INVESTIGATION INTO THE SOUND POWER LEVEL REDUCTION ACHIEVED USING ACOUSTIC JACKETS ON A CENTRIFUGAL FAN AND AN AXIAL FAN

Mark SIMMS

VENTAC GROUP, Fitzwilliam House, Blessington, County Wicklow, Republic of Ireland

SUMMARY

This project was assessing the insertion loss of acoustic jackets for fans. The project looks at both a centrifugal fan and an axial fan. The insertion losses are calculated using sound power level measurements before and after the jackets are mounted on the fans. The sound power level was measured using sound intensity measurements according to ISO Standard 9614. From the sound intensity measurements noise maps were generated. Locations radiating high levels of sound energy (or hot spots) could be focused on. This was used in the case of the centrifugal fan to improve the design.

INTRODUCTION

Fan jackets (Figure 1) are a very common way of reducing noise from fans but the noise reduction achieved with this method can be limited by the operational requirements of the noise source. This project was to see how efficient they were at reducing the noise radiated from fans and if there were any opportunities to improve the performance.

The jackets used were of a typical acoustic jacket construction. They had a polymeric barrier (5 kg/m^2) sandwiched between 22 kg /m^3 fibreglass with an impermeable fibreglass facing. The jackets were mounted in a typical fashion using velcro strapping. The ducting was left untreated and was a key route for flanking noise that would exist in real installations.

The centrifugal fan was a direct driven, 2 pole, 3 kW, high pressure fan and the axial fan was a 0,55 kW cylindrical cased axial fan

Figure 1: Cross section of acoustic jacket material (top left), example of an acoustic jacket on an axial fan (top right) and example of an acoustic jacket on centrifugal fan (bottom)

METHODOLOGY

The sound power level was tested using sound intensity to the standard:

ISO 9614-1 Acoustics – Determination of sound power levels of noise sources using sound intensity. Part 1 Measurement at Discrete points

The fan was mounted on a hard acoustically reflective floor. A parallel piped measurement grid was set up around the fan. The sound intensity is measured at 60 locations across the grid. The measurements were taken using a G.R.A.S. 40-AI intensity probe with paired microphones divided by a 12 mm spacer.

The measurement grid had the dimensions of 1.6 m wide by 1.6 m long by 1.2 m high. The measurement locations on the grid were centred on grid squares 400 mm by 400 mm.

Prior to measuring the sound intensity two checks are measured on the source. The *stationnarity check* ensures that the noise source is source is steady and non-fluctuating. The *calibration check* ensures the directivity of the measurements is consistent in both positive and negative directions.

The sound power level of the source tested is determined from the 60 measurements. The sound intensity at each measurement location is multiplied by the area of the grid square to give the *partial sound power*. *Sound power* of the source is the sum of the *partial sound power* results.

The *sound power level* of the source is determined by the formula:

$$L_W = 10\lg[\ |P|/P_0]\ \text{dB}$$

L_W sound power level

$|P|$ is the magnitude of the sound power of the source

P_0 is the reference sound power (= 10^{-12} W)

The A-weighting values specified in ISO 3744 are applied in each third octave frequency band from 50 Hz to 6.3 kHz to determine the A-weighted sound power level of the source.

The noise maps are generated by displaying the partial sound pressure levels at each location in the grid in a graphical format with areas of increased sound power indicated by bright red. The map shows the five faces of the grid in the 2 dimensions of an un-folded box as outlined in Figure 2.

