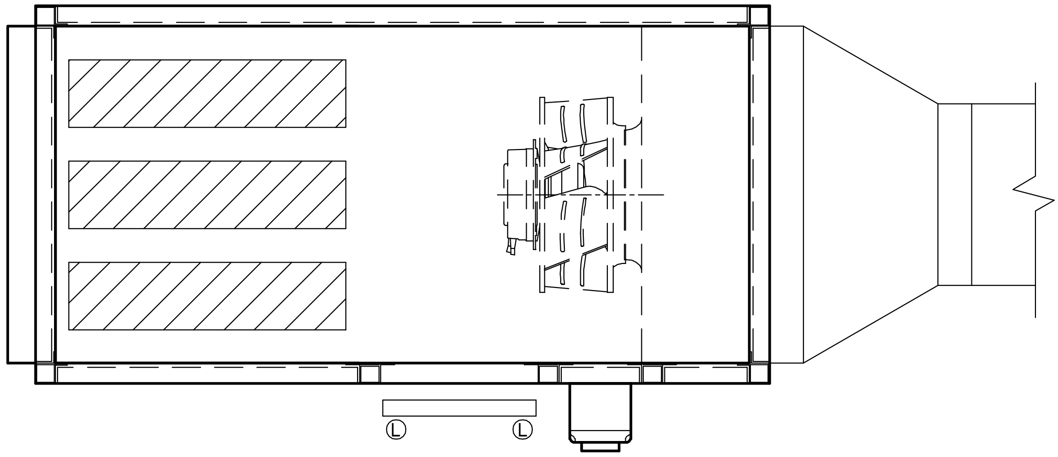


ILLUSTRATION 29

Correct Attenuator Positioning

To avoid “Flanking” of the attenuator, the attenuators should be inbuilt into the mechanical ventilation plant or bolted directly on to it below as in *Illustration 30.*

CORRECT ATTENUATION POSITION

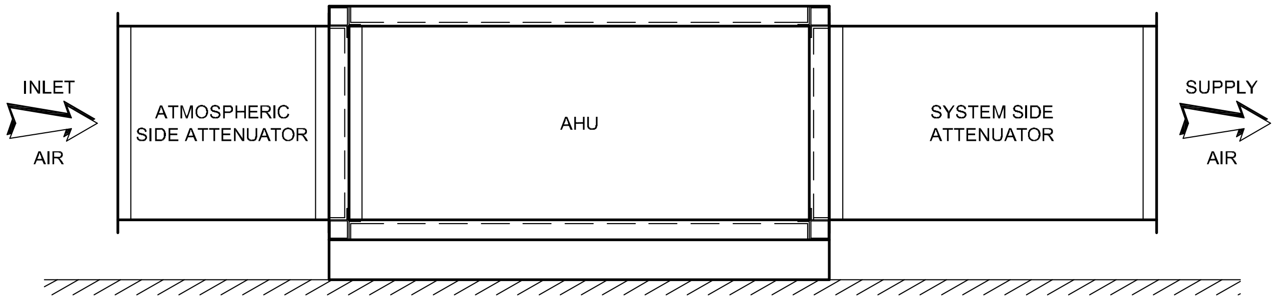


Illustration 30

Principles of Passive Attenuation

The common practice of mechanical ventilation plant attenuation which uses absorption as its principle is passive attenuation.

There is another form of attenuation which is active attenuation which uses out of phase wave formation to destroy the wave requiring decay. This principle is commonly used for narrow frequency bands, and is not suited for white noise.

As previously described the effect of passive absorption attenuators is a function of:-

a. Frequency Band
b. Absorption coefficient of attenuator material
c. Splitter depth
d. Airway width
e. Airway velocity
f. Attenuator length

Attenuator Insertion Loss

As sound waves pass through the attenuator splitters, the absorbent material held in the splitter will absorb sound energy, and convert it to heat energy.

A sound wave will have the following parameters, wave length, depending on the frequency. The wave lengths are longitudinal and depending on the frequency will complete a cycle which will repeat the same pattern over again, see *Illustration 31*.

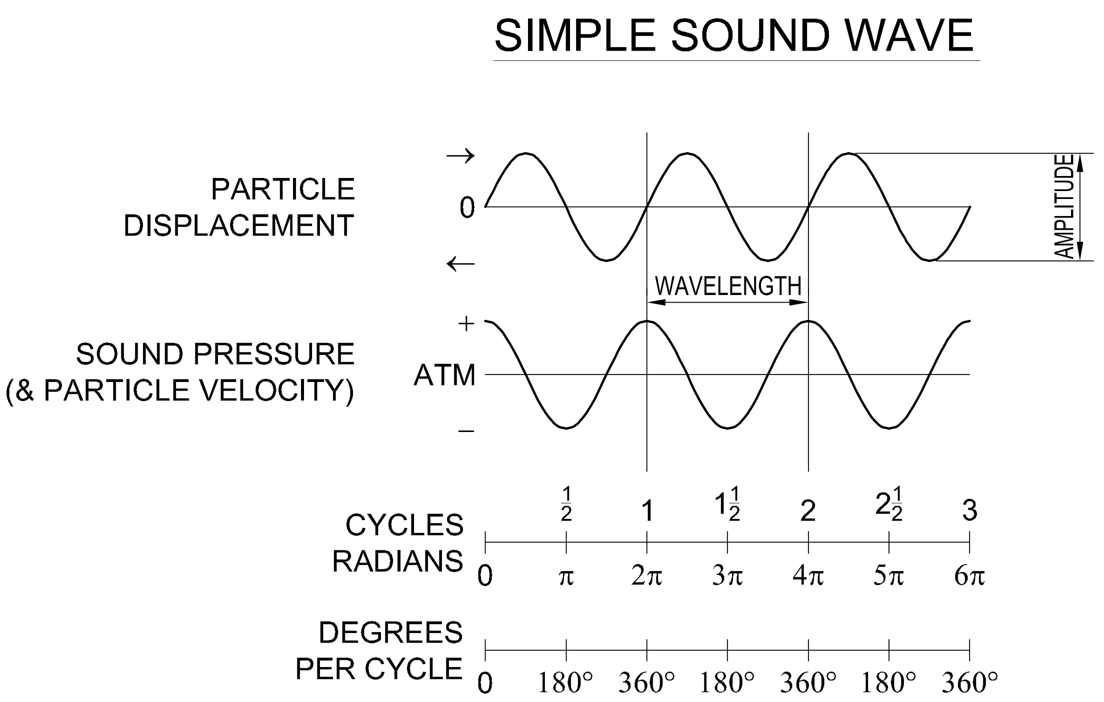


Illustration 31

The frequency influences the wavelength, with the wavelength being one complete cycle. As shown in *Illustration 32* low frequency sound waves are long and high frequency waves are short.

Therefore when passed through a splitter attenuator long waves will have significantly less number of insertions into the absorbent material than short waves. This can be seen with the example of different splitter thicknesses and airway widths.

When the sound wave is decayed after insertion into the attenuating absorbent material, this reduces the amplitude, hence the expression "insertion loss". The amount of amplitude reduction is reflected in the material used.

WAVE LENGTH / FREQUENCY CHART

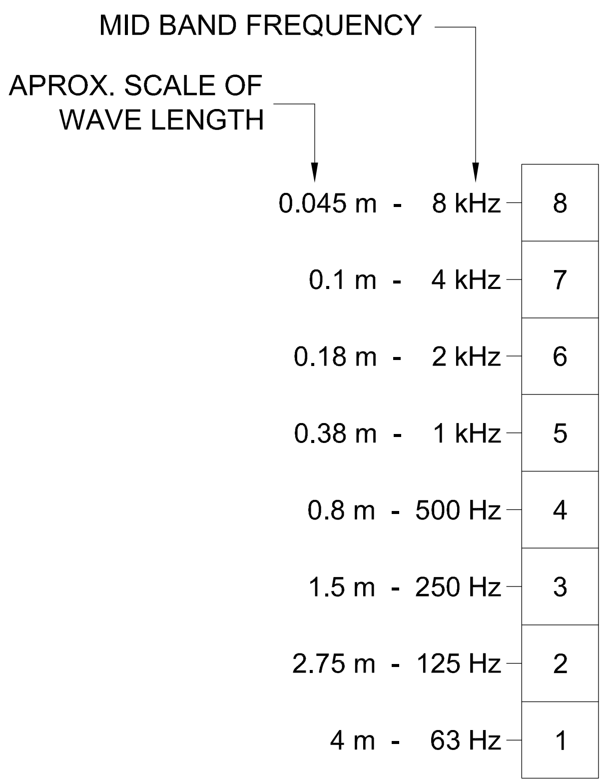


Illustration 32

Once the insertion loss for a cycle on each octave band is determined, based on an airway width size and splitter depth, then the length of the attenuator determines its insertion loss over the full octave bands 1<8 or 63HZ < 8KHZ.

Below are examples of laboratory tested attenuators using 50mm, 75mm and 100mm deep splitters with faced universal slab 60Kg/m^3 density with varying airway widths from 50mm to 125mm, and varying lengths from 600mm long to 2400mm long.

50mm Splitter Depth

Air Passage mm	Length Metres	63	125	250	500	1000	2000	4000	8000
50mm	600	4	5	11	21	35	28	30	24
	900	5	6	12	33	50	41	36	19
	1200	5	6	15	45	55	54	41	27
	1500	7	8	17	53	55	55	50	36
	1800	12	13	31	55	55	55	55	43
	2100	12	13	40	55	55	55	55	50
	2400	13	14	44	55	55	55	55	55
75mm	600	2	2	7	18	29	26	18	16
	900	3	3	10	26	44	36	26	29
	1200	3	3	12	33	55	50	34	31
	1500	4	5	14	38	55	55	45	37
	1800	5	6	16	50	55	55	53	45
	2100	5	6	18	55	55	55	55	50
	2400	4	5	25	55	55	55	55	55
100mm	600	2	2	4	19	33	25	19	17
	900	2	3	6	26	42	36	26	298
	1200	4	5	10	33	54	46	34	31
	1500	5	6	10	37	53	53	35	32
	1800	5	6	10	44	55	55	40	34
	2100	7	8	12	55	55	55	53	48
	2400	8	9	15	55	55	55	55	52
125mm	600	1	1	3	15	24	23	14	12
	900	1	1	5	22	35	31	20	18
	1200	1	2	7	29	47	40	27	26
	1500	3	4	6	32	47	45	30	29
	1800	4	5	8	36	52	52	33	31
	2100	5	6	9	49	55	51	41	37
	2400	5	6	11	55	55	55	49	42

75mm Splitter Depth

Air Passage mm	Length Metres	63	125	250	500	1000	2000	4000	8000
50mm	600	5	9	15	26	38	32	34	27
	900	7	12	18	40	55	48	48	45
	1200	9	14	24	54	55	55	55	55
	1500	10	16	30	55	55	55	55	55
	1800	12	22	46	55	55	55	55	55
	2100	14	24	55	55	55	55	55	55
	2400	15	27	55	55	55	55	55	55
75mm	600	4	6	7	20	31	26	21	17
	900	5	8	12	31	45	40	33	14
	1200	6	9	18	40	55	55	43	20
	1500	8	11	19	48	55	55	50	36
	1800	9	13	21	50	55	55	55	42
	2100	11	15	32	55	55	55	55	49
	2400	12	16	43	55	55	55	55	55
100mm	600	3	5	7	20	31	26	21	20
	900	4	7	12	28	39	38	38	27
	1200	5	9	17	36	49	49	38	25
	1500	7	11	18	39	53	53	40	29
	1800	8	12	21	47	55	55	47	34
	2100	10	14	23	55	55	55	55	40
	2400	11	16	28	55	55	55	55	45
125mm	600	2	3	5	17	23	23	15	11
	900	3	4	9	24	34	31	22	14
	1200	4	6	13	32	45	41	29	17
	1500	6	8	14	36	47	46	32	20
	1800	7	12	18	40	52	52	36	21
	2100	8	12	20	50	55	51	44	24
	2400	9	13	23	55	55	55	52	28

<u>100mm Splitter Depth</u>

Air Passage mm	Length Metres	63	125	250	500	1000	2000	4000	8000
50mm	600	6	12	22	31	40	40	40	30
	900	8	16	27	45	55	55	55	50
	1200	10	20	36	55	55	55	55	55
	1500	13	24	42	55	55	55	55	55
	1800	15	30	51	55	55	55	55	55
	2100	17	34	55	55	55	55	55	55
	2400	19	38	55	55	55	55	55	55
75mm	600	5	8	11	24	31	32	24	20
	900	6	11	19	34	45	45	39	28
	1200	7	14	26	46	55	55	52	38
	1500	9	17	30	48	55	55	55	42
	1800	10	20	34	50	55	55	55	46
	2100	12	23	40	55	55	55	55	55
	2400	13	24	45	55	55	55	55	55
100mm	600	4	7	11	21	31	29	21	20
	900	5	9	16	30	39	39	31	26
	1200	6	12	23	40	51	51	41	29
	1500	8	15	26	43	53	53	45	32
	1800	9	17	30	47	55	55	49	36
	2100	11	20	35	55	55	55	55	43
	2400	12	23	40	55	55	55	55	47
125mm	600	3	5	7	17	23	23	16	12
	900	4	7	13	25	32	32	23	15
	1200	5	9	19	33	42	42	30	18
	1500	7	12	22	38	47	47	34	20
	1800	8	14	26	43	52	52	39	23
	2100	9	17	30	50	55	51	46	28
	2400	10	19	34	55	55	55	52	30

OVERVIEW OF FANS USED IN MECHANICAL PLANT

With the growing requirement for energy reduction, driven by continual changes to the building regulations, the specific fan powers (w/m/s) have been significantly reduced.

Since the drive to reduce power consumption in recent years, the commercial fan industry has seen many changes. The age old argument over “backward” versus “forward” curved fans has been decided.

The argument focused for the backward curved fan on, higher efficiency hence less energy consumed plus a non-overloading impellor, whereas the forward curved fan which runs at lower peripheral speeds which generate less noise.

Since the argument for the forward curved impellor is lost, then the issue of noise generation has been brought into significance.

A further change to commercial fans in recent years has seen the increase in the use of plenum type plug fans. These have no casework volute, with the impellor directing the airflow as indicated below.

Typical Plenum/Plug Fan Arrangement

Plenum/Plug Fan Sound Generation

Sound generated by Plug Fans (Backward Curved) impellor tend to have sound level predominance from 250Hz to 2000Hz, and this octave band area gives high relationship to dbA levels due to little ‘A’ scale weighting influence.

This challenge has given rise to developing both attenuators and ventilation plant casework to offer higher insertion loss and sound reduction loss to make a difference to overall noise emissions.

In the case of increasing mid band insertion loss the reduction in airway width will offer a solution together with the selection of absorbent splitter material and the width.

With regard to mechanical plant casework sound reduction, very little is known about this subject, in fact many manufacturers offer casework sound reduction values, but do not state how they were achieved.

Ventilation Plant Casework Design

As described in previous sections, breakout noise from mechanical plant has a damaging impact on the surrounding or room it is located in. We have offered in this section why it is important the research and testing Air Handlers have engaged in will offer the designer more confidence in selecting mechanical ventilation plant to match the requirements laid down in Building Bulletin 93.

Sound Reduction Index Testing Program

Under the supervision and expertise of David Pinchbeck (Air Handlers Founder & Owner), a team was formed which included Salford University Acoustics Testing Laboratory, Dr Bill Whitfield of Noise.co.uk and members of Air Handlers Technical Team headed by Filipe Beirao and Donald Cole.

Various composite materials were researched, and work with the manufacturers proved successful, in improving both the sound reduction index and absorption properties of these materials.

Testing was carried out initially in Air Handlers Northern Ltd Test Facility. These tests were based on an actual Heat Reclaim Unit with known fan sound power level, and attenuators sized to reduce the inlet/outlet noise levels to 35 dBA.

With a known test reverberation time, this standard HRU would act as a datum for panel and frame performance. Results of this testing programme showed certain panel compositions decayed better on certain frequencies, also the importance of acoustic insulation to the frame.

Once a set of acoustic panel composites were configured, a test program was agreed with Salford University Acoustics Laboratory. This test programme was carried out over two periods, with modifications being carried out to the panel composite giving higher performance; Salford University Acoustic Laboratory is a UKAS accredited Test Facility.

All tests were carried out in accordance with BS EN ISO 10140-2:2010 as outlined in report 1429 and 2060 illustrated below.

