

## Noise control calculations for an office air conditioning system

The purpose of this appendix is to show the basic calculation procedure used in determining the sound pressure level in octave bands from an office air conditioning system such as the one shown in Figure 8.6 in Chapter 8. The reader is referred to Reference 8.4, Chapters 4 and 41 through 43 and to ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) handbooks for more detailed information. Much of the data in this chapter is from tables provided in Reference 8.4, Chapter 42. Note that some manufacturers of heating, ventilation and air conditioning (HVAC) equipment have dedicated air conditioning system noise calculators available on the internet.

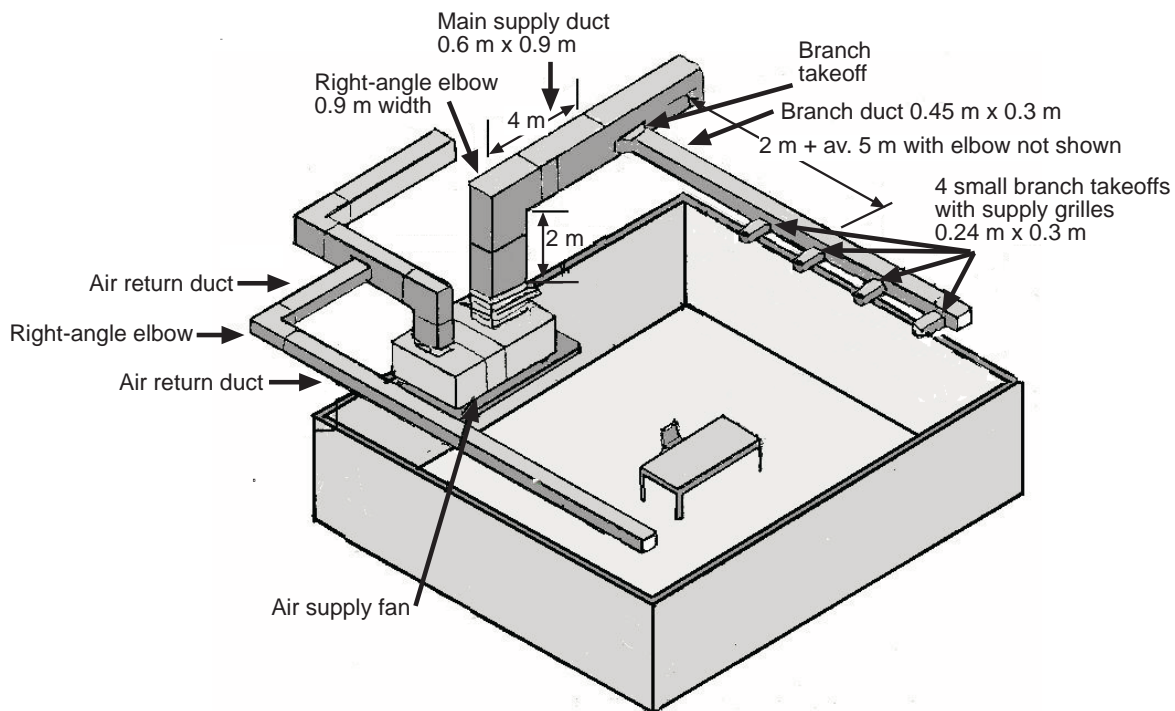


Figure A2.1. A typical office air conditioning system (somewhat simplified version of Figure 8.6 in book). Drawing contains anomalies and is not to scale. Only intended to show general layout of the system.

The office floor dimensions are assumed to be 10 m by 10 m and the ceiling height as 3 m. The ceiling is covered by sound absorptive tiles.

The noise level in the office is calculated from knowledge of the sound power radiated into the office. To find this noise power one must first know the noise power that is generated by the supply system. Noise power in an air conditioning system is generated 1) at the fans, 2) in the canals, and 3) at the exhausts. The noise power is reduced 4) by canal branching and 5) by sound reflection at exhausts.