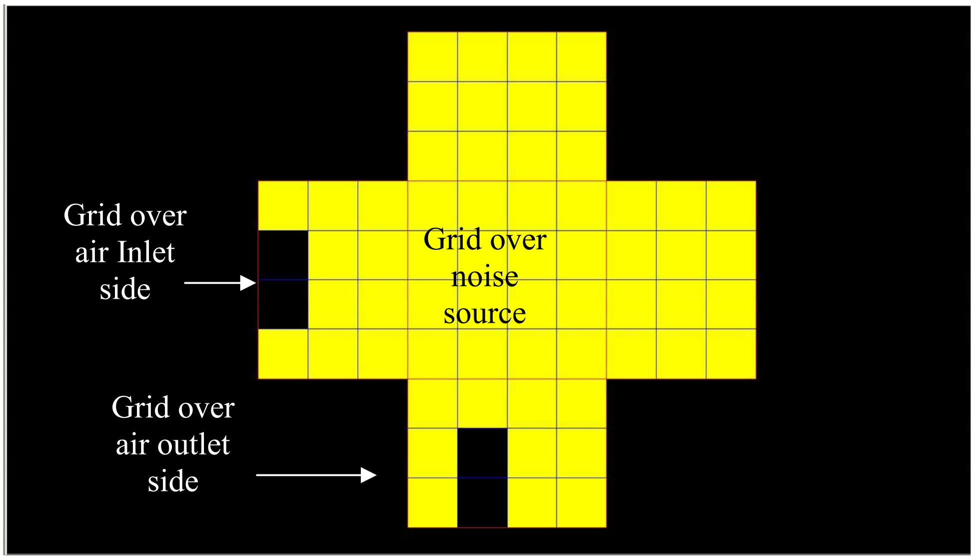

Figure 2: Intensity mesh plot indicating the relative positions of measurement grids to centrifugal fan noise source

RESULTS

Insertion Loss of Acoustic Jacket over Axial fan

The A-weighted sound power level of the cylindrical cased axial fan was tested before and after the fan was wrapped in an acoustic jacket (Figure 3). The acoustic jacket extended to cover the flexible connections of the fan casing to the ductwork. The test fan is a two pole cased axial 0.55 kW fan, 355mm in diameter.

Figure 3: Tested axial fan without and with acoustic jacket; grid indicates locations of the intensity probe

Table 1: Insertion Loss of Acoustic Jacket over Axial fan

Parameter	Axial Fan	Axial Fan + Jacket	Insertion Loss
A-weighted Sound Power Level L_{WA} dB(A)	75.2	71.3	3.9

Axial Fan results

This jacket represents the simplest arrangement where the whole fan can be wrapped and there is no requirement for openings for cooling air due to the inherent nature of the machine. There is the noise flanking pathway of the thin duct walls. The ducting is connected by flexible connectors to the fan to prevent structure-borne noise travelling along it. The acoustic jacket is designed to overlap the flexible connecters as they are regarded as a potential area for airborne noise radiation.

The frequency analysis (Figure 4) suggests the jacket performed well in the mid to high frequency bands but also achieved a 3.1 decibel reduction at 200Hz which is the same band as the blade pass frequency. The blade pass frequency is responsible for the tonal element of fan noise. Tonal noise typically causes significantly more disturbance than broadband noise. Reducing the peak at this frequency was important not only reducing the overall sound power level of the fan but also in mitigating the impact of the noise.