Test Samples

Description of Test Samples

The following panel enclosure systems built and tested in the 3600mm x 2400mm aperture in the transmission suite. A hollow aluminium pentapost frame was installed within the aperture and fixed in place with clamping bolts. The frame contained a recess nominally 45mm deep with a 6mm foam seal fitted to the closing edge of the open frame section.

Source Room

Receiving Room

Fitting Frame and Panels
into Test Aperture

Checking Panel Thickness

Recording Test Results
in the Control Room

Microphone Positions
in Receiving Room

Sound Source Positions

Plywood Diffusers
Receiving Room

SALFORD UNIVERSITY TEST REPORT 1429

The following enclosure panels, consisting of a layered core (described from receiver to source room) contained within two 0.7mm steel cassettes of a number of thicknesses; were then installed into the open sections and fixed in place using toggles located on the frame on the receiving room side of the system. All units reported are nominal unless otherwise stated.

Test A

Test Reference:	1429-1197
Sample Reference:	"PB50"
Sample Description:	45mm thick panel with a core consisting of four layers. The mass per unit area of one panel was measured to be equal to 34.2kg/m$^{2.}$

Test B

Test Reference:	1429-1199
Sample Reference:	"ASPB50"
Sample Description:	45mm thick panel with a core consisting of three layers. The mass per unit area of one panel was measured to be equal to 35.2kg/m^2.

Test C

Test Reference:	1429-1200
Sample Reference:	"ASTSAS50"
Sample Description:	45mm thick panel with a core consisting of three layers. The mass per unit area of one panel was measured to be equal to 45.6kg/m^2.

Test D

Test Reference:	1429-1201
Sample Reference:	"ASTSFG50"
Sample Description:	45mm thick panel with a core consisting of three layers. The mass per unit area of one panel was measured to be equal to 38.1 kg/m^2.

Test E

Test Reference:	1429-1203
Sample Reference:	"ASTSPB50"
Sample Description:	Modifications were made by fixing Acoustic Barrier Insulation to the source room side of the frame.

Test F

Test Reference: 1429-1206

Sample Reference: "PBFG50 with 1.2mm plastisol"

Sample Description: 45mm thick panel with a core consisting of three layers. The steel cassette on the source side of the system under test was replaced by a 1.2mm plastisol cassette. The mass per unit area of one panel was measured to be equal to 29.7kg/m^2.

Test G

Test Reference: 1429-1208

Sample Reference: "PB25"

Sample Description: 25mm thick panel with a core consisting of two layers. Profile sections were used to provide secure fitting between the panel and the frame toggles. The mass per unit area of one panel was measured to be equal to 22.2kg/m^2.

FRAME DETAILS

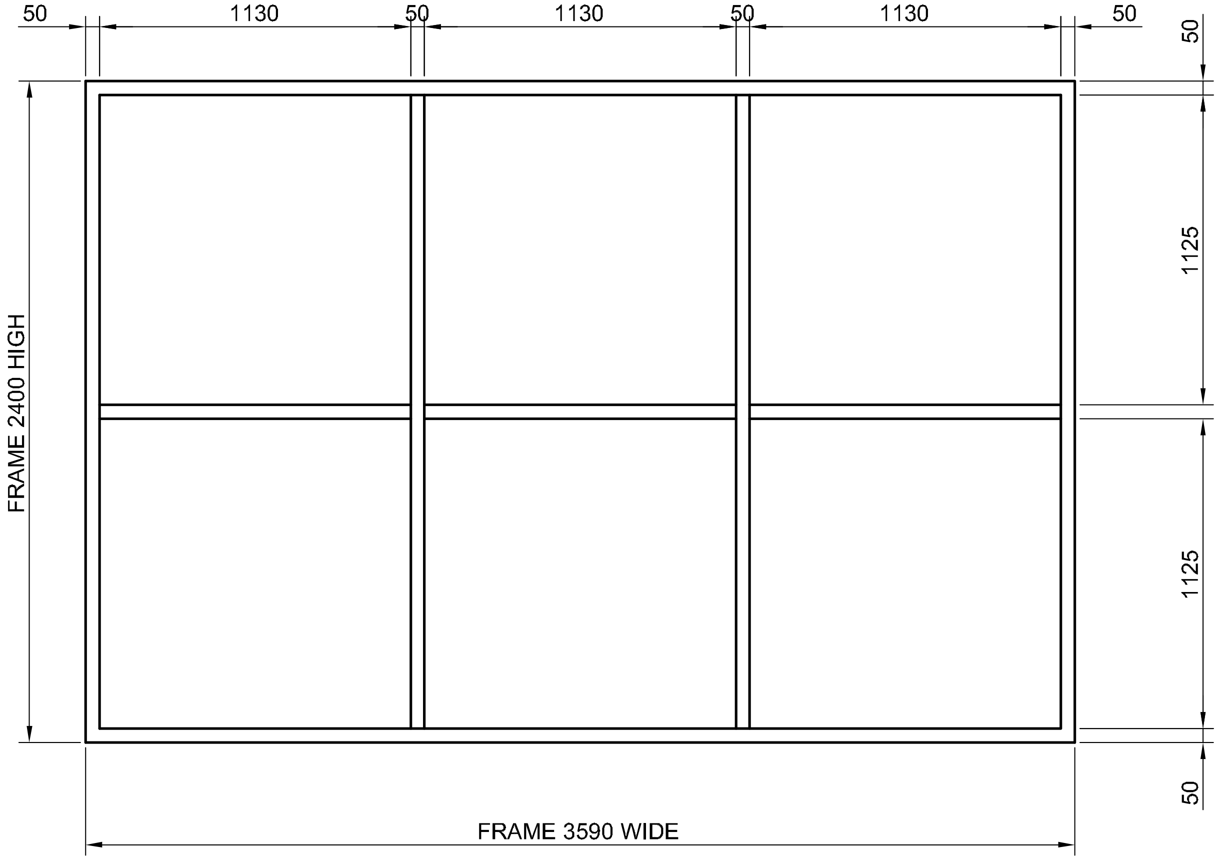


Illustration 33

50mm PANEL FIXING METHODS

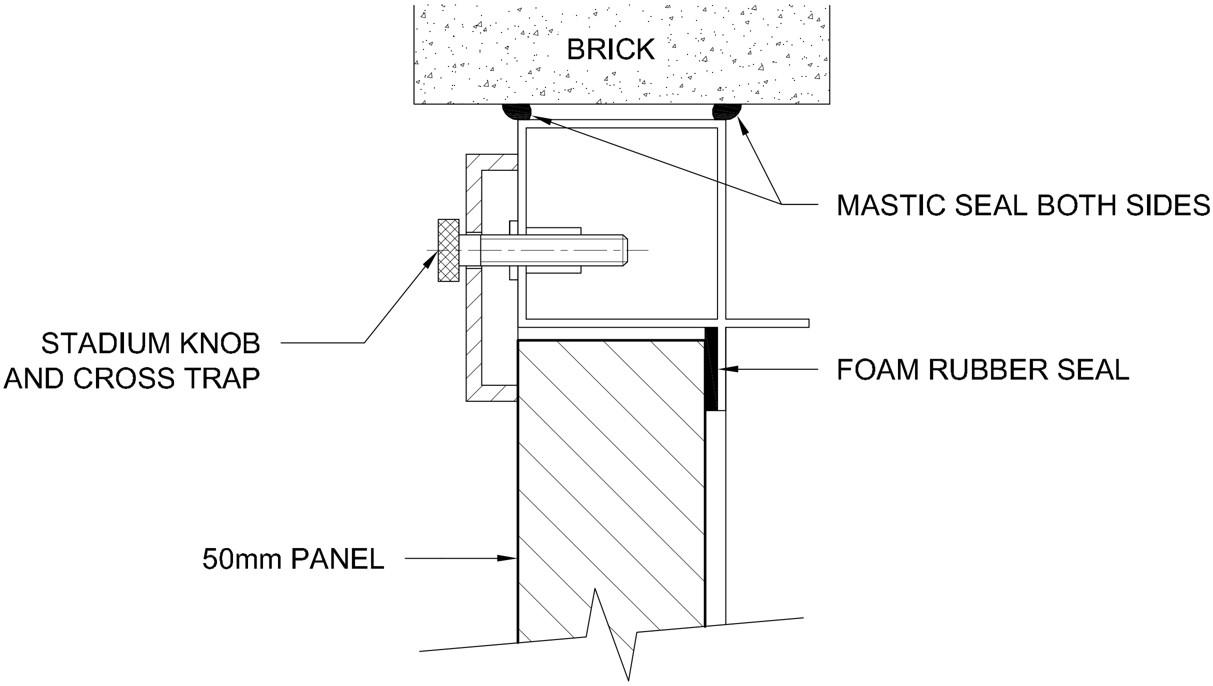


Illustration 34

25mm PANEL FIXING METHODS

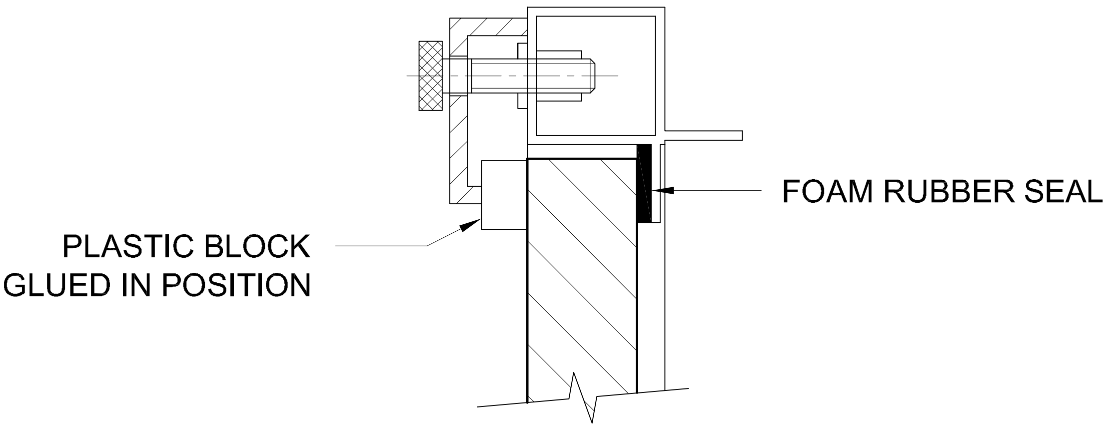


Illustration 35

LIFTING HANDLES

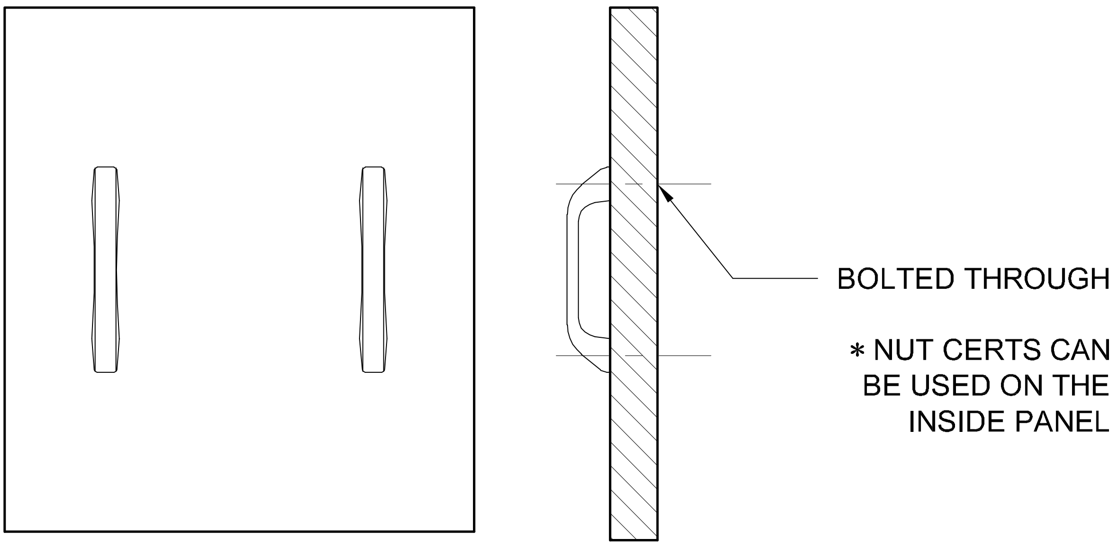


Illustration 36

FIXING METHODS FOR PENTAPOST FRAME TO BRICK WALL CHAMBER

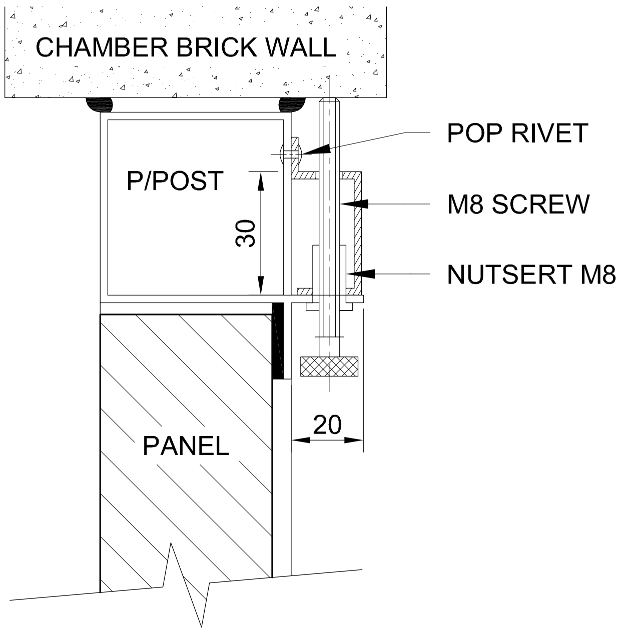


Illustration 37

DESCRIPTION OF TEST PROCEDURE

The test procedure adopted follows that detailed in BS EN ISO 10140: Part: 2010, "Acoustics – Laboratory measurements of sound insulation of building elements; Part 2: Measurement of airborne sound insulation".

The measurements are performed in the large transmission suite at the University of Salford. The suite comprises two structurally isolated reverberant rooms with a test opening between them in which the test specimen is inserted. The vertical sides of the test aperture and the base are made from dense brick, whilst the soffit is made from reinforced concrete. Both rooms have been designed with hard surfaces and non-parallel walls. The smaller source room has 6 plywood diffusers and the larger receiving room has 11 plywood diffusers, to increase the diffusivity of the sound field in these areas.

The test involves producing a known sound field in the source room and measuring the resultant sound level difference between the source room and the receiving room with the specimen installed in the test aperture. This level difference is then corrected so as to take into account the equivalent absorption area of the receiving room.

The Sound Reduction Index, R (dB), is defined in BS EN SIO 10140 – Part 2: 2010 as:

$$R = L_1 - L_2 + 10 \log_{10} \frac{S}{A} \qquad \textbf{(1)}$$

Where:
L_1 is the average sound pressure level in the source room (dB)
L_2 is the average sound pressure level in the receiving room (dB)
S is the area of the rest specimen (m^2)
A is the equivalent absorption area of the receiving room (m^2)

Generation of Sound Field in the Source Room

Wide band, random noise from the generator in the real time analyser was amplified and reproduced in the source room using alternately one of two fixed loudspeaker systems, (La, Lb and Lc). Omni-directional loudspeakers were used. The output of the generator was set with the intention that the sound pressure level in the receiving room was at least 15dB higher than the background level in any frequency band. The loudspeakers were positioned in the corners of the room and at such a distance from the test specimen that the direct radiation upon it was not dominant.

Frequency Range of Measurements

The sound pressure levels were measured using one-third octave band filters. Measurements covered all the one-third octave bands having centre frequencies in the range from 50Hz to 5000Hz. At the request of the client, measurements were also taken at the one-third octave band frequencies 6.3 kHz, 8 kHz and 10 kHz. The sound reduction indices at these additional frequencies are presented in Appendix A.

Measurement of Sound Pressure Levels

Sound pressure levels were measured simultaneously in the source and receiving rooms using loudspeaker **La** as the sound source. Measurements were recorded at a minimum of 5 fixed microphone positions in each room, using an averaging time of 32 seconds and the average level in each room was calculated on an energy basis in each one-third octave frequency band. The procedure was then repeated with loudspeaker **Lb** and **Lc** as the sound source.

The overall average level difference in each frequency band was then calculated as the arithmetic average of the two sets of results.