The noise spectrum generated by fans and duct air flow is dominated by low frequencies while the spectrum of the noise generated by flow at the exhaust grilles is dominated by

mid and high frequencies. This noise is critically dependent on the flow velocity since the noise generated will be proportional to the flow velocity raised to the power of 4 to 8. For low flow velocities at the grill (less than 5 m/s) there is usually little flow noise produced. Intakes usually do not generate noise in the room due to turbulence by the grilles since the turbulence generated by the inflow will be mainly on the inside of the duct.

To calculate the resulting noise in the office one must first obtain the appropriate fan noise data from the fan manufacturer's data sheet. That data should provide octave band power levels in the frequency range of 63 Hz to 4 kHz. The acoustic power traveling along the ventilation duct is reduced by the various attenuation and reflection components. If noise is generated in the ducts, then that noise must also be estimated and appropriately entered as a sound source component.

The following text and tables give the attenuation values due to the various mechanisms previously listed. The results are then entered (as negative numbers) into Table A2.1, which gives the total power level at the outlets. The resulting room noise sound pressure levels are then calculated using a relationship between sound power level and sound pressure level.

Below is an example of how the data in Table A2.1 is calculated. The air supply system shown in Figure A2.1 consists of

- 1) Air supply fan, 25 m<sup>3</sup>/s and pressure 375 Pascal
- 2) 2 m of unlined duct 0.6 m by 0.9 m, little attenuation
- 3) Elbow without turning vanes, 90° bend, 0.9 m wide
- 4) 4 m of lined duct 0.6 m by 0.9 m
- 5) 90° branch to a lined branch duct, 0.45 m by 0.3 m
- 6) 2 m + 5 m of lined branch duct, 0.45 m by 0.3 m that includes an unlined elbow not shown in Figure A2.1
- 7) 4 small branch takeoffs at about 2 m inter-distance, 0.24 m by 0.3 m
- 8) Supply grilles installed where each small branch duct takeoff meets room

Even if the ductwork does not contain any sound attenuators (acoustical linings or sound traps), only a fraction of the acoustical energy generated by the fan actually reaches any one room because of energy division at branch takeoffs, energy losses due to duct wall vibration, and wave reflections at elbows and at the diffuser and grille terminals. To avoid expenditures for unnecessary over-treatment of ductwork, credit should be taken for these attenuations.

**Table A2.1 Line 1: Fan power level**

The fan’s sound power level  $PWL (L_W)$  will depend on the air volume flow rate and the air pressure. It is best to use the manufacturer’s data. If such data is not available, one can use Table 41.1 in Reference 8.4 that lists typical data for various types of fans. The data in that table must then be adjusted for the actual flow rate as described in the reference. The table below uses data from experience. The 63 Hz octave band was not used in these calculations but should be included in a practical case.

Line	Item	Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
1	Fan $PWL$ , from manufacturer or Ref 8.4, Ch 41, Table 41.1 adjusted for flow rate.	86	82	78	71	68	66

**Table A2.1 Line 2: Attenuation of unlined duct**

The first piece of duct (2 m) is unlined so it gives little noise attenuation. The table below lists typical data :

Ref 8.4, Ch 42, Table 42.2	Perimeter/Area-cross-section (m/m <sup>2</sup> )	Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
Octave-band attenuation in dB/m	Over 12	.9	.3	0	0	0	0
Octave-band attenuation in dB/m	5 - 12	.3	.3	0	0	0	0
Octave-band attenuation in dB/m	Under 5	.3	.3	0	0	0	0

In our case the duct is 0.6 m x 0.9 m, which gives a perimeter of 3 m and a perimeter-to-duct area ratio of about 5.5, so we obtain:

Line	Attenuation [dB] of 2 m supply duct, 0.6 m by 0.9 m, no lining	Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
2		1	1	0	0	0	0

**Table A2.1 Line 3: Attenuation of unlined elbow**

An unlined elbow does not give much noise attenuation. The table below lists typical attenuation data. Other data may be found in Ref 8.4, Ch 42, Table 42.5:

	Duct dimension Width	Attenuation [dB]	Octave-Band Center Frequency [Hz]					
			125	250	500	1 k	2 k	4 k
No lining	0.13 m, 5"	Attenuation [dB]	0	0	1	5	7	5
No lining	0.25 m, 10"	Attenuation [dB]	0	1	5	7	5	3
No lining	0.50 m, 20"	Attenuation [dB]	1	5	7	5	3	3
No lining	1.00 m, 40"	Attenuation [dB]	5	7	5	3	3	3