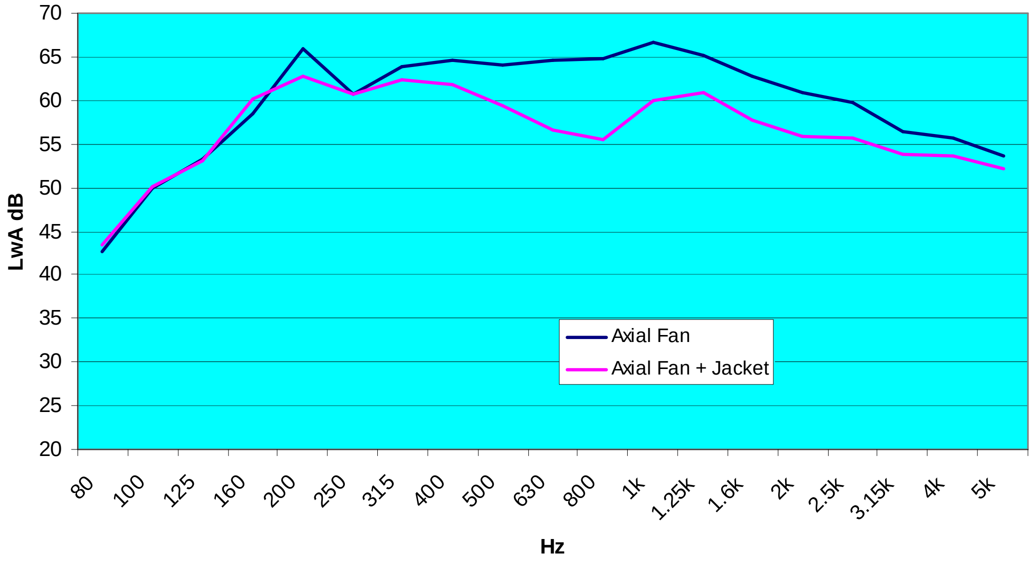

Figure 4: Third octave frequency analysis of sound power level of axial fan before and after an acoustic jacket was wrapped around the fan.

Insertion Loss of Acoustic Jacket over the Scroll Case of the Centrifugal Fan

The A-weighted sound power level of the centrifugal fan was tested before and after the scroll casing of the fan was wrapped in an acoustic jacket (Figure 5). The test fan is a 2 pole direct drive 3 kW centrifugal fan.

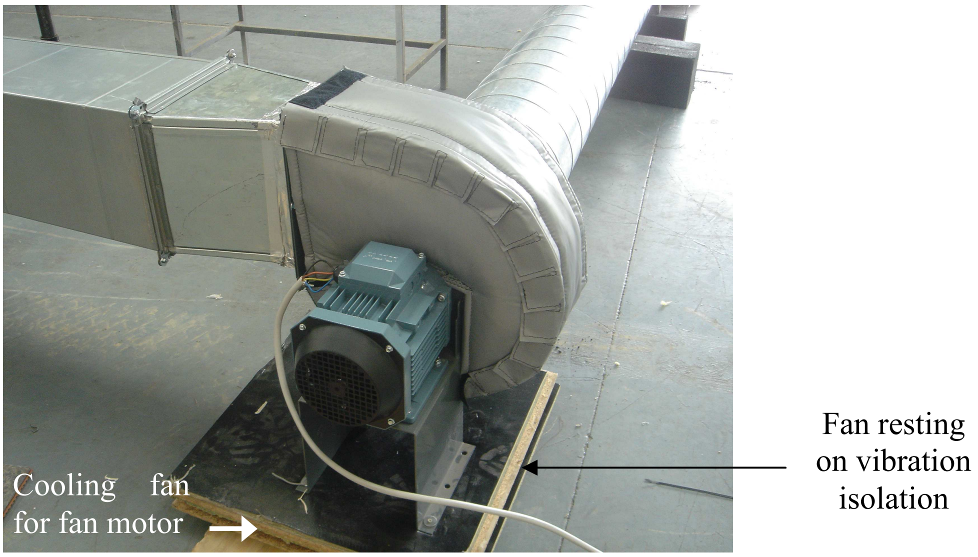

Figure 5: Acoustic jacket on scroll casing of the fan

Table 2: Insertion Loss of Acoustic Jacket on Scroll Case of the Centrifugal Fan

Parameter	Centrifugal Fan	Centrifugal Fan+ Jacket on fan scroll	Insertion Loss
A-weighted Sound Power Level L_{WA} dB(A)	82.8	80.4	2.4