For each set of microphone/loudspeaker positions, the distances separating microphones from other microphones, room boundaries and diffusers, were greater than 0.7m and the distances separating microphones from the sound source and the test specimen were greater than 1m.

Measurement and Evaluation of the Equivalent Absorption Areas

The correction term of equation (1) containing the equivalent absorption area, *A*, was evaluated from the reverberation time and calculated using Sabine's formula:

$$A = \frac{\mathbf{0.16\ V}}{\mathbf{T}} \qquad \textbf{(2)}$$

Where:
V is the volume of the receiving room (m^3)
T is the reverberation time (s)

The reverberation time of the receiving room was measured using a decay technique. The decays were produced by exciting the room with wide band random noise and stopping the excitation once the room became saturated. The resulting decaying sound field was monitored at 6 fixed microphone positions using a one-third octave band real time analyser. The sound spectrum was sampled at 32 millisecond intervals and stored in memory. Five decays were measured at each microphone position and averaged. The time taken for the sound to decay by 20dB was measured and multiplied by three to give the reverberation time. The measurements were repeated using an alternative sound source. The results from each set of position were averaged (ie 60 reverberation decays at each frequency).

<u>EQUIPMENT</u>

	Departmental Record No
Norwegian Electronics 1/3 octave band real time analyser type 840 with in-built random noise generator	RTA2
Quad 510 power amplifier	PA7
2 of omni-directional broadband loudspeakers (source room)	LS10-LS11
2 of broadband loudspeakers (receiving room)	LS3-LS4
3 of Bruel & Kjaer random incidence condenser microphones type 4166 in the source room	M2-M4
3 of G.R.A.S. random incidence condenser microphones type 40AP in the source room	M21,M22,M25
5 of Brunel & Kjaer random incidence condenser microphone type 4166 in the receiving room	M7-M9 M18, M19
1 if G.R.A.S. random incidence condenser microphones type 40AP in the receiving room	M20

2 of Norsonic Multiplexers type 834A	MP1-MP2
HP Brio Pentium personal computer and related peripheral equipment (printer, plotter, monitor etc.)	COM6
Yamaha GQ1031BII graphic equalizer	GEQ1

RESULTS

The sound reduction indices at one-third octave band intervals, *R,* are given in the tables overleaf.

Source room volume: $136m^3$

Receiving room volume: $220m^3$

Sample sizes: 2400mm x 3600mm

TEST RESULTS
TO BS EN ISO 10140-2(2010)

Certified by: Salford University Acoustic Testing Laboratory

Also given in the attached tables and computed from the one-third octave band sound reduction indices, is the weighted sound reduction index, R_w, calculated according to ISO 717/1-1996. This evaluation is based on laboratory measurement results obtained by an engineering method.

APPENDIX A

At the client's request, the sound reduction index was measured at the additional frequencies of 6.3 kHz, 8 kHz and 10 kHz, the results of which are presented below.

Test Reference	Sound Reduction Index, *R* [dB]		
	6.3 kHz	8 kHz	10 kHz
1429-1197	39.8[1]	40.1[1]	32.6[2]
1429-1199	40.0[1]	40.2[1]	32.9[2]
1429-1200	34.5	24.3	7.4
1429-1201	39.9	40.1	35.5[1]
1429-1203	48.2[1]	46.9[1]	37.1[2]
1429-1206	48.3[1]	47.4[1]	36.3[2]
1429-1208	46.3[1]	45.4[1]	35.2[2]

[1] Correction for background applied to the result
[2] Background levels too high – minimum value for *R*.

SALFORD UNIVERSITY TEST REPORT 2060

Test H

Test Reference: 2060-1673
Sample Reference: "PB18 Double Skin"
Sample Description: 18mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 18.0mm and 27.5kg/m^2 respectively.

Test I

Test Reference: 2060-1678
Sample Reference: "PB/TS/PB25 Triple Skin"
Sample Description: 25mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45.0mm and 41.0kg/m^2 respectively.

Test J

Test Reference: 2060-1679
Sample Reference: "AS/TS/SBP50 Triple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45.0mm and 53.6kg/m^2 respectively.

Test K

Test Reference: 2060-1680
Sample Reference: "PB/SPB25 Double Skin"
Sample Description: 25mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 25.0mm and 34.2kg/m^2 respectively

Test L

Test Reference: 2060-1681
Sample Reference: "SBP18 Double Skin"
Sample Description: 18mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 18.6mm and 30.6kg/m^2 respectively.

Test M

Test Reference: 2060-1683
Sample Reference: "AS/TS/FG50 Triple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45.0mm and 39.9kg/m^2 respectively.

Test N

Test Reference: 2060-1684
Sample Reference: "AS/TS/PB50 Triple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 49.9kg/m^2 respectively.

Test O

Test Reference: 2060-1685
Sample Reference: "FG/TS/PB50 Triple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 40.0kg/m^2 respectively.

Test P

Test Reference: 2060-1686
Sample Reference: "FG25 Double Skin"
Sample Description: 25mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 21.3kg/m^2 respectively.

Test Q

Test Reference: 2060-1686
Sample Reference: "FG50 Double Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 23.3kg/m^2 respectively.

Test R

Test Reference: 2060-1689
Sample Reference: "AS/QS/PB50 Quadruple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 62.9 kg/m^2 respectively.

Test S

Test Reference: 2060-1675
Sample Reference: "FG18 Double Skin"
Sample Description: 18mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 18.0mm and 20.9kg/m^2 respectively.

Test T

Test Reference: 2060-1676
Sample Reference: "FG/TS/SBP50 Triple Skin"
Sample Description: 50mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 45mm and 43.3kg/m^2 respectively.

Test U

Test Reference: 2060-1674
Sample Reference: "AS25 Double Skin"
Sample Description: 25mm nominally thick panel.
The measured thickness and mass per unit area of one panel was measured to be equal to 30.3mm and 31.3kg/m^2 respectively.

APPENDIX A

At the client's request, the sound reduction index was measured at the additional octave band frequency of 8 kHz, the results of which are presented below.

Test Reference	**Sound Reduction Index, *R* [dB] 8 kHz**
2060-1673	39.6
2060-1674	40.8
2060-1675	39.9
2060-1676	41.5
2060-1678	40.1
2060-1679	39.6
2060-1680	39.6
2060-1681	39.2
2060-1685	40.3
2060-1686	39.7
2060-1687	40.4
2060-1689	43.7

DESCRIPTION OF TEST PROCEDURE

The test procedure adopted follows that detailed in BS EN ISO 10140: Part 2: 2010, "Acoustics – Laboratory measurements of sound insulation of building elements; Part 2: "Measurement of airborne sound insulation".

The measurements are performed in the large transmission suite at the University of Salford. The suite comprises two structurally isolated reverberant rooms with a test opening between them in which the test specimen is inserted. The vertical sides of the test aperture and the base are made from dense brick, whilst the soffit is made from reinforced concrete. Both rooms have been designed with hard surfaces and non-parallel walls. The smaller source room has 6 plywood diffusers and the larger receiving room has 11 plywood diffuses, to increase the diffusivity of the sound field in these areas.

The test involves producing a known sound field in the source room and measuring the resultant sound level difference between the source room and the receiving room with the specimen installed in the test aperture. This level difference is then corrected so as to take into account the equivalent absorption area of the receiving room.

The Sound Reduction Index, R (dB), is defined in BS EN ISO 10140-Part 2:2010 as:

$$R = L_1 - L_2 + 10\log_{10}\frac{S}{A} \qquad (1)$$

Where:

L_1 is the average sound pressure level in the source room (dB)
L_2 is the average sound pressure level in the receiving room (dB)
S is the area of the test specimen (m^2)
A is the equivalent absorption area of the receiving room (m^2)

Generation of Sound Field in the Source Room

Wide band, random noise from the generator in the real time analyser was amplified and reproduced in the source room using alternately one of two fixed loudspeaker systems, (**La, Lb and Lc**). Omni-directional loudspeakers were used. The output of the generator was set with the intention that the sound pressure level in the receiving room was at least 15dB higher than the background level in any frequency band. The loudspeakers were positioned in the corners of the room and at such a distance from the test specimen that the direct radiation upon it was not dominant.

Frequency Range of Measurements

The sound pressure levels were measured using octave band filters. Measurements covered all the octave bands having centre frequencies in the range from 50Hz to 8000Hz.

Measurement of Sound Pressure Levels

Sound pressure levels were measured simultaneously in the source and receiving rooms using loudspeaker **La** as the sound source. Measurements were recorded at 6 fixed microphone positions in each room, using an averaging time of 16 seconds and the average level in each room was calculated on an energy basis in each octave frequency band. The procedure was then repeated with loudspeaker **Lb** and **Lc** as the sound source. The overall average level difference in each frequency band was then calculated as the arithmetic average of the two sets of results.

For each set of microphone/loudspeaker positions, the distances separating microphones from other microphones, room boundaries and diffusers, were greater than 0.7m and the distances separating microphones from the sound source and the test specimen were greater than 1m.

Measurement and Evaluation of the Equivalent Absorption Areas

The correction term of equation (1) containing the equivalent absorption area, *A*, was evaluated from the reverberation time and calculated using Sabine's formula:

$$A = \frac{0.16\,V}{T} \qquad (2)$$

Where:

V is the volume of the receiving room (m^3)
T is the reverberation time (s)

The reverberation time of the receiving room was measured using a decay technique. The decays were produced by exciting the room with wide band random noise and stopping the excitation once the room became saturated. The resulting decaying sound field was monitored at 6 fixed microphone positions using an octave band real time analyser. The sound spectrum was sampled at 32 millisecond intervals and stored in memory. Five decays were measured at each microphone position and averaged. The time taken for the sound to decay by 20dB was measured and multiplied by three to give the reverberation time. The measurements were repeated using an alternative sound source. The results from each set of positions were averaged (ie 60 reverberation decays at each frequency).

<u>EQUIPMENT</u>

	Departmental Record No
Norwegian Electronics 1/3 octave band real time analyser type 840 with in-built random noise generator(used in octave band mode)	RTA2
Quad 510 power amplifier	PA7
2 of omni-directional broadband loudspeakers (source room)	LS10-LS11
2 of broadband loudspeakers (receiving room)	LS3-LS4
3 of Bruel & Kjaer random incidence condenser microphones type 4166 in the source room	M2-M4
3 of G.R.A.S. random incidence condenser microphones type 40AP in the source room	M21,M22,M25
5 of Brunel & Kjaer random incidence condenser microphone type 4166 in the receiving room	M7-M9 M18, M19
1 if G.R.A.S. random incidence condenser microphones type 40AP in the receiving room	M20
2 of Norsonic Multiplexers type 834A	MP1-MP2
Vaisala Digital Hygrometer	HM1
Resonant Sensor Barometer DPI 141	CL6
HP Brio Pentium personal computer and related peripheral equipment (printer, plotter, monitor etc.)	COM6
Yamaha GQ1031BII graphic equalizer	GEQ1

RESULTS

The sound reduction indices at one-third octave band intervals, *R,* are given in the tables overleaf.

Source room volume: $136m^3$

Receiving room volume: $220m^3$

Sample sizes: 2400mm x 3600mm

Environment:[1] Please refer to the table below

Test Reference		Source Room		Receiving Room	
	Ambient Pressure [kPa]±0.1	Temperature [°C]±0.3	Relative Humidity [%]±0.21	Temperature [°C]±0.3	Relative Humidity [%]±0.21
2060-1673	101.2	22.0	31.8	23.1	30.0
2060-1674	101.2	20.7	36.1	20.0	37.3
2060-1675	101.2	21.8	33.4	29.9	38.7
2060-1676	101.2	22.8	32.6	21.2	36.9
2060-1678	101.3	21.1	34.9	21.5	38.5
2060-1679	101.3	20.1	33.3	19.8	36.7
2060-1680	101.3	19.7	36.1	19.7	37.3
2060-1681	101.3	19.3	38.9	19.6	37.0
2060-1683	100.8	21.8	36.3	19.6	39.6
2060-1684	100.8	20.7	38.9	20.4	40.0
2060-1685	100.8	19.8	40.9	20.1	39.9
2060-1686	100.8	19.4	42.4	19.7	40.6
2060-1687	100.8	20.0	42.7	20.2	41.6
2060-1688	99.6	20.6	39.8	20.2	42.7
2060-1689	99.6	20.7	38.2	20.0	39.2

Also given in the attached tables and computed from the octave band sound reduction indices, is the weighted sound reduction index, R_w, calculated according to ISO 717-1:2013. This evaluation is based on laboratory measurement results obtained by an engineering method.

The results here presented relate only to the items tested and described in this report.

AIR HANDLING UNIT CASEWORK SRI VALUES								
Reference	**63Hz**	**125 Hz**	**250 Hz**	**500 Hz**	**1K Hz**	**2K Hz**	**4K Hz**	**8K Hz**
FG18	18.3	18.1	24	38.8	39.9	34.5	33.1	39.9
FG25	16.5	16.6	26.6	38.1	39.2	33.4	32.1	39.7
FG50	15.7	18.4	33.7	36.2	37.5	32.4	35	40.4
PB18	21	25.9	28.2	29.3	33.1	34.5	33.7	39.8
SBP18	21.6	24.9	27.9	28	34.8	35.5	33.6	39.2
PB25	20.4	23.0	22.9	23.3	33.6	37	37.7	45.5
PBSBP25	22.5	26.2	29.4	30	36.4	35.2	33.6	39.6
PB50	21.8	23.8	22.1	26	35.6	35.2	32.0	40.1
AS25	19.9	23.5	29.4	36	39.1	34.1	34.5	40.8
ASPB50	20.8	21.3	29.6	39	41.4	34.8	32.8	40.2
PBFG50	19.6	18.3	33.3	39.8	38.6	36.1	39.3	46.8
TRIPLE SKIN								
PB/TS/PB25	23.8	26.6	29	30.6	36.7	35.6	33.3	40.1
PB/TS/FG50	21.1	21.0	36.3	39.2	41.7	34.7	32.9	40.2
SPB/TS/SPB50	21.7	24.2	36.4	37.3	37.6	33.4	35.7	41.5
AS/TS/FG50	21	23.7	34.2	36.1	37	32.7	36.2	40.3
AS/TS/PB50	23.4	23.5	34.3	39.8	38.9	36	39.8	46.9
AS/TS/SBP50	24.1	27.1	34.8	37	37.2	33.5	36.2	39.6
AS/TS/AS50	21.6	25.6	32	37	40.8	33.6	32.9	24.3
QUADRUPLE SKIN								
AS/QS/PB50	24.1	36.6	34.7	39.8	40.7	37.8	39.7	43.7

Insertion Loss Graph's

The following graphs relate to the Test Reports 1429 &2060. The test range for these graphs is 63 KHz to 4 KHz, results for 8K Hz are given in Appendix 'A'.