With lining the attenuation is higher. Lining is usually applied as shown in Figure A2.2 below

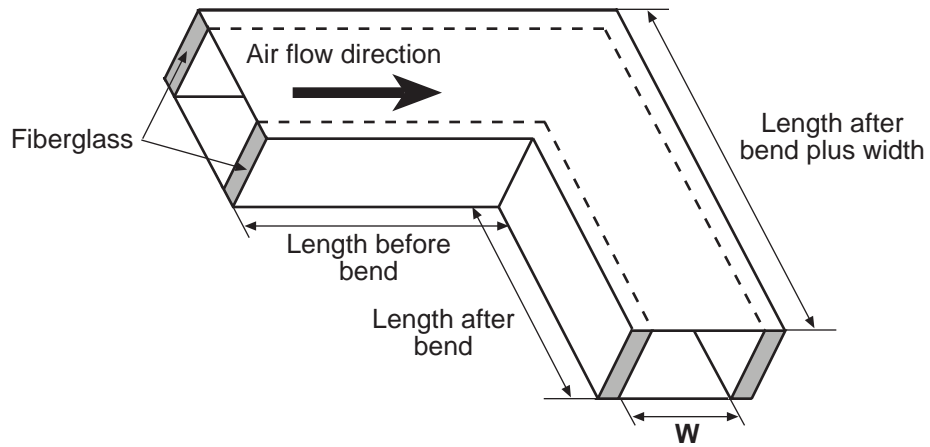


Figure A2.2. Duct elbow definitions for table below.

The table below lists typical attenuation data:

	Duct dimension "W"	Attenuation [dB]	Octave-Band Center Frequency [Hz]					
			125	250	500	1 k	2 k	4 k
Width "W" is defined in figure above								
Lining before & after	0.13 m, 5"		0	0	1	6	12	14
Lining before & after	0.25 m, 10"		0	1	6	12	14	16
Lining before & after	0.50 m, 20"		1	6	12	14	16	18
Lining before & after	1.00 m, 40"		6	12	14	16	18	18

The elbow in Figure A2.1 is not lined, using the closest dimensions, we obtain for Table A2.1 Line 3 the following values:

Line	Attenuation [dB] of elbow, no lining	Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
3		5	7	5	3	3	3

**Table A2.1 Line 4: Attenuation of unlined duct**

Using the data given above for Table A2.1 Line 2 we obtain for the 4 m of unlined duct 0.6 m by 0.9 m:

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
4	Attenuation [dB] of 4m supply duct, 0.6 m by 0.9 m, no lining	1	1	0	0	0	0

**Table A2.1 Line 5: Attenuation of branch takeoff**

The noise reduction by branch duct takeoffs depends on the geometry shown in this figure:

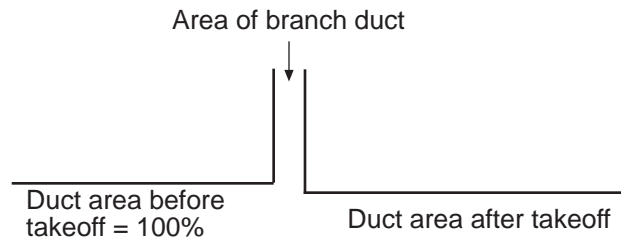


Figure A2.3. Duct branch area definition for table below.

The table below shows the attenuation obtained for various area ratios (Ref 8.4, Ch 42, Table 42.4).