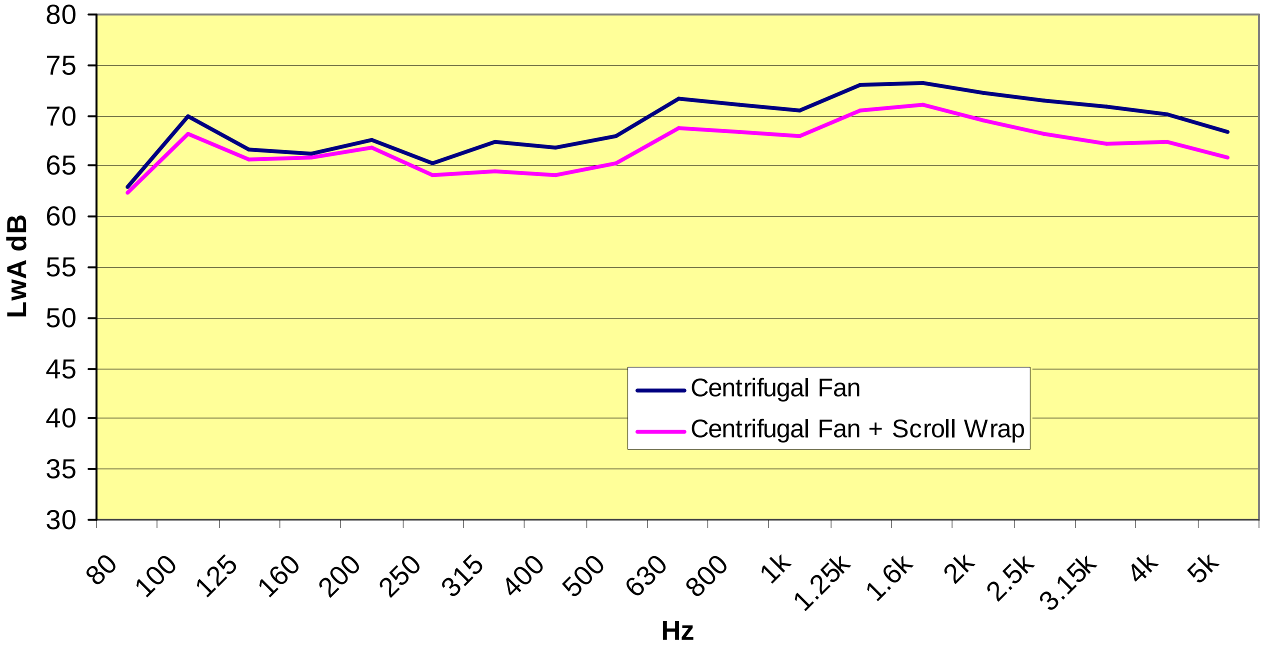

Figure 6: Third octave frequency analysis of sound power level of centrifugal fan before and after an acoustic jacket was wrapped around the scroll of fan