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client: Air Handlers Northern — Date of test: 07/01/15
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: FG18 DOUBLE SKIN

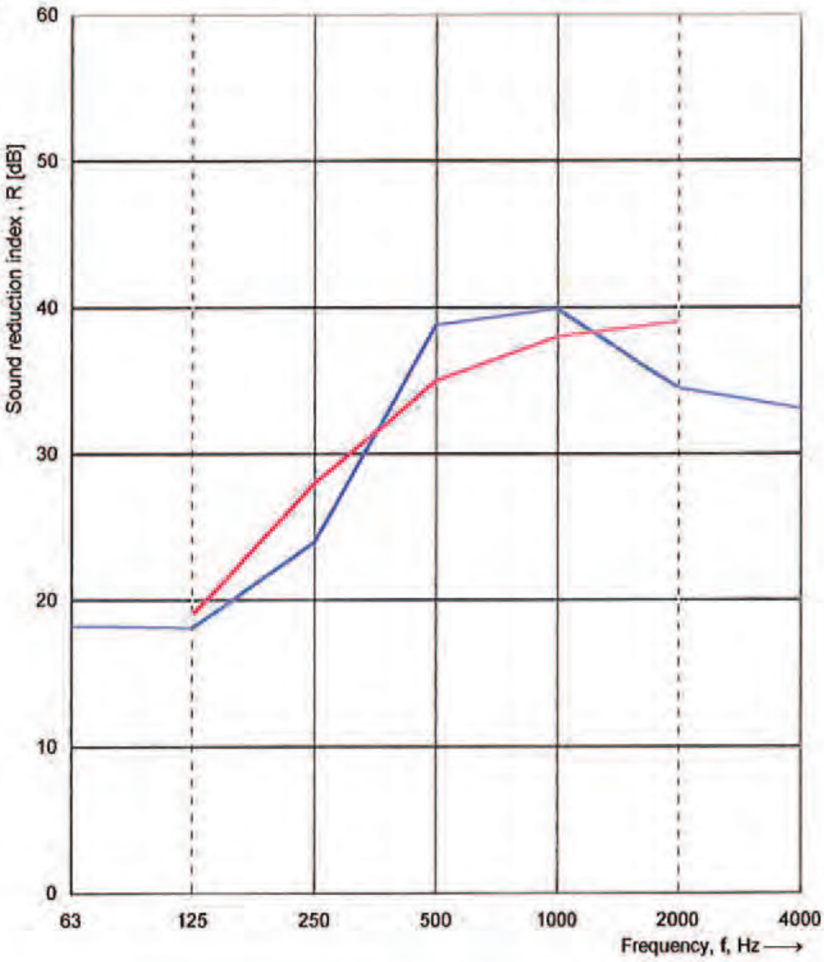
Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP
Static pressure: 101.2 ± 0.0 kPa
Air temperature: 19.9 ± 0.3 °C
Relative air humidity: 38.7 ± 3.0 %
Mass per unit area: 19.4 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R
Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	18.3
125	18.1
250	24.0
500	38.8
1000	39.9
2000	34.5
4000	33.1



Rating according to BS EN ISO 717-1
$R_w(C; C_{tr})$ = 35 (-2 ; -6) dB
Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -2 dB $C_{125\text{-}4000}$ = -2 dB
$C_{tr,63\text{-}2000}$ = -6 dB $C_{tr,63\text{-}4000}$ = -7 dB $C_{tr,125\text{-}4000}$ = -6 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory
Test Reference: 2060-1675

Date: 07/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

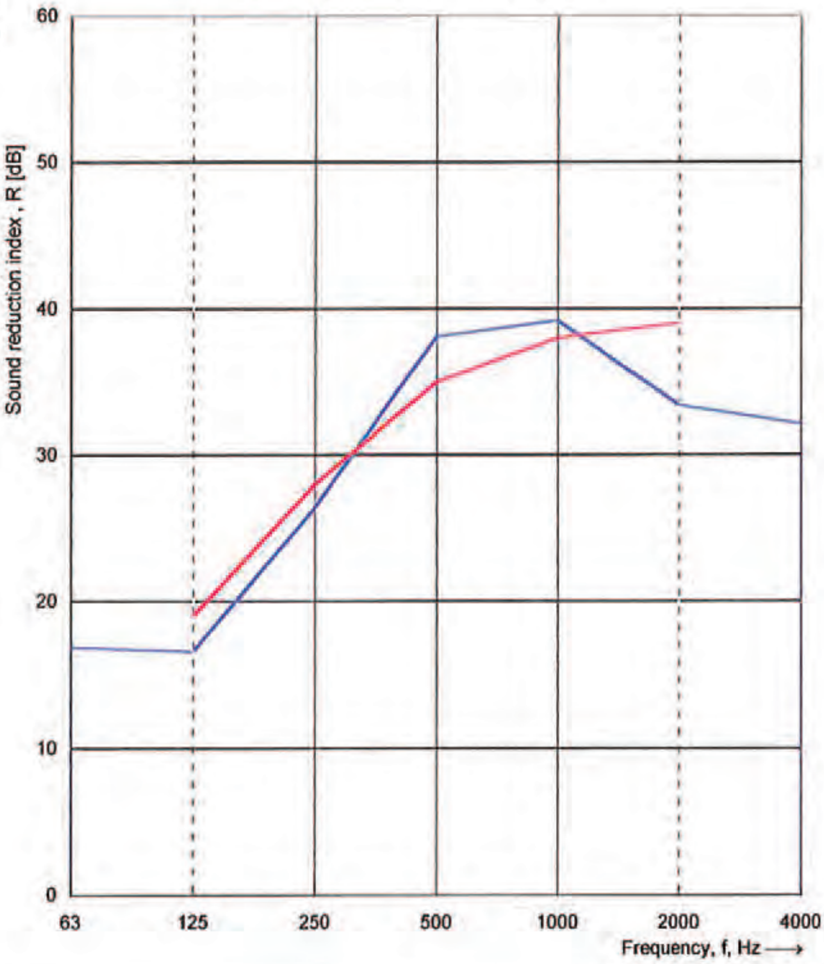
Client:	Air Handlers Northern	Date of test:	09/01/15
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	FG25 DOUBLE SKIN		
Description of the specimen:	Acoustic Panel System Please refer to Section 1 for a full description of the test object		
Curing time:	NAP		
Static pressure:	100.8 ± 0.0 kPa		
Air temperature:	19.7 ± 0.3 °C		
Relative air humidity:	40.6 ± 3.0 %		
Mass per unit area:	21.3 kg/m²		
Area, S, of the test sample:	8.64 m²		
Source room volume:	136 m³		
Receiving room volume:	220 m³		

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	16.9
125	16.6
250	26.4
500	38.1
1000	39.2
2000	33.4
4000	32.1



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 35 (-2 ; -6) dB

$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -3 dB $C_{125\text{-}4000}$ = -3 dB

$C_{tr,63\text{-}2000}$ = -7 dB $C_{tr,63\text{-}4000}$ = -7 dB $C_{tr,125\text{-}4000}$ = -7 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1686

Date: 09/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client: Air Handlers Northern

Date of test: 09/01/15

Manufacturer: Client

Test room identification: Acoustic Transmission Suite

Test specimen mounted by: Client

Product identification: FG50 DOUBLE SKIN

Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP

Static pressure: 100.8 ± 0.0 kPa

Air temperature: 20.2 ± 0.3 °C

Relative air humidity: 41.6 ± 3.0 %

Mass per unit area: 23.3 kg/m²

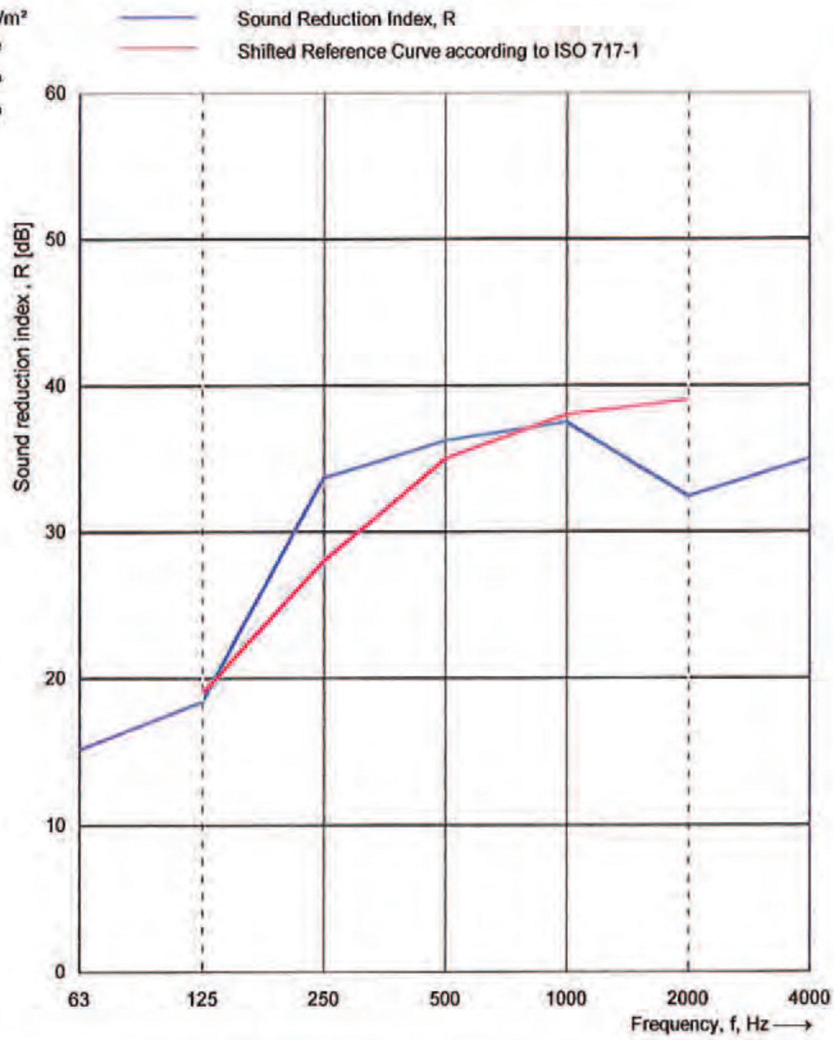
Area, S, of the test sample: 8.64 m²

Source room volume: 136 m³

Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/1 octave [dB]
63	15.2
125	18.4
250	33.7
500	36.2
1000	37.5
2000	32.4
4000	35.0



Rating according to BS EN ISO 717-1

$R_w (C ; C_{tr})$ = 35 (-2 ; -4) dB

$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -2 dB $C_{125\text{-}4000}$ = -2 dB

$C_{tr,63\text{-}2000}$ = -6 dB $C_{tr,63\text{-}4000}$ = -6 dB $C_{tr,125\text{-}4000}$ = -5 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1687

Date: 09/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

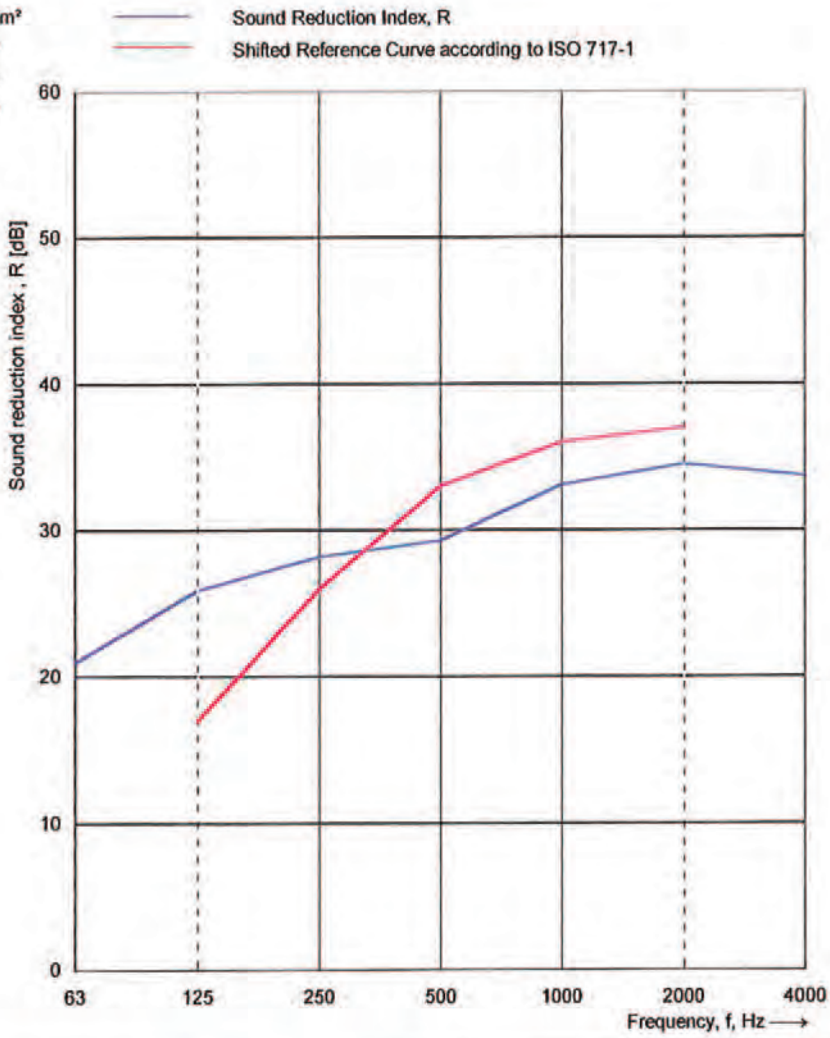
Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern
Manufacturer:	Client
Test room identification:	Acoustic Transmission Suite
Test specimen mounted by:	Client
Product identification:	PB18 DOUBLE SKIN
Description of the specimen:	Acoustic Panel System Please refer to Section 1 for a full description of the test object
Curing time:	NAP
Static pressure:	101.2 ± 0.0 kPa
Air temperature:	23.1 ± 0.3 °C
Relative air humidity:	30.0 ± 3.0 %
Mass per unit area:	18.0 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

Date of test: 07/01/15

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/1 octave [dB]
63	21.0
125	25.9
250	28.2
500	29.3
1000	33.1
2000	34.5
4000	33.7



Rating according to BS EN ISO 717-1

$R_w (C ; C_{tr})$ = 33 (-1 ; -2) dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{63\text{-}2000}$ = -1 dB $C_{63\text{-}4000}$ = -1 dB $C_{125\text{-}4000}$ = -1 dB

$C_{tr,63\text{-}2000}$ = -3 dB $C_{tr,63\text{-}4000}$ = -3 dB $C_{tr,125\text{-}4000}$ = -3 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1673

Date: 07/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern	Date of test:	08/01/15
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	SBP18 DOUBLE SKIN		
Description of the specimen:	Acoustic Panel System Please refer to Section 1 for a full description of the test object		

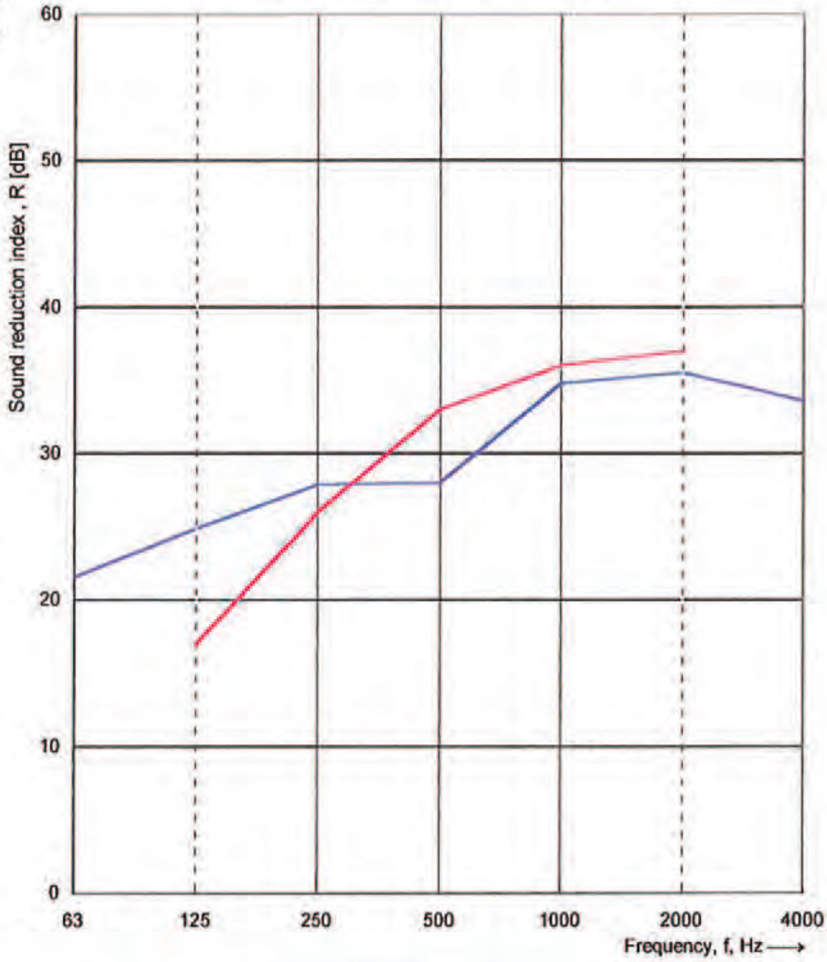
Curing time:	NAP
Static pressure:	101.3 ± 0.0 kPa
Air temperature:	19.6 ± 0.3 °C
Relative air humidity:	37.0 ± 3.0 %
Mass per unit area:	30.6 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	21.6
125	24.9
250	27.9
500	28.0
1000	34.8
2000	35.5
4000	33.6



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 33 (0 ; -2) dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{63\text{-}2000}$ = -1 dB $C_{63\text{-}4000}$ = -1 dB $C_{125\text{-}4000}$ = -1 dB

$C_{tr,63\text{-}2000}$ = -3 dB $C_{tr,63\text{-}4000}$ = -3 dB $C_{tr,125\text{-}4000}$ = -3 dB