Percent area of branch takeoff duct to main duct before takeoff	5 %	10 %	15 %	20 %	30 %	40 %	50 %	60 %
Attenuation [dB] to be subtracted from PWL before takeoff	13	10	8	7	5	4	3	2

In our case the branch is 0.45 m x 0.3 m, the takeoff area ratio is 25% so using interpolation we obtain the following dB-values:

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
5	Branch takeoff, 25% of the area. Attenuation [dB]	6	6	6	6	6	6

**Table A2.1 Line 6: Attenuation of unlined duct**

Using the data given in the section dealing with Table A2.1 Line 2 we obtain for the 2 m of unlined duct 0.45 m by 0.3 m that has a perimeter-to-duct area ratio of about 11:

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
6	Attenuation [dB] of 6 m branch duct, 0.45 m by 0.3 m, no lining	1	1	0	0	0	0

**Table A2.1 Line 7: Attenuation of unlined elbow**

This unlined elbow is not shown in Figure A2.1, using the closest dimensions, we obtain for Table A2.1 Line 7 the following values:

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
7	Attenuation [dB] of elbow, no lining	1	5	7	5	3	3

**Table A2.1 Line 8: Attenuation of unlined duct**

Using the data given in the section dealing with Table A2.1 Line 2 we obtain for the 5 m of unlined duct 0.45 m by 0.3 m that has a perimeter-to-duct area ratio of about 11. (Of course the duct length to each takeoff varies, but we compromise and use an average.)

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
8	Attenuation [dB] of 5 m branch duct, 0.45 m by 0.3 m, no lining	2	2	0	0	0	0

**Table A2.1 Line 9: Attenuation of branch takeoffs**

Additional small branch takeoffs lead to further attenuation. Using the data given in the section dealing with Table A2.1 Line 5, with a branch takeoff area ratio of 25% we obtain the following approximate dB-values to deduct for the four outlets.

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
9	Attenuation [dB] due to final branch air division to four diffusers, 25% of the noise power goes to each.	6	6	6	6	6	6

**Table A2.1 Line 10: Attenuation due to reflections at point where air enters the room**

Next the reflection loss is calculated. The table below from Ref 8.4, Ch 42, Table 42.6 shows the reflection loss values in dB as a function of frequency for sound arriving at the duct opening to the room. They apply to a duct terminating in a grille or diffuser in ceiling or wall, away from other duct terminations.

Round Duct	Square Duct	Octave-Band Center Frequency [Hz]					
Duct Diameter	Dimensions	125	250	500	1 k	2 k	4 k
0.15 m	0.023 m <sup>2</sup>	12	8	4	1	0	0
0.30 m	0.09 m <sup>2</sup>	8	4	1	0	0	0
0.60 m	0.36 m <sup>2</sup>	4	1	0	0	0	0
1.20 m	1.4 m <sup>2</sup>	1	0	0	0	0	0

In our case the rectangular duct openings have each an area of about 0.08 m<sup>2</sup> so using the next available value we obtain the following dB-values:

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
10	Attenuation [dB] due to grille end reflection, 0.24 m by 0.3.	8	4	1	0	0	0

**Table A2.1: Summary of data**

Line	Item	Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
1	Fan PWL, from manufacturer or Ref 8.4, Ch 41, Table 41.1 [dB]	86	82	78	71	68	66
2	Attenuation of 2 m supply duct, 0.6 m by 0.9 m, no lining [dB]	-1	-1	0	0	0	0
3	Attenuation of elbow, no lining [dB]	-5	-7	-5	-3	-3	-3
4	Attenuation of 4 m supply duct, 0.6 m by 0.9 m, no lining [dB]	-1	-1	0	0	0	0
5	Final branch air division to four grilles, 25% of the noise power goes to each [dB]	-6	-6	-6	-6	-6	-6
6	Attenuation of 2 m branch duct, 0.45 m by 0.3 m, no lining [dB]	-1	-1	0	0	0	0
7	Attenuation of elbow, no lining [dB]	-1	-5	-7	-5	-3	-3
8	Attenuation of 5 m branch duct, 0.45 m by 0.3 m, no lining [dB]	-2	-2	0	0	0	0
9	Final branch air division to four grilles, 25% of the noise power goes to each [dB]	-6	-6	-6	-6	-6	-6
10	Grille end reflection, 0.24 m by 0.3 [dB]	-8	-4	-1	0	0	0
11	Sound Power Level, PWL, at one grille from above [dB]	55	53	54	51	50	48

## Calculation of room sound pressure levels

Since the room has most of its sound absorption in the ceiling due to the ceiling tiles the reverberant sound field in the room will not be as diffuse as if the sound absorption would have been due to a more evenly placed absorption. This will complicate the calculation of the octave band sound pressure levels from the octave band sound power levels in Table A2.1. Various methods are available to handle such cases. An empirical equation can be found in Reference 8.4, Chapter 4 (Equation 4.10a) that can possibly be used under certain conditions, but we will first demonstrate the use of the classical method and assume the reverberant sound field is diffuse.