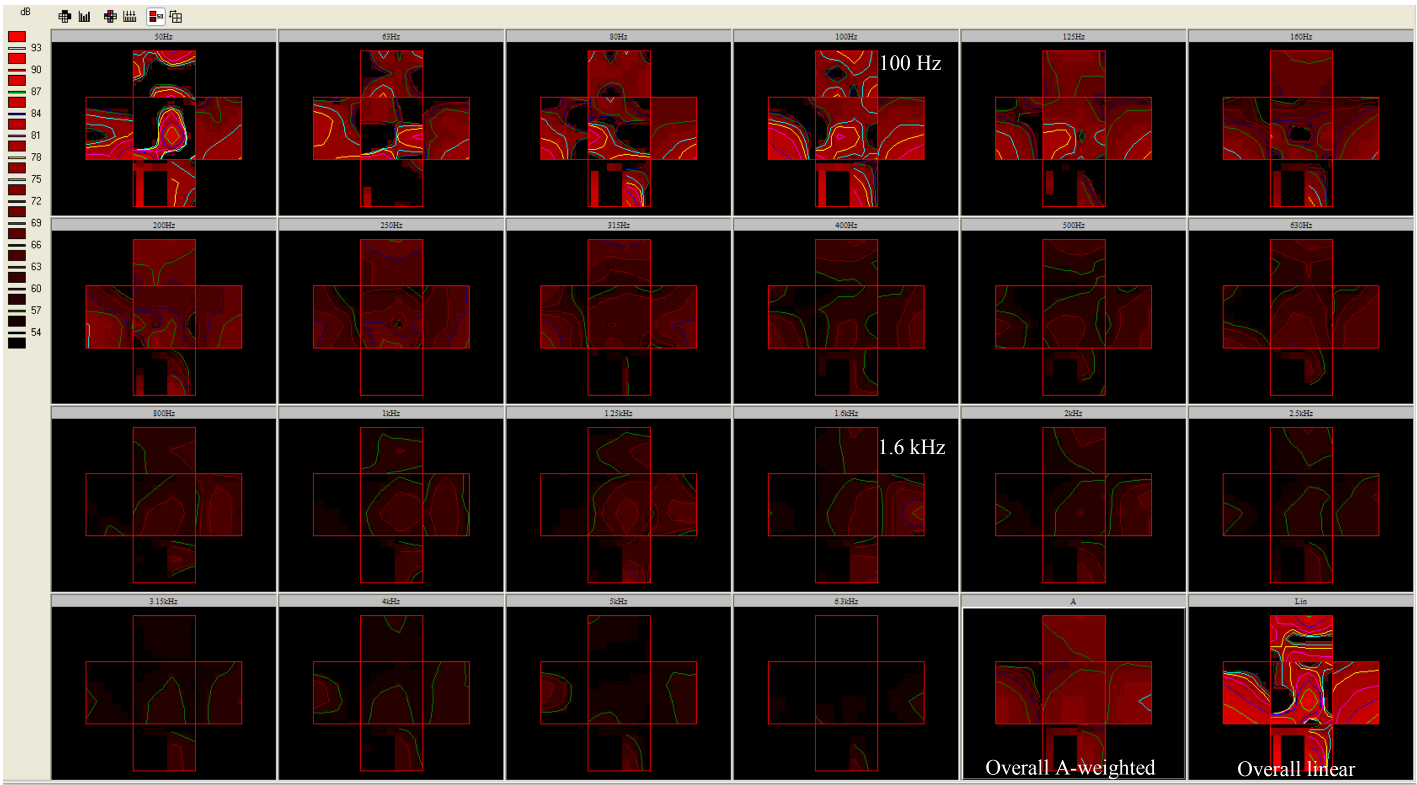

Figure 7: Noise map of centrifugal fan with wrap on scroll casing (linear weighting)

Centrifugal fan with acoustic jacket on scroll case results

The high frequency bands are much more important on the centrifugal fan than was seen on the axial fan in the test above (Figure 6). From the noise map above (Figure 7) the high frequency noise was found to be centred on the fan motor and its cooling fan. This can be seen by the 'hot spots' in this area, both on the overall A-weighted level noise map and particularly on the noise map of the 1.6 kHz third octave band.

The centrifugal fan is more complex than the axial fan which is entirely enclosed by the fan casing and ductwork. The higher frequency of noise the easier it is to acoustically insulate with mass therefore the cooling impeller on the fan motor was determined to be the source of the high frequency noise because this noise source is exterior to the fan casing. This was supported by subjective observation of the test technician. This finding suggested that the only way to reduce the sound power level of the fan further was to treat the fan motor

Insertion loss of acoustic jacket over the scroll case & fan motor of the centrifugal fan

The A-weighted sound power level of the centrifugal fan was tested before and after the scroll casing of the fan was wrapped in an acoustic jacket.

Figure 8: Fan motor jacket

The jacket is designed to allow cooling air to flow over the radiator fins of the motor (Figure 8). The jacket supports itself from the top of the fan motor but it extends horizontally out beyond the motor and its cooling impeller. This creates space for the air to enter the inlet grill of the cooling impeller. There is an air inlet opening at the base of the jacket allowing the cooling air flow in through two 90^{o} bends. The bends maximise the attenuation achieved by the sound absorptive properties of the jacket. The air passes over the cooling fins of the motor before exiting through a vent on both sides of the motor.