Name of test institute:	The University of Salford, Acoustic Test Laboratory
Test Reference:	2060-1681
Date: 08/01/15	Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

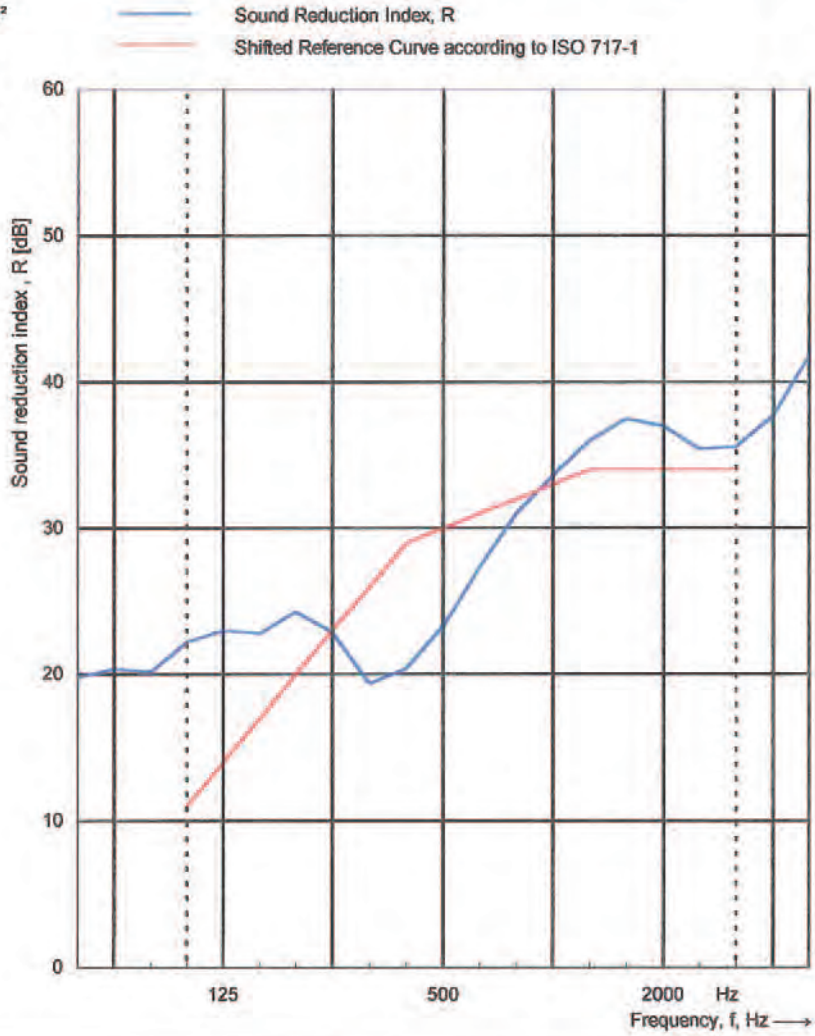
Client: Air Handlers Northern
Date of test: 01/10/13
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: "PB25" Panel Enclosure System

Description of the specimen: Please refer to Section 1 for a full description of the test object
Frame capped using "AcousticShield" on the source room side of the test object

Curing time: NAP
Static pressure: 100.8 kPa
Air temperature: 21.4 °C
Relative air humidity: 51.6 %
Mass per unit area: 22.2 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

FREQUENCIES 50HZ, 63HZ AND 80HZ ARE NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/3 octave [dB]
50	19.8
63	20.4
80	20.2
100	22.2
125	23.0
160	22.8
200	24.3
250	22.9
315	19.4
400	20.4
500	23.3
630	27.3
800	31.0
1000	33.6
1250	36.0
1600	37.5
2000	37.0
2500	35.4
3150	35.6
4000	37.7
5000	41.8



Rating according to BS EN ISO 717-1

$R_w(C;C_{tr})$ = 30 (-1 ; -3) dB

$C_{50-3150}$ = -1 dB $C_{50-5000}$ = 0 dB $C_{100-5000}$ = 0 dB

$C_{tr,50-3150}$ = -3 dB $C_{tr,50-5000}$ = -3 dB $C_{tr,100-5000}$ = -3 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory
No. of test report: 1429-1208

Date: 01/10/13 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client: Air Handlers Northern

Date of test: 08/01/15

Manufacturer: Client

Test room identification: Acoustic Transmission Suite

Test specimen mounted by: Client

Product identification: PB/SBP50 DOUBLE SKIN

Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP

Static pressure: 101.3 ± 0.0 kPa

Air temperature: 19.7 ± 0.3 °C

Relative air humidity: 37.3 ± 3.0 %

Mass per unit area: 34.2 kg/m²

Area, S, of the test sample: 8.64 m²

Source room volume: 136 m³

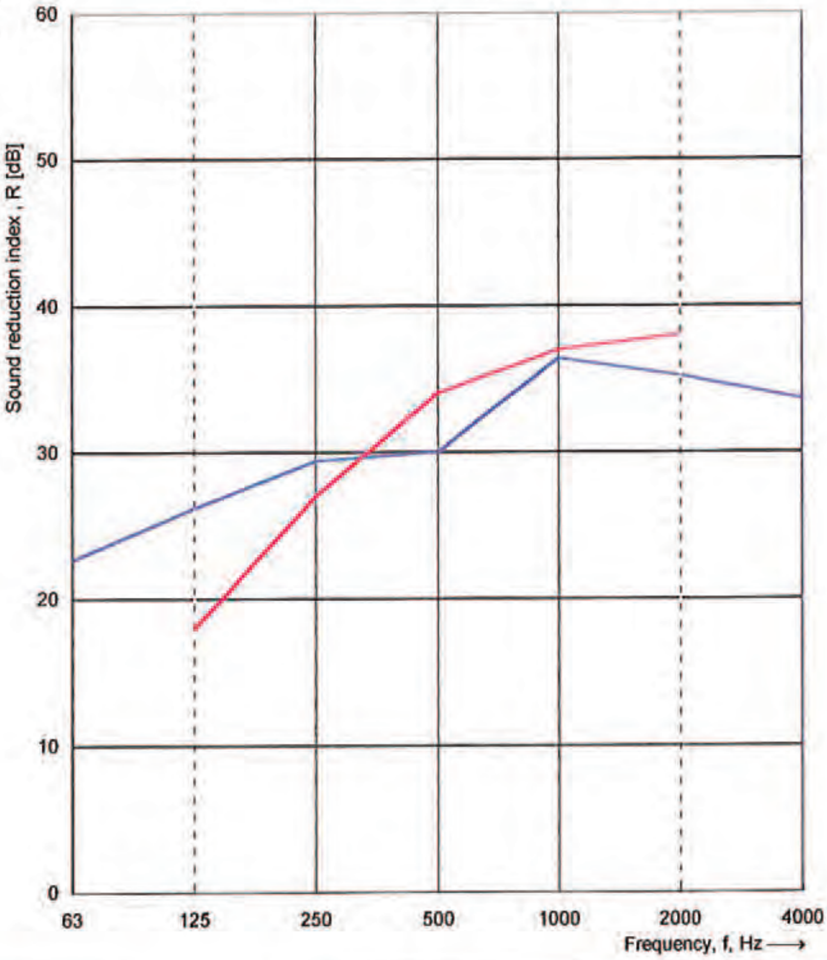
Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	22.7
125	26.2
250	29.4
500	30.0
1000	36.4
2000	35.2
4000	33.6



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 34 (0 ; -2) dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{63-2000}$ = 0 dB $C_{63-4000}$ = -1 dB $C_{125-4000}$ = -1 dB

$C_{tr,63-2000}$ = -2 dB $C_{tr,63-4000}$ = -2 dB $C_{tr,125-4000}$ = -3 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1680

Date: 08/01/15

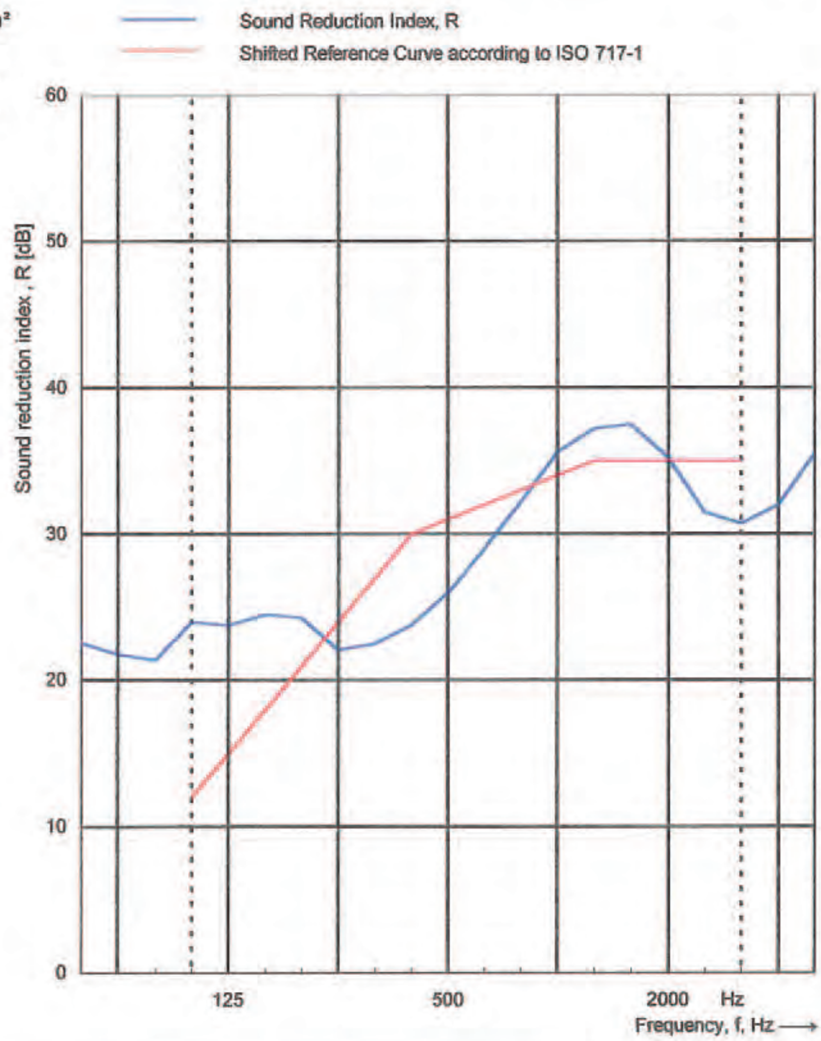
Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern	Date of test:	26/09/13
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	"PB50" Panel Enclosure System		
Description of the specimen:	Please refer to Section 1 for a full description of the test object		
Curing time:	NAP		
Static pressure:	101.7 kPa		
Air temperature:	21.1 °C		
Relative air humidity:	43.1 %		
Mass per unit area:	34.2 kg/m²		
Area, S, of the test sample:	8.64 m²		
Source room volume:	136 m³		
Receiving room volume:	220 m³		

FREQUENCIES 50HZ, 63HZ AND 80HZ ARE NOT UKAS ACCREDITED



Frequency f [Hz]	R 1/3 octave [dB]
50	22.6
63	21.8
80	21.4
100	24.0
125	23.8
160	24.5
200	24.3
250	22.1
315	22.5
400	23.8
500	26.0
630	29.0
800	32.2
1000	35.6
1250	37.2
1600	37.5
2000	35.2
2500	31.5
3150	30.7
4000	32.0
5000	35.5

Rating according to BS EN ISO 717-1

R_w (C ; C_{tr}) = 31 (-1 ; -2) dB

$C_{50\text{-}3150}$ = -1 dB $C_{50\text{-}5000}$ = 0 dB $C_{100\text{-}5000}$ = 0 dB

$C_{tr,50\text{-}3150}$ = -3 dB $C_{tr,50\text{-}5000}$ = -3 dB $C_{tr,100\text{-}5000}$ = -2 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory

No. of test report: 1429-1197

Date: 26/09/13 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern	Date of test:	07/01/15
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	AS25 DOUBLE SKIN		
Description of the specimen:	Acoustic Panel System Please refer to Section 1 for a full description of the test object		

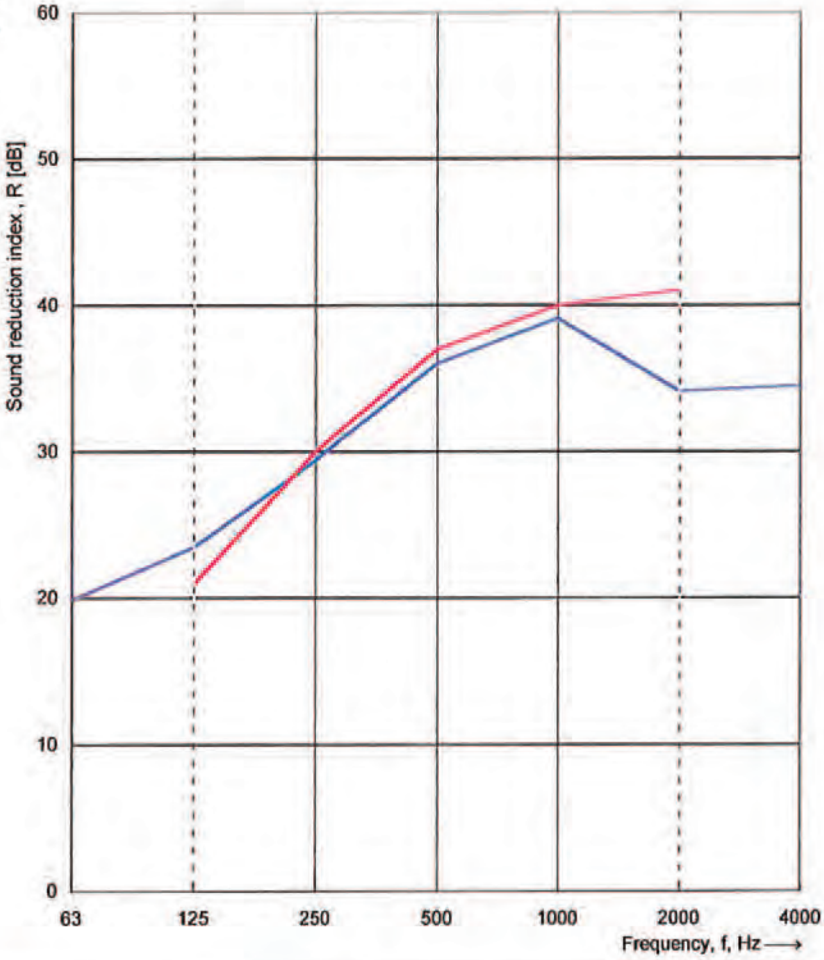
Curing time:	NAP
Static pressure:	101.2 ± 0.0 kPa
Air temperature:	20.0 ± 0.3 °C
Relative air humidity:	37.3 ± 3.0 %
Mass per unit area:	31.3 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	19.9
125	23.5
250	29.4
500	36.0
1000	39.1
2000	34.1
4000	34.5



Rating according to BS EN ISO 717-1

$R_w (C ; C_{tr})$ = 37 (-2 ; -4) dB

$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -3 dB $C_{125\text{-}4000}$ = -3 dB

$C_{tr,63\text{-}2000}$ = -5 dB $C_{tr,63\text{-}4000}$ = -5 dB $C_{tr,125\text{-}4000}$ = -5 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute:	The University of Salford, Acoustic Test Laboratory
Test Reference:	2060-1674
Date: 07/01/15	Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern
Manufacturer:	Client
Test room identification:	Acoustic Transmission Suite
Test specimen mounted by:	Client
Product identification:	"ASPB50" Panel Enclosure System
Description of the specimen:	Please refer to Section 1 for a full description of the test object