The reverberation time of a typical modern office with acoustic tile ceiling and typical furnishings is about 0.8 seconds, so the absorption obtained from Sabine's equation on page 35 of the book is 61 metric sabins. The average sound absorption coefficient is 0.19 which is acceptable for the use of Sabine's equation. The diffusivity of the reverberant sound field is reduced by the large sound absorbing ceiling but increased by the presence of the objects in the room.

The direct sound pressure level  $SPLD$  is calculated from (this equation does not appear in the book)

$$SPLD \approx PWL - 11 - 20 \log(R) \text{ [dB]}$$

where  $PWL$  is the sound power level of the source, in this case one of the grille openings, and  $R$  the distance to the source in meters.

The reverberant sound pressure level  $SPLR$  is given by the Equation 3.3 on page 40 in the book, where  $A$  is the absorption area in metric sabins.

$$SPLR \approx PWL - 10 \log(4/A) \text{ [dB]}$$

Let us assume that the workplace head position is in the room center at about 5.5 m from the closest wall mounted grilles and at 6.3 m from the farther grilles. The office worker is reached by both direct and reverberant sound from the various grilles. The reverberation radius is defined as the distance from the sound source at which the sound pressure contributions from the direct and reverberant sound fields are equal, that is  $SPLD = SPLR$ . In this case the reverberation radius is about 1.6 m so if the sound field is diffuse the workplace is in the reverberant field. For reasons of completeness, however, we will calculate the direct sound levels as well.

At a distance of 5.5 m  $SPLD \approx PWL - 26$  dB, and at 6.3 m distance  $SPLD \approx PWL - 27$  dB.

However, when sound sources are on walls or in corners they radiate more power; this is not discussed in the book. The grilles are wall mounted and will each radiate 3 dB more sound than if it ends far from a sound reflecting surface. A corner grille will radiate 6 dB more.



We then obtain the *SPLD* contributions from one of the close grilles as

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
11	Sound Power Level, <i>PWL</i> , at one grille from above [dB]	55	53	54	51	50	48
12	Radiation increase due to wall placement [dB]	+3	+3	+3	+3	+3	+3
13	Distance attenuation [dB]	-26	-26	-26	-26	-26	-26
14	Direct Sound Pressure Level, <i>SPLD</i> , from corner grille [dB]	32	30	31	28	27	25

Similarly we obtain the *SPLD* contributions from the three wall grilles as

		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
15	<i>SPLD</i> , from close wall grille 2 [dB]	32	30	31	28	27	25
16	<i>SPLD</i> , from far wall grille 1 [dB]	31	29	30	27	26	24
17	<i>SPLD</i> , from far wall grille 2 [dB]	31	29	30	27	26	24

Using Figures 1.7 and 1.8 in the book we find that the total level given by the direct sound of the four grilles will be about 6 dB higher than that of the noisiest grille.

		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
18	Total Direct Sound Pressure Level, <i>SPLD</i> , from grilles [dB]	37	35	36	33	32	30

We next need to calculate the octave band reverberant sound pressure levels in the room. We obtain

$$SPLR \approx PWL + 10 \log(4/61) \approx PWL - 12 \text{ [dB]}$$

The same reasoning as earlier about the positioning of the grilles leads to the *SPLR* being

Line		Octave-Band Center Frequency [Hz]					
		125	250	500	1 k	2 k	4 k
11	Sound Power Level, <i>PWL</i> , at one diffuser from above [dB]	55	53	54	51	50	48
19	Radiation increase due to corner placement [dB]	+3	+3	+3	+3	+3	+3
20	Attenuation due to room sound absorption [dB]	-12	-12	-12	-12	-12	-12
21	<i>SPLR</i> , from close wall grille 1 [dB]	46	44	45	42	41	39

Because the reverberant sound ideally is independent of source distance, all grilles contribute equally so that (using Figures 1.7 and 1.8 in the book) we find that the total level given by the reverberant sound of the four grilles will be about 6 dB higher than that of one grille.