Table 3: Insertion Loss of Acoustic Jacket on the Scroll Case and the Fan Motor of the Centrifugal fan

Parameter	Centrifugal Fan	Centrifugal Fan+ Jacket on fan scroll	Centrifugal Fan+ Jacket on fan scroll +Jacket on motor fan	Insertion Loss
A-weighted Sound Power Level L_{WA} dB(A)	82.8	80.4	78.6	4.2

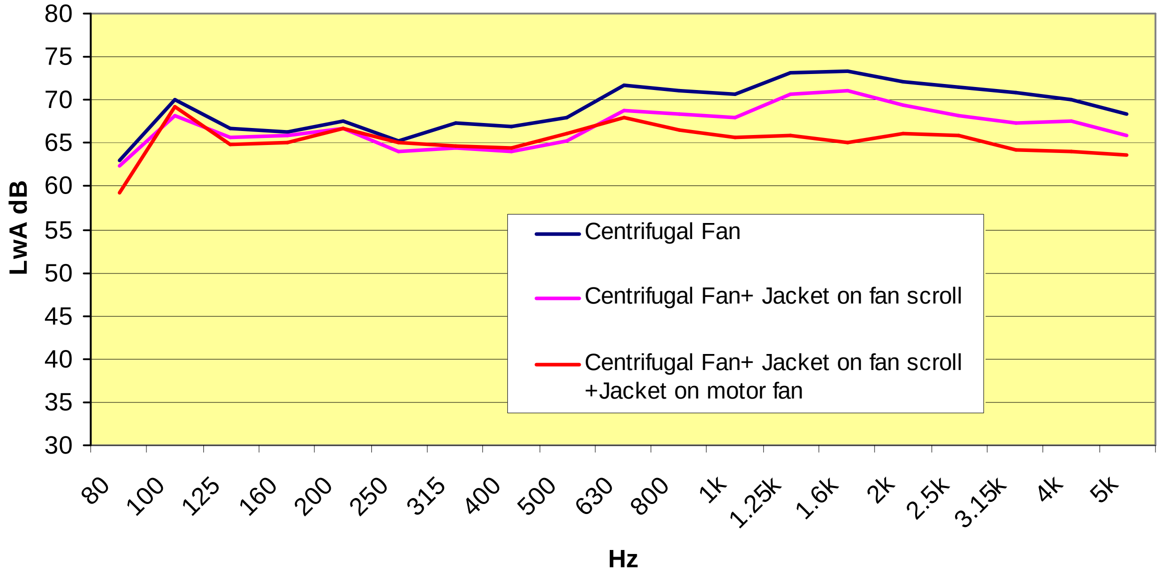

Figure 9: Third octave frequency analysis of sound power level of centrifugal fan before and after the different acoustic jacket options were fitted

Centrifugal fan with acoustic jacket on scroll case and fan motor results

The reductions achieved by the addition of the jacket over the motor are clear to see in the frequency bands from 800Hz and higher. The reduction in the noise levels in the mid to high frequencies means that these frequencies are no longer the highest in the spectrum.

The low frequency peak at 100Hz becomes dominant of the spectrum at around 70 decibels including A-weighting. This means it is difficult to achieve further reductions without reducing this peak. An acoustic jacket with an area mass of 5kgm^{-2} is ineffective at 100Hz due the mass law, which states that the higher the frequency of the sound trying to penetrate the material, the better the material is at attenuating the sound.

This means that the barrier would have to have significantly more mass or stiffness to be an effective insulator in these frequencies and a rigid enclosure may be more practical in these circumstances.

DISCUSSION

This project assessed how effective an acoustic jacket is at insulating typical industrial fan configurations. The acoustic jacket is one of the simplest solutions for noise treatment of industrial machinery but its effectiveness is limited by the following factors.