Date of test: 26/09/13

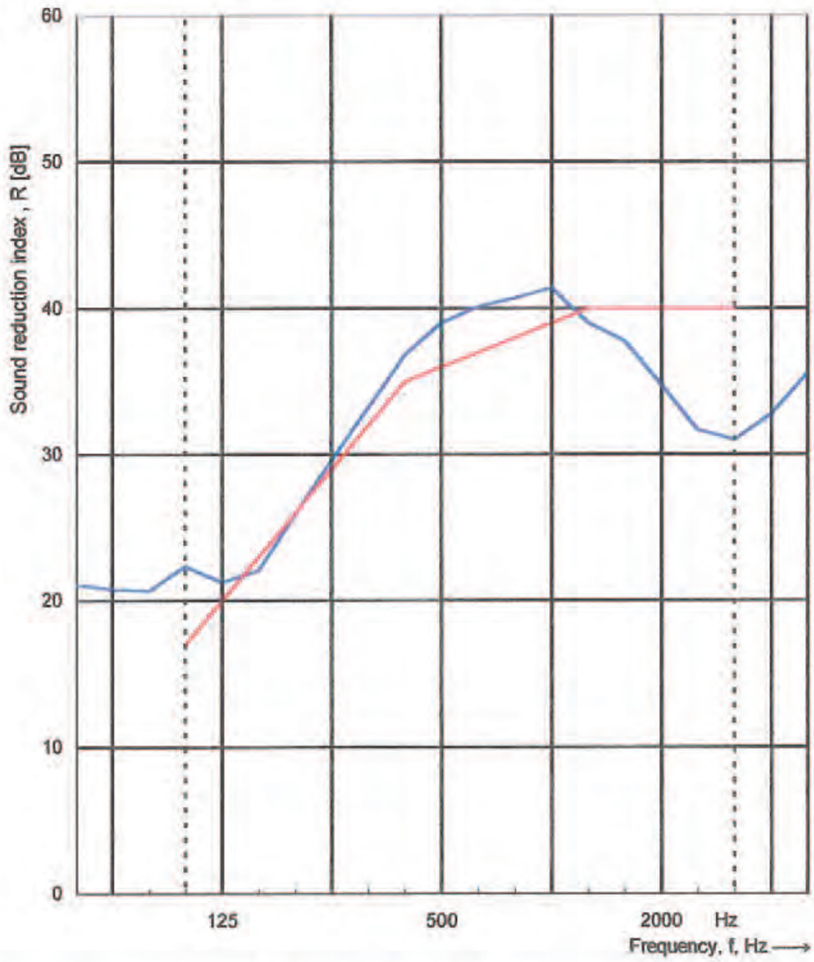
Curing time:	NAP
Static pressure:	101.7 kPa
Air temperature:	21.6 °C
Relative air humidity:	48.0 %
Mass per unit area:	35.2 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

FREQUENCIES 50HZ, 63HZ AND 80HZ ARE NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/3 octave [dB]
50	21.1
63	20.8
80	20.7
100	22.4
125	21.3
160	22.1
200	25.9
250	29.6
315	33.2
400	36.8
500	39.0
630	40.1
800	40.7
1000	41.4
1250	39.0
1600	37.8
2000	34.8
2500	31.7
3150	31.0
4000	32.8
5000	35.6

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1



Rating according to BS EN ISO 717-1

$R_w (C ; C_{tr})$ = 36 (-2 ; -3) dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{50\text{-}3150}$ = -2 dB $C_{50\text{-}5000}$ = -2 dB $C_{100\text{-}5000}$ = -2 dB

$C_{tr,50\text{-}3150}$ = -4 dB $C_{tr,50\text{-}5000}$ = -4 dB $C_{tr,100\text{-}5000}$ = -3 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory

No. of test report: 1429-1199

Date: 26/09/13 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

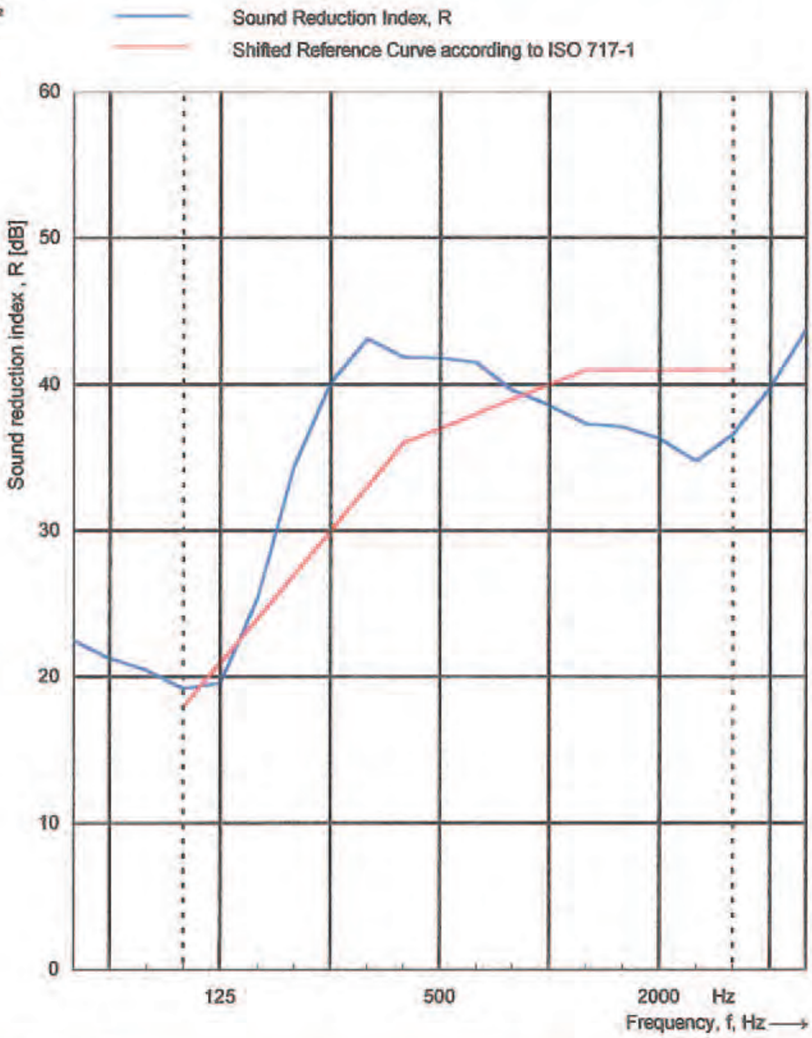
Client: Air Handlers Northern — Date of test: 01/10/13
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: "PBFG50 with 1.2mm plastisol" Panel Enclosure System

Description of the specimen: Please refer to Section 1 for a full description of the test object
Frame capped using "AcousticShield" on the source room side of the test object

Curing time: NAP
Static pressure: 100.8 kPa
Air temperature: 21.2 °C
Relative air humidity: 49.6 %
Mass per unit area: 29.7 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

FREQUENCIES 50HZ, 63HZ AND 80HZ ARE NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/3 octave [dB]
50	22.5
63	21.3
80	20.5
100	19.2
125	19.6
160	25.4
200	34.4
250	40.1
315	43.1
400	41.9
500	41.8
630	41.5
800	39.6
1000	38.6
1250	37.3
1600	37.1
2000	36.3
2500	34.8
3150	36.6
4000	39.8
5000	43.8



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 37 (-1 ; -3) dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{50-3150}$ = -1 dB $C_{50-5000}$ = 0 dB $C_{100-5000}$ = 0 dB
$C_{tr,50-3150}$ = -4 dB $C_{tr,50-5000}$ = -4 dB $C_{tr,100-5000}$ = -3 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory
No. of test report: 1429-1206

Date: 01/10/13 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern	Date of test:	08/01/15
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	PB/TS/PB25 TRIPLE SKIN		
Description of the specimen:	Acoustic Panel System Please refer to Section 1 for a full description of the test object		

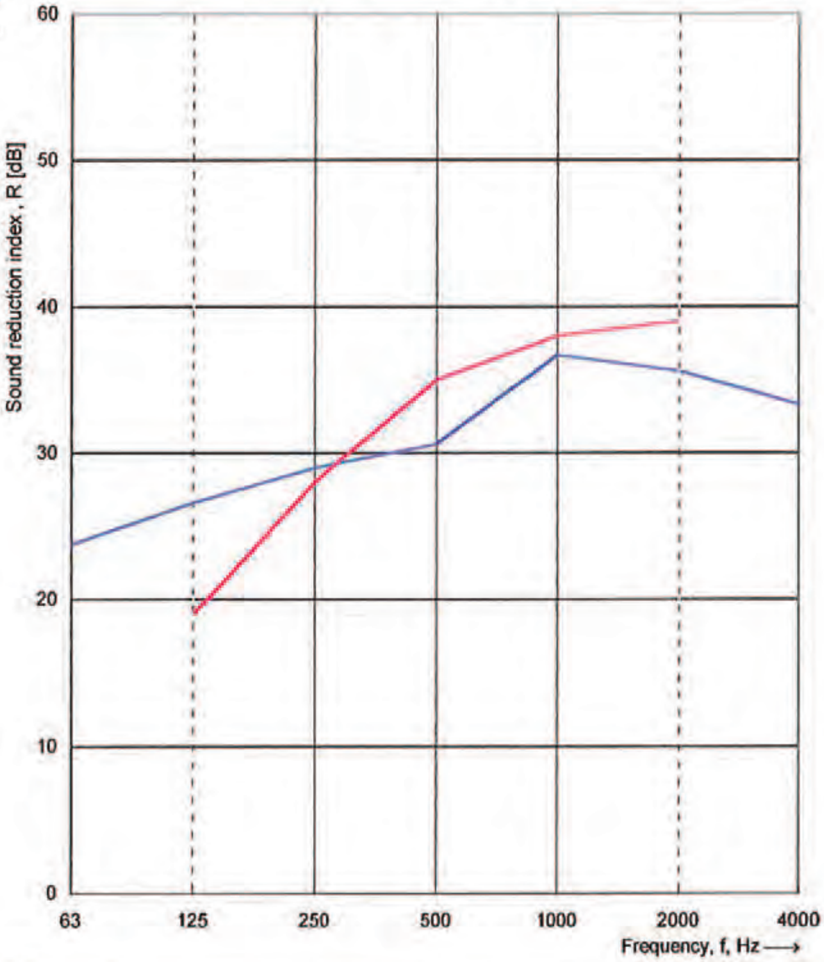
Curing time:	NAP
Static pressure:	101.3 ± 0.0 kPa
Air temperature:	21.5 ± 0.3 °C
Relative air humidity:	34.9 ± 3.0 %
Mass per unit area:	41.0 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	23.8
125	26.6
250	29.0
500	30.6
1000	36.7
2000	35.6
4000	33.3



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 35 (-1 ; -2) dB

$C_{63-2000}$ = -1 dB $C_{63-4000}$ = -2 dB $C_{125-4000}$ = -2 dB

$C_{tr,63-2000}$ = -3 dB $C_{tr,63-4000}$ = -3 dB $C_{tr,125-4000}$ = -3 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1678

Date: 08/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client: Air Handlers Northern Date of test: 09/01/15
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: AS/TS/FG50 TRIPLE SKIN

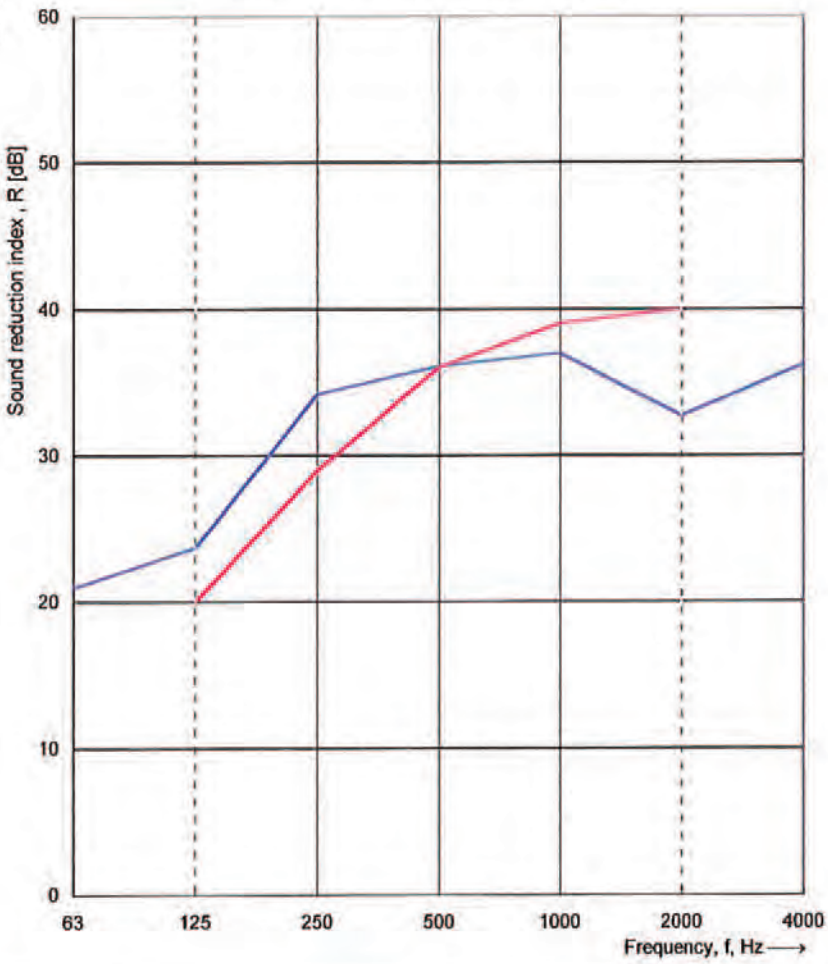
Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP
Static pressure: 100.8 ± 0.0 kPa
Air temperature: 19.6 ± 0.3 °C
Relative air humidity: 39.6 ± 3.0 %
Mass per unit area: 50.7 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R
Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	21.0
125	23.7
250	34.2
500	36.1
1000	37.0
2000	32.7
4000	36.2



Rating according to BS EN ISO 717-1
R_w (C ; C_{tr}) = 36 (-2 ; -3) dB
$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -2 dB $C_{125\text{-}4000}$ = -2 dB
$C_{tr,63\text{-}2000}$ = -4 dB $C_{tr,63\text{-}4000}$ = -4 dB $C_{tr,125\text{-}4000}$ = -4 dB
Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory
Test Reference: 2060-1683

Date: 09/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

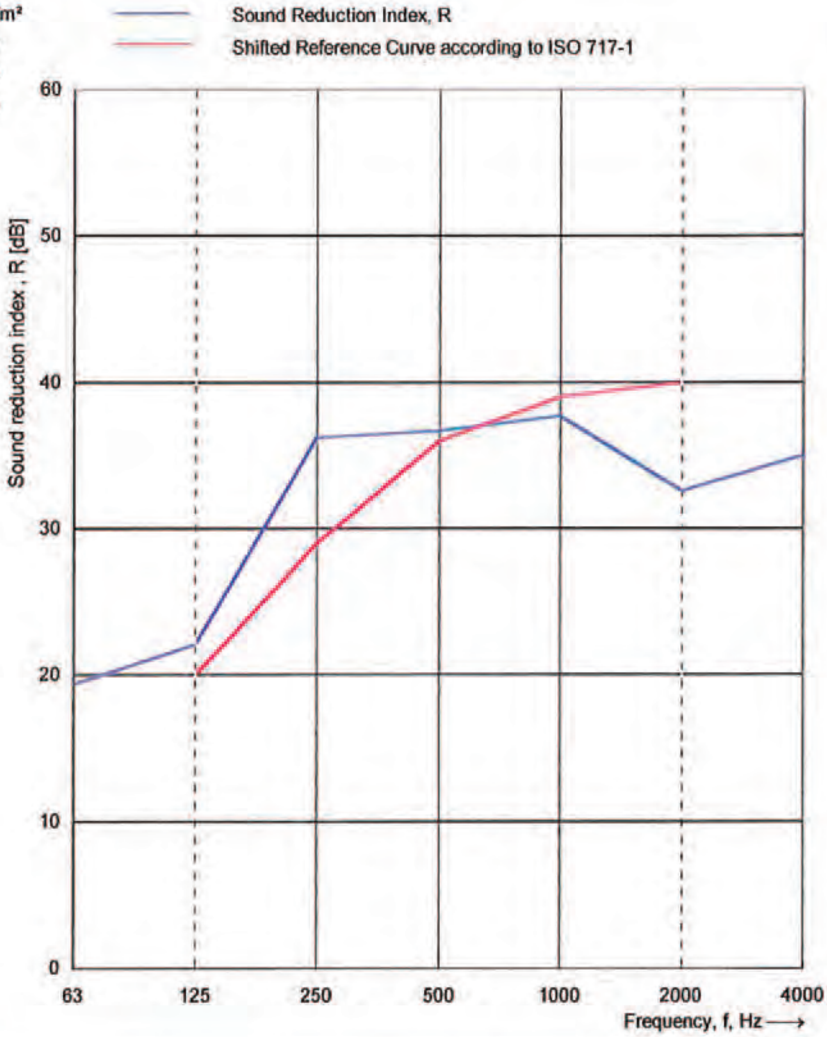
Client: Air Handlers Northern
Date of test: 09/01/15
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: FG/TS/PB50 TRIPLE SKIN

Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP
Static pressure: 100.8 ± 0.0 kPa
Air temperature: 20.1 ± 0.3 °C
Relative air humidity: 39.9 ± 3.0 %
Mass per unit area: 40.0 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/1 octave [dB]
63	19.4
125	22.1
250	36.2
500	36.7
1000	37.7
2000	32.6
4000	35.0



Rating according to BS EN ISO 717-1
$R_w(C ; C_{tr})$ = 36 (-2 ; -3) dB
Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

$C_{63\text{-}2000}$ = -2 dB $C_{63\text{-}4000}$ = -2 dB $C_{125\text{-}4000}$ = -2 dB
$C_{tr,63\text{-}2000}$ = -4 dB $C_{tr,63\text{-}4000}$ = -5 dB $C_{tr,125\text{-}4000}$ = -4 dB

Name of test institute: The University of Salford, Acoustic Test Laboratory
Test Reference: 2060-1685

Date: 09/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

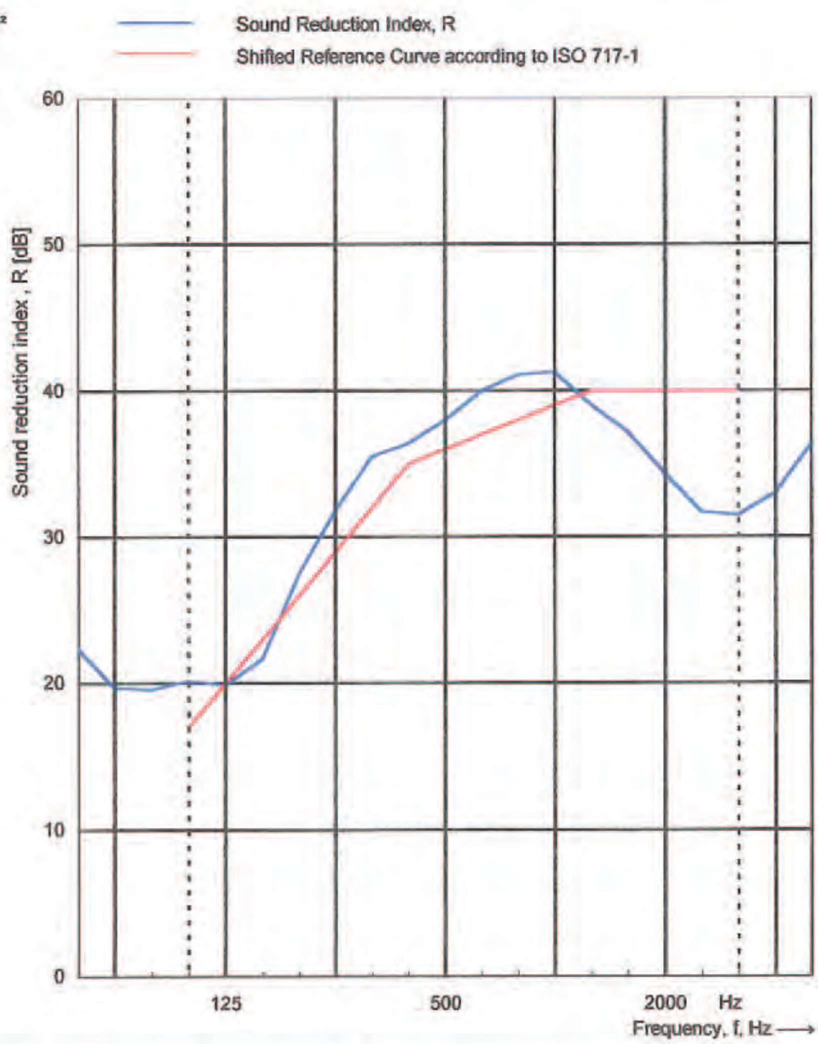
Client: Air Handlers Northern
Date of test: 27/09/13
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: "ASTSFG50" Panel Enclosure System

Description of the specimen: Please refer to Section 1 for a full description of the test object

Curing time: NAP
Static pressure: 101.6 kPa
Air temperature: 21.6 °C
Relative air humidity: 49.3 %
Mass per unit area: 38.1 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

FREQUENCIES 50HZ, 63HZ AND 80HZ ARE NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/3 octave [dB]
50	22.4
63	19.7
80	19.6
100	20.2
125	19.9
160	21.7
200	27.5
250	31.8
315	35.5
400	36.4
500	38.0
630	40.0
800	41.1
1000	41.3
1250	39.0
1600	37.2
2000	34.3
2500	31.7
3150	31.5
4000	33.0
5000	36.4



Rating according to BS EN ISO 717-1

$R_w (C ; C_{tr})$ = 36 (-2 ; -3) dB

$C_{50-3150}$ = -2 dB $C_{50-5000}$ = -2 dB $C_{100-5000}$ = -2 dB

$C_{tr,50-3150}$ = -5 dB $C_{tr,50-5000}$ = -5 dB $C_{tr,100-5000}$ = -4 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory
No. of test report: 1429-1201

Date: 27/09/13 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

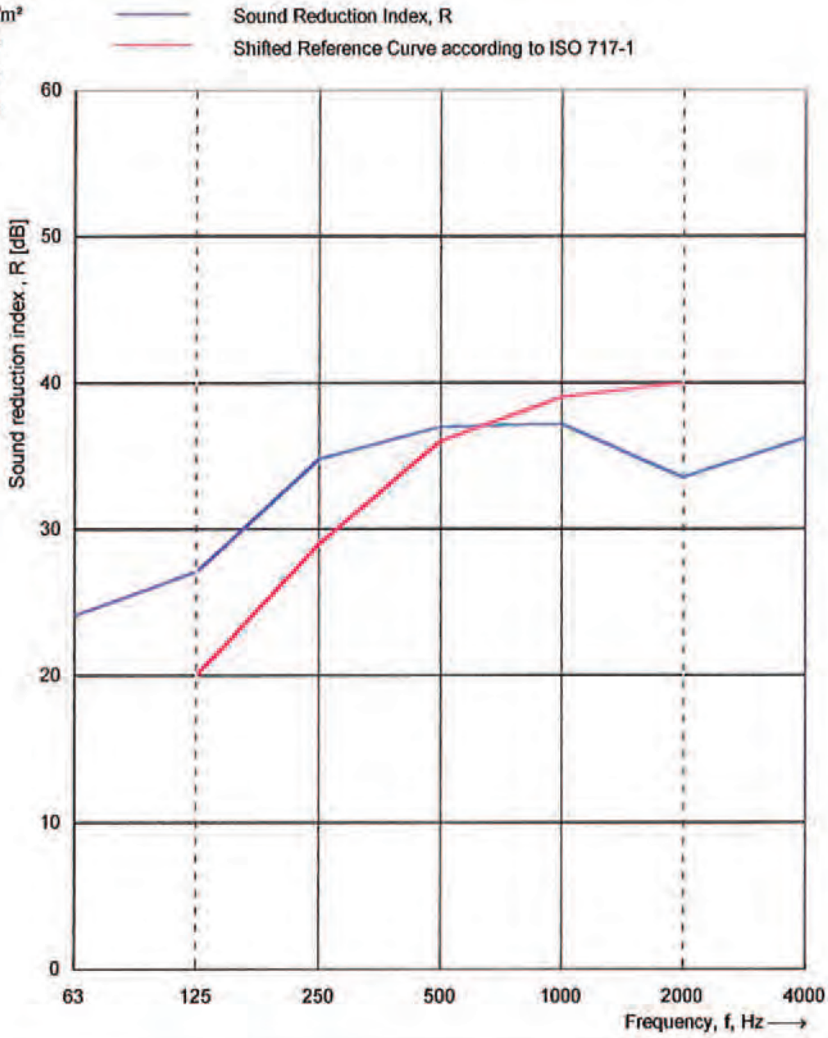
Client: Air Handlers Northern
Date of test: 08/01/15
Manufacturer: Client
Test room identification: Acoustic Transmission Suite
Test specimen mounted by: Client
Product identification: AS/TS/SBP50 TRIPLE SKIN

Description of the specimen: Acoustic Panel System
Please refer to Section 1 for a full description of the test object

Curing time: NAP
Static pressure: 101.3 ± 0.0 kPa
Air temperature: 19.8 ± 0.3 °C
Relative air humidity: 36.7 ± 3.0 %
Mass per unit area: 53.6 kg/m²
Area, S, of the test sample: 8.64 m²
Source room volume: 136 m³
Receiving room volume: 220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Frequency f [Hz]	R 1/1 octave [dB]
63	24.1
125	27.1
250	34.8
500	37.0
1000	37.2
2000	33.5
4000	36.2



Rating according to BS EN ISO 717-1

$R_w(C;C_{tr})$ = 36 (-1 ; -1) dB

$C_{63-2000}$ = -1 dB $C_{63-4000}$ = -1 dB $C_{125-4000}$ = -1 dB

$C_{tr,63-2000}$ = -2 dB $C_{tr,63-4000}$ = -2 dB $C_{tr,125-4000}$ = -2 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory
Test Reference: 2060-1679

Date: 08/01/15 Signature:

Sound reduction index according to BS EN ISO 10140-2

Laboratory measurement of sound insulation of building elements

Client:	Air Handlers Northern	Date of test:	12/01/15
Manufacturer:	Client		
Test room identification:	Acoustic Transmission Suite		
Test specimen mounted by:	Client		
Product identification:	AS/QS/PB50 QUADRUPLE SKIN		
Description of the specimen:	Acoustic Panel System - FULLY SEALED Please refer to Section 1 for a full description of the test object		

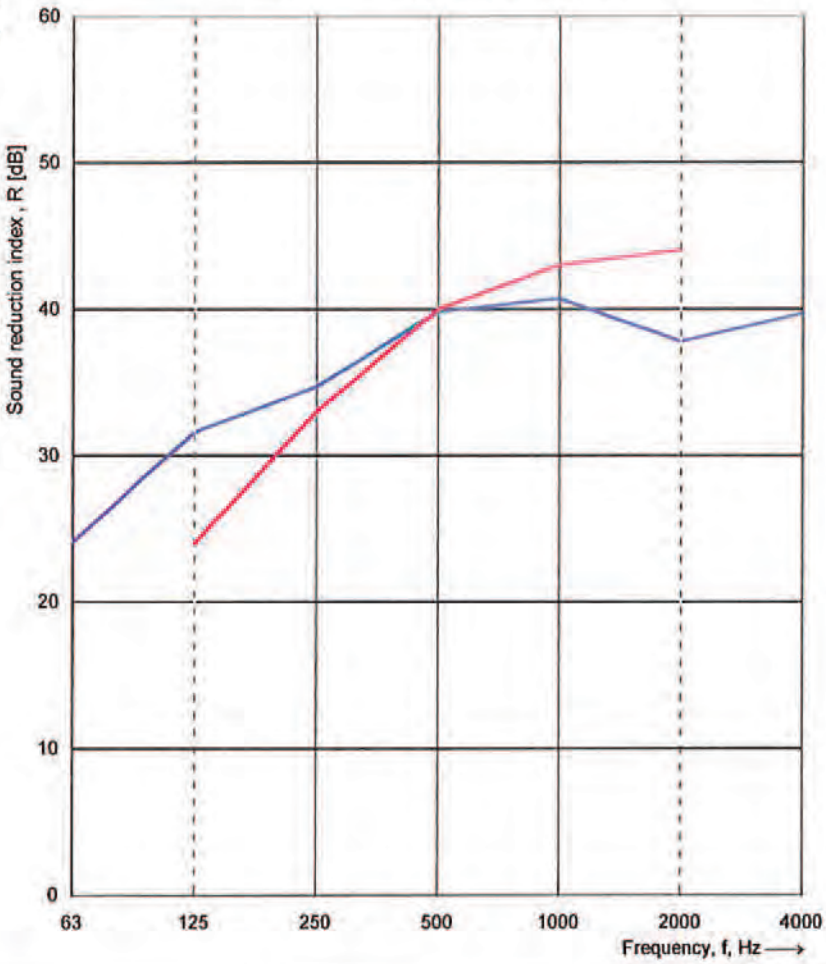
Curing time:	NAP
Static pressure:	99.6 ± 0.0 kPa
Air temperature:	20.0 ± 0.3 °C
Relative air humidity:	39.2 ± 3.0 %
Mass per unit area:	62.9 kg/m²
Area, S, of the test sample:	8.64 m²
Source room volume:	136 m³
Receiving room volume:	220 m³

THE FREQUENCY 63HZ IS NOT UKAS ACCREDITED

Sound Reduction Index, R

Shifted Reference Curve according to ISO 717-1

Frequency f [Hz]	R 1/1 octave [dB]
63	24.1
125	31.6
250	34.7
500	39.8
1000	40.7
2000	37.8
4000	39.7



Rating according to BS EN ISO 717-1

$R_w(C ; C_{tr})$ = 40 (-1 ; -2) dB

$C_{63\text{-}2000}$ = -1 dB $C_{63\text{-}4000}$ = -1 dB $C_{125\text{-}4000}$ = -1 dB

$C_{tr,63\text{-}2000}$ = -3 dB $C_{tr,63\text{-}4000}$ = -4 dB $C_{tr,125\text{-}4000}$ = -3 dB

Evaluation based on laboratory measurement results obtained in one-third-octave bands by an engineering method.

Name of test institute: The University of Salford, Acoustic Test Laboratory

Test Reference: 2060-1689

Date: 12/01/15 Signature:

University of Salford MANCHESTER

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TEST REPORT No : 2060-1 **DATE OF ISSUE : 19 February 2015**

INTERNATIONAL STANDARD METHOD FOR MEASUREMENT OF AIRBORNE SOUND INSULATION OF BUILDING ELEMENTS BS EN ISO 10140-2 : 2010

CLIENT:	Air Handlers Northern Bute Street Weaste Salford M50 1DU
JOB NUMBER:	ACOUS/02060
TEST SAMPLE:	Various Acoustic Enclosure Panels
MANUFACTURER:	Client
DATE RECEIVED:	5 November 2014
DATE OF TEST:	7 – 9 & 12 January 2015

Signed:............................ Approved:..............................

I G Rattigan C Lomax
Laboratory Manager Quality Manager

University of Salford MANCHESTER

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UKAS TESTING
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TEST REPORT No : 1429-A **DATE OF ISSUE : 7 November 2013**

BS EN ISO 10140-2 : 2010
INTERNATIONAL STANDARD METHOD FOR MEASUREMENT OF AIRBORNE SOUND INSULATION OF BUILDING ELEMENTS

CLIENT:	Air Handlers Northern Bute Street Weaste, Salford M50 1DU
JOB NUMBER:	ACOUS/01429
MANUFACTURER:	Air Handlers Northern
TEST SAMPLE:	Various panel enclosure systems
DATE RECEIVED:	19 July 2013
DATE OF TEST:	26 September - 1 October 2013

Signed: .... Approved: ....

I G Rattigan
Laboratory Manager

D J McCaul
Technical Manager

Factory Noise Tests

It is becoming increasingly more common to Factory Test Mechanical Ventilation Plant for performance, which often includes noise breakout. An example of such a Noise Test shows how the sound insulation of a supply and extract ventilation plant was factory tested for an extremely noise sensitive project at Bodleian Library at Oxford University.

The Air Handling Unit was tested for air leakage and volumetric performance prior to the noise testing. Suitable attenuators were provided on the test rig to reduce the atmospheric and system side noise emission to a level well below the Air Handling Unit casework noise emission. The test report describes the noise test and method of evaluation the sound power and pressure.

Air Handlers Northern Ltd has such a testing facility at their manufacturing plant in Salford, Manchester.

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Measurement of Machine Noise Emissions

Prepared: 28th July 2013

Report No – 14076-1
Client – Air Handlers (Northern) Ltd

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1. Contents

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2. Scope

2.1.1. This report details the findings of a machinery sound power and pressure survey of an Air Handling Unit (AHU) at Air Handlers (Northern) Ltd, Albert Proctor House, Bute St, Salford, Manchester M50 1DU.

2.1.2. The sound power measurement and calculation procedure was in accordance with ISO 3746:2009 *"Acoustics — Determination of sound power levels of noise sources using sound pressure — Survey method using an enveloping measurement surface over a reflecting plane"*.