		<i>Octave-Band Center Frequency [Hz]</i>					
<i>Line</i>		125	250	500	1 k	2 k	4 k
22	<i>Total Reverberant Sound Pressure Level, SPLR, from grilles [dB]</i>	52	50	51	48	47	45

We notice that the reverberant sound will be dominant as discussed earlier.

		<i>Octave-Band Center Frequency [Hz]</i>					
<i>Line</i>		125	250	500	1 k	2 k	4 k
18	<i>Total Direct Sound Pressure Level, SPLD, from grilles [dB]</i>	37	35	36	33	32	30
22	<i>Total Reverberant Sound Pressure Level, SPLR, from grilles [dB]</i>	52	50	51	48	47	45
23	<i>Total Sound Pressure Level, SPL, from grilles [dB]</i>	52	50	51	48	47	45

Since the reverberant sound pressure level may be overestimated due to the lack of sound field diffusivity the levels in Line 21 will be too high. If we use the empirical equation mentioned earlier we obtain the following table for the total sound pressure level:

		<i>Octave-Band Center Frequency [Hz]</i>					
<i>Line</i>		125	250	500	1 k	2 k	4 k
24	<i>Total Sound Pressure Level, SPL, from all grilles [dB] by Ref 8.4, Ch. 4, Eq. 4.10a</i>	50	47	47	43	41	38

We note that the levels given in Line 24 are much higher than the direct sound levels in Line 18 but not as high as the total sound pressure levels in Line 23 calculated using the classical method.

### Comparison with noise criteria

Let us assume that we want the office noise to be less than NC-35 (cf. Figure 8.1 on page 119 in the book)

<i>Line</i>	<i>Octave-Band Center Frequency [Hz]</i>	125	250	500	1 k	2 k	4 k
23	<i>Total Sound Pressure Level, SPL [dB] from grilles</i>	52	50	51	48	47	45
25	<i>NC-35 Criteria [dB]</i>	52	45	40	36	34	33
26	<i>Needed additional attenuation [dB]</i>	0	5	11	12	13	12

Clearly we need to bring the octave band sound levels down, even if the values in Line 24 were correct we would still need to add extra attenuation. This is best done when installing the air conditioning system by using sound-absorbing internal duct lining.

## **Sound attenuation in ductwork**

Sound-absorbing internal duct lining are blankets and semi-rigid boards. Each manufacturer provides dB/distance coefficients and information regarding strength, longevity, fire resistance and compliance with code requirements, resistance to erosion at high air velocities, and freedom from odor and bacteria growth whether wet or dry. Newer treatments are available for critical areas that require a treatment impervious to air (generally a thin plastic film facing over the treatment) or washable (basically a stainless-steel-mesh treatment).

Duct linings may be designed for the dual function of providing sound absorption and thermal insulation. Thickness varies between 12 mm and 50 mm, usually adequate for thermal insulation, but very limited in sound absorption at low frequencies. The approximate attenuation of various lined ducts can be found in Reference 8.4, Chapter 42, Table 42.3.

In this case the needed attenuation can be provided either by extending internal lining throughout the supply duct system or by use of a packaged silencer inserted in the ductwork. The use of lined ducts will generally only give insufficient low frequency sound attenuation. One can also use lined elbows instead of lined ducts but the low-frequency performance will be similar. Reference 8.4, Chapter 42, Table 42.5 gives data for lined elbows. Improved performance at low frequencies can be obtained using sound-absorptive plenums and/or resonant sound absorbers.

We can also refer to a manufacturer's chart for performance data and select an appropriate silencer. Typically a silencer of 2 meters length or longer will be adequate in this system.

It must be stressed that the return system must also be analyzed for its performance.

## **Sound-absorptive plena**

A plenum in an air conditioning system is an enclosed volume which is lined with sound-absorptive material and has inlet and outlet ducts that have surface areas that are much smaller than the wall areas in which the ducts are set. Sound-absorption provided by a lined plenum at the discharge will often be an economical way of reducing duct-transmitted noise. Reference 8.4, Chapter 42 gives some advice on the construction of such plena.