The close wrapped nature of the jacket means that the isolation of the acoustic barrier from the noise source is much less than is possible with, for example, an acoustic enclosure where the barrier is physically separated from the noise source. The good operation of a mechanical noise source typically requires cooling air to flow across it to prevent over heating. In the case of fans; the service duct work they are a part of is a flanking pathway that the noise can use to by-pass the acoustic jacket.

The axial fan represents the simplest design for an acoustic jacket as all the noise source elements of the fan are enclosed by the fan casing within the ducting. The amount of reduction that can be achieved on the fan will be quickly limited by the noise flanking through the attached ducting. In the case of this testing the ducting was lightweight spiral that is in wide spread use and has low sound insulation properties.

The centrifugal fan is a more complex noise source than the axial fan because the motor is outside the ductwork. In practise the use of an acoustic jacket is often solely on the scroll of the fan. This was found to give less than a 3 decibel reduction which is what is generally considered to be required for an audible reduction in noise. To improve the performance of the fan the sound power measurement data was examined in the form of a noise map. The cooling fan on the motor itself was found to be an important noise source.

To treat this, the jacket enclosing this area was manufactured to extend past the cooling fan to create an air space within the jacket down to an inlet vent at the base of the jacket. The cooling air leaves the jacket through passive vents adjacent to the cooling fins of the motor. Both the air inlet and air outlet force the air through 90^o bends to mitigate the noise energy escaping through these areas.

The limiting factor in the reductions achievable on this fan was the low frequency noise which acoustic jackets are insufficient to treat. Noise is proportionately harder to treat the lower the frequency is so significantly more mass and a separating airspace is required to effectively treat 100Hz.

This project was designed to generate practical data for treating noise problems in real life situations. During the test work it was found that the reductions achieved by simply wrapping the scroll casing of a centrifugal fan was not as high as might be expecting for such a widely used treatment. The project then developed an improved design and established its performance.

APPENDICES

Schedule of test equipment

- Black Solo Type 1 Integrated Sound Level Meter, manufactured by 01dB-Metravib, serial number 65098
- Preamplifier PRE21S, serial number 15495
- Microphone MCE212, serial number 110019
- Sound Calibrator Type 1251, manufactured by Norsonic, serial number 20803
- GRAS Intensity Probe 50AI-B, serial number 38196
- GRAS Intensity Microphone 40AI, serial number 71134
- GRAS Intensity Microphone 40AI, serial number 71137
- GRAS Intensity Preamplifier 26AA, serial number 37803
- GRAS Intensity Preamplifier 26AA, serial number 37804
- 01DB Harmonie Multi Channel Analyser, serial number 04299

Fan Details

Centrifugal Fan

Code: CMT/2-280/115-3kW
Manufacturer: S&P
Motor: 3 kW
Type: Forward curved centrifugal fan 400/3/50Hz

Axial Fan

Code: TCBB/2-355/H
Manufacturer: S&P
Motor: 0.55 kW
Type: Cylindrical cased axial fan

Acoustic Performance of Acoustic Jacket Material

SOUND TRANSMISSION LOSS – REPORT REFERENCE 1295

VINYL FACING, 22KG FIBREGLASS, 5KG POLYMERIC, 22KG FIBREGLASS, VINYL FACING

Measurements

Sound transmission loss measurements were conducted in the reverberation chambers of the Ventac Laboratories on the 15th of August 2007.

Specimen Description - Brief

Volume of test room (m^3): 26
Type of noise used: White Noise
Area of tested specimen (m^2): 1.0
Temperature of test room (°C): 17

Results

Frequency f, (Hz)	R (third octave), dB
100	8.4
125	14.1
160	7.5
200	13.7
250	12.6
315	16.2
400	18.8
500	18.9
630	18.4
800	22.2
1000	27.7
1250	35.8
1600	41.1
2000	40.5
2500	38.2
3150	35.3
4000	36.8
5000	36.6
Rw (C; Ctr) (dB)	24 (-1; -5)