3. Introduction

3.1. Introduction

3.1.1. The client scheduled a witnessed sound pressure level / power Level test at the Northern factory in Salford in order to establish the machinery noise emissions. The sound emissions from the Air Handlers AHU have been measured using guidance from international measurement standards.

3.2. Measurement Criteria

3.2.1. It is understood that as part of the validation requirement for the air handling plant supplied to the Bodleian Library, University of Oxford a validated witnessed test is required in order to assess the sound pressure level break out / sound power levels generated by the plant

3.2.2. One or more of the following noise data may be required:

- Sound power level;
- Sound pressure level at 1 m distance, free field condition;
- Sound pressure level at operator's position, free field condition.

3.2.3. For sound power measurements, standards in the ISO 3740 series or ISO 9614 series should be used.

3.2.4. The AHU is remotely operated and there is no defined operator's position; therefore, this element has been excluded from the report.

3.3. BS EN ISO 3746:2009

3.3.1. The ISO3740 series of documents specifies various methods for determining the sound power levels of machines, equipment and their sub-assemblies. BS EN ISO3746:2009 specifies a method for measuring the sound pressure levels of a noise source and the calculation procedure to derive the item's sound power level.

3.3.2. The enveloping surface method can be used for any of three grades of accuracy and is used in this instance for grade 3 (survey) accuracy.

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4. Survey

4.1. Machine/equipment under test

Equipment Type	Air Handling Unit

Manufacturer	Air Handlers (Northern) Ltd
Part	Bodleian Library
Serial Number	AHN5952-4
Year of Manufacture	2013
Dimensions	5.15m x 1.31m x 1.82m

Table 1 - Machine/equipment details

4.2. Test Conditions

4.2.1. The air handling unit was run at full load for the duration of the measurements. The unit's main noise sources were the supply and extract fan/motor units; as part of a complete machine, all the metal components were radiating/re-radiating sound. It was noted that the noise emissions from the unit were relatively uniform across the casework area.

4.2.2. The unit was placed on a concrete (acoustically hard) floor inside the factory at Air Handlers (Northern) Ltd. An indicative layout plan is shown in Figure 1. Other items of equipment and machinery were present in the workshop but none breached the measurement surface.

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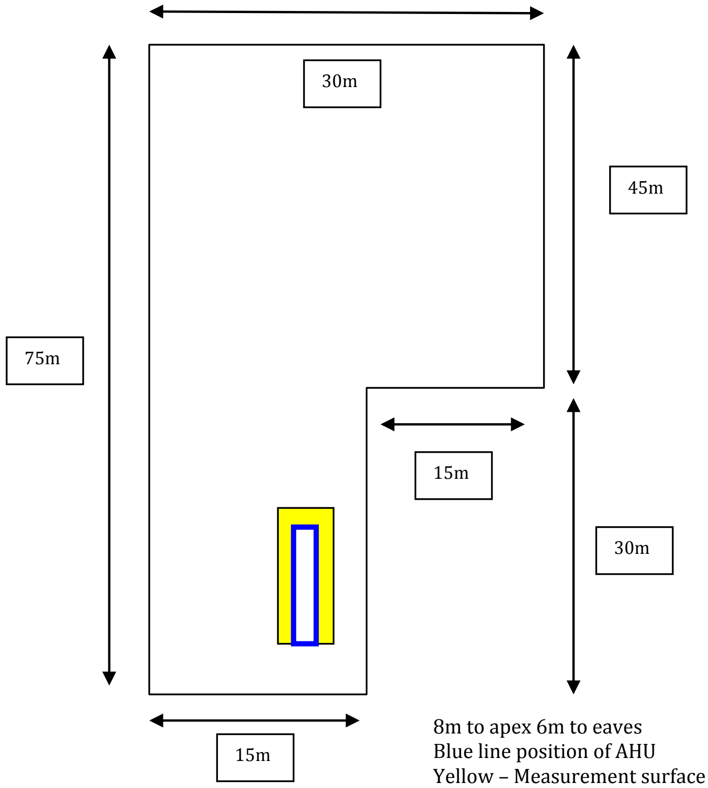

Figure 1 – Layout of the test environment (not to scale)

4.3. Acoustic environment

4.3.1. The workshop consisted of a concrete floor, concrete block walls, and a metal cladding exposed roof. The workshop contained a number of smaller pieces of equipment and two further air handling units that were under construction. The approximate surface areas of the workshop walls, floor and ceiling were as follows:

- Concrete floor – 1800m^2
- Concrete block walls – 1260m^2
- Metal clad ceiling – 1850m^2

4.3.2. There were few absorptive materials in the room and its total volume was approximately 12600m^3.

4.4. Instrumentation

4.4.1. The following instrumentation was used during the survey:

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Type	Manufacturer	Product Code	Serial Number
Sound Level Meter	Norsonic	Nor118	31477
Calibrator	Norsonic	Nor1251	33376

Table 2 – Layout of the test environment

4.4.2. In addition a sound level monitoring position was placed inside the AHU to monitor the steady state sound pressure level for the duration of the test. The details are tabulated below:

Type	Manufacturer	Product Code	Serial Number
Sound Level Meter	Norsonic	Nor140	1405533
Calibrator	Norsonic	Nor1251	33767

Table 3 – Layout of the test environment

4.4.3. The equipment was calibrated before and after the surveys. The calibration was as follows:

	Before	After
Norsonic 118	113.9	113.9
Norsonic 140	113.9	113.9

Table 4 – Calibration details

4.5. Survey Method

4.5.1. Sound pressure measurements were made at discrete points on a defined measurement surface as prescribed by BS EN ISO 3746:2009.

4.5.2. An imaginary reference box was considered, a parallelepiped encompassing all parts of the machinery. Because of the shape of the air handling unit, the reference box was defined by the machine itself.

4.5.3. The measurement surface was taken to be a parallelepiped with its sides 1m from the AHU. Sound pressure level measurements were made at the positions defined by the procedure in Annex C of BS EN ISO 3746: 2009 of each side of this measurement surface (7 in total). Figure 2 shows the reference box/machine, measurement surface, and the discrete measurement positions.

4.5.4. The acoustics of the measurement environment were taken into account by measuring the reverberation time of the room. Measurements were made according the survey grade method contained in ISO 3382-2:2008 *"Acoustics – measurement of room acoustic parameters – Part 2: Reverberation time in ordinary rooms".*

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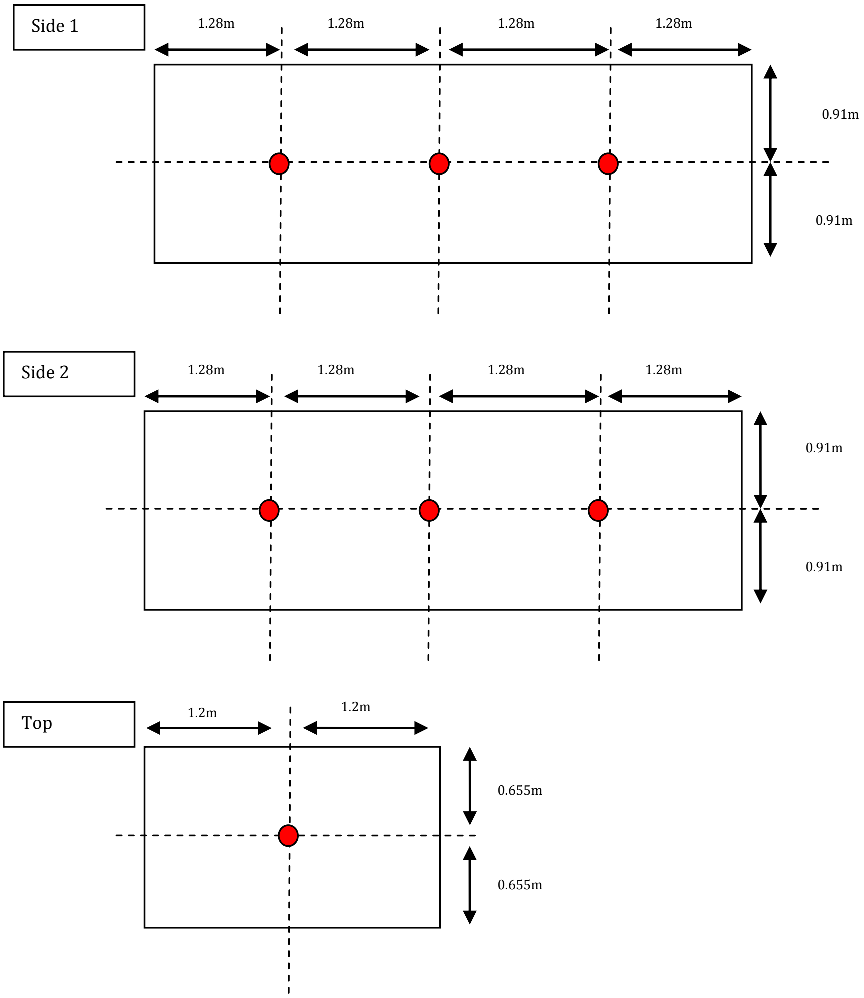

Figure 2 – An illustration showing the reference box, measurement surface and measurement positions for the survey.

4.6. Measurement timescale

4.6.1. The continuous equivalent sound pressure level was measured for 1-min at each measurement location. The following measurements were recorded:

$L_{Aeq,30\ sec}$

4.7. Internal Measurement

4.7.1. The continuous equivalent sound pressure level was measured inside the AHU over 10 second intervals. The data shows the continuous sound pressure level inside the AHU was steady over the survey period. The minimum sound pressure level was 99.7dB LAeq,T and the maximum during the survey period was 100.6dB LAeq,T. The "A" Weighted Sound Pressure Level is graphically illustrated below:

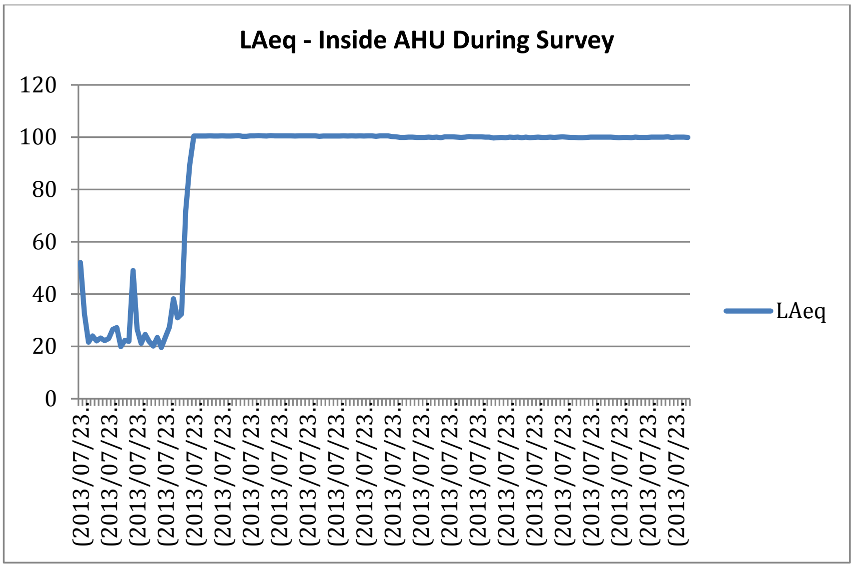

Figure 3: Sound Pressure Levels Lp internal to ANU Fan Section

4.8. Sound pressure level at 1m, SPL

4.8.1. The average sound pressure level in the j[th] frequency band 1m from the air handling unit was calculated by averaging the sound pressure level measurements at positions 1-7 using:

$$\overline{L_{p_j}} = 10\,\mathrm{Log}_{10}\left[\frac{1}{N}\sum_{i=1}^{N}10^{L_{p\,ji}/10}\right]$$

4.8.2. Where:

$\overline{L_{p_j}}$ is the average sound pressure level in the j[th] frequency band;

$L_{p_{j,i}}$ is the sound pressure level in the j[th] frequency band at the i[th] measurement position;

4.8.3. The average corrected sound pressure level for the j[th] frequency band is given by the following:

$$\overline{L_{pf_j}} = \overline{L_{p_j}} - K_1 - K_2$$

K_1 is the background correction for background noise, see appendix for details;

K_2 is the correction due to the reverberant field, see appendix for details.

4.9. Sound power level, SWL

4.9.1. The sound power level was calculated from the sound pressure level measurements made over the measurement surface defined in Figure 2. The sound power level is defined as:

$$L_{Wj} = \overline{L_{pf_j}} + 10\log_{10}\frac{S}{S_0}\,dB$$

4.9.2. Where:

L_{wj} is the sound power level in the j[th] frequency band

S is the area of the measurement surface

S_0 is 1m

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5. Results

5.1. Sound pressure level at 1m, SPL

5.1.1. The sound pressure levels at 1m have been corrected for both background noise and the reverberant field such that they represent the free-field condition. NOTE: dB(A) figure taken from sound level meter record i.e. not calculated from spectrum.

	dB(A)	Octave band centre frequency, Hz								
		31.5	63	125	250	500	1k	2k	4k	8k
SPL (dB)	49.7	60.0	56.3	49.9	52.8	44.8	43.5	42.5	39.0	31.2

Table 5 – Sound pressure level results – Free field

5.2. Sound power level, SWL

5.2.1. The sound power levels have been corrected for both background noise and the reverberant field. NOTE: dB(A) figure taken from sound level meter record i.e. not calculated from spectrum.

	dB(A)	Octave band centre frequency, Hz								
		31.5	63	125	250	500	1k	2k	4k	8k
SWL (dB)	68.9	79.3	75.6	69.1	72.1	64.1	62.8	61.8	58.3	50.5

Table 6 – Sound power level results

6. Conclusions

6.1.1. The sound emissions from the Bodleian Library AHU have been measured and evaluated in accordance with BS EN ISO3746: 2009 such that the following quantities have been calculated:

- Sound power level;
- Sound pressure level at 1 m distance, free field condition.

Dr Bill Whitfield **BA, MSc, PhD, MIOA**
Noise and Vibration Consultant

7. Bibliography

Reference	Title
[redacted]	
BS EN ISO 3746:2009	*"Acoustics — Determination of sound power levels of noise sources using sound pressure — Survey method using an enveloping measurement surface over a reflecting plane".*
ISO 3382-2:2008	*"Acoustics – measurement of room acoustic parameters – Part 2: Reverberation time in ordinary rooms".*

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8. Appendix

8.1. Correction due to background noise

8.1.1. The measured sound pressure levels may need to be corrected for any background noise that was present during the survey. The background correction is determined using the following equation:

$$K_1 = -10 \log\left(1 - 10^{-0.1\left(\overline{L_{pA}} - \overline{L'_{pA}}\right)}\right)$$

8.1.2. Where:

K_1 is the correction for background noise

$\overline{L_{pA}}$ is the average A-weighted sound pressure level

$\overline{L'_{pA}}$ is the average A-weighted background noise sound pressure level

Average A-weighted sound pressure level, $\overline{L_{pA}}$	51.2dB
Average A-weighted background noise sound pressure level, $\overline{L'_{pA}}$	36.7dB
Correction for background noise, K_1	$\boxed{\text{0dB}}$

(Background noise is sufficiently low to be neglected)

8.2. Reverberant field correction term

8.2.1. The sound pressure level measurements are to be corrected for the effects of the reverberant field created by the surfaces of the test room. The correction for the effects of these reflections is given by the reverberation method from ISO 3746:1995:

$$K_2 = 10\,Log_{10}\left[1 + 4\left(S/A\right)\right] dB$$

8.2.2. Where:

A is the equivalent sound absorption area of the room at 1kHz, in square metres;

S is the area of the measurement surface, in square metres.

8.2.3. The equivalent sound absorption area, A, is calculated using the following equation (ISO 3746:1995):

$$A = 0.16\left(V/T\right)$$

8.2.4. Where:

V is the volume of the test room.

T is the reverberation time at 1kHz measured to ISO 3382:2008.

Measured reverberation time	2.605s
Total volume of test room	12600m^3
Area of the measurement surface, S	84.8m^2
Reverberant field correction, K_2	$\boxed{1.58\text{dB}}$

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8.3. Noise Data Sheet

Sound Source		
Type		
Technical Data		
Dims	5.15mL x 1.31m W x 1.82m H	
Operating Conditions	Supply	Extract
Air Vol m3/s	1.3	1.28
Static Pressure	250pa	250pa
Test Environment	Vol m3 (approx)	
Factory Floor	12600	