Table 16 .... Approximate Levels of Sound Power Generated by Air Flowing Through Elbows[a] With Turning Vanes at 2000 fpm

Duct Size Sq Ft	Octave Band Center, cps						
	106	212	425	850	1700	3400	6900
0.25	52	52	51	48	44	38	31
0.5	57	56	55	51	47	41	34
1	62	60	58	54	50	44	37
2	73	66	62	57	53	47	40
	125	250	500	1000	2000	4000	8000
0.25	52	52	50	47	43	36	29
0.5	57	56	54	50	45	39	32
1	62	60	57	53	49	42	35
2	71	65	61	56	52	45	38

[a] Valid for 30 to 90 deg elbow turn angles.

as long as the vanes are sufficiently rigid and braced to prevent vibration. For air flow velocities other than 2000 fpm, add velocity correction factors from Table 17.

Technical Committee 1.7 of ASHRAE is sponsoring a research project for establishing design information on air flow noise in other duct elements. The following discussion and the procedure for applying air flow noise data illustrated in *Example 4* should be helpful when more comprehensive data become available.

Table 17 shows that even a moderate decrease in air velocity will reduce the level of the air flow noise considerably. If space permits, it is therefore usually possible to avoid air flow noise problems simply by sizing the duct and fittings so that the air flow noise level is well below the fan noise level at any point in the system. This is illustrated in *Example 4*.

Example 4: Calculate the required duct attenuation for the system shown in Fig. 15, using the same fan as in *Example 3*. Assume that the permissible octave band power levels are 3 db lower than calculated in *Example 2*, to allow for an equal amount of sound coming through the return duct.

Solution: An examination of Fig. 15 shows three possibly significant sound sources: the fan, the 24 × 36 in. vaned elbow, and the 12 × 18 in. branch take-off with vanes.

Each of these sources should be treated separately, and the requirements for added attenuation calculated to reduce the sound from each source to the design goal in the room. The attenuation requirements should then be added in accordance with Table 2 to give the total required attenuation. It is most important that the actual attenuation required for each be determined as the algebraic difference between the sound power level from that source and the allowable level (this difference may have either positive or negative values), and that they always be added in accordance with Table 2.

Table 18 presents the calculation for the fan sound output and that of the vanes in the branch take-off. Note that the attenuation required for sound from the vanes is over 20 db less than that required for sound from the fan. The former, therefore, has no effect when added to the latter in accordance with Table 2.

Air flow velocity in the 24 × 36 in. elbow is 1000 ft/min, and in the branch take-off is 800 ft/min. According to Table 17, this difference in velocity amounts to a 6 db difference in sound generation. When the attenuation of the large duct and of the branch take-off (lines 3 and 5 of Table 18) is considered, it is clear that this source will be of no more significance in determining required attenuation than was the 24 × 36 in. elbow. Accordingly, line 20 of Table 18 shows the total required attenuation as identical to that calculated as required for the fan alone.

In the *quiet* part of the system, between the sound treatment and the air-conditioned room, even relatively low air flow velocities may regenerate excessive noise. In order to achieve low design goals, *duct velocities in this part of the system should be correspondingly low, and fittings should be designed for low turbulence.* As a precaution, it is desirable to line the last five feet of duct before each air outlet in low velocity systems with one inch thick fiber blanket, in order to attenuate any unforeseen fitting noise.

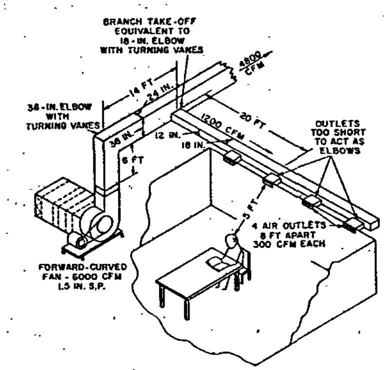

Fig. 15 .... Air-Conditioning Supply System

In high velocity, high pressure air distribution systems, the duct velocities and static pressures are such that the air flow and valve noises have to be controlled by sound attenuating terminals. Although sound attenuation is one of their most important features, commercial high velocity terminals are primarily selected on the basis of their sound power ratings, as described in Step 2. This is so because the sound power rating expresses the net effect of the sound generated by the mixing and throttling valves minus the attenuation provided by the attenuator section of the unit. Separate attenuation ratings for terminal units are used only to determine what, if any, additional duct attenuation is needed to take care of the low frequency part of the fan sound.

Frequently overlooked sources of noise in high velocity systems are the walls of ducts and fittings, as well as fan noise transmitted through the return duct. The sound treatment

Table 17 .... Effect of Air Flow Velocity on Noise Generated by Air Flow in Duct Fittings[a]

Velocity, fpm	800	1000	1200	1500	2000	2500	3000	4000
Decibels to be added to noise generated at 2000 fpm	−24	−18	−13	−7.5	0	+6	+10	+18

[a] To interpolate between velocity values listed, add 1 db for each 4 percent increase in velocity.

Table 18 .... Sample Calculation of Duct Attenuation Required to Reduce Fan and Fitting Noise for Example 4

Line	Item	Reference	Octave Band Center, cps						
			106	212	425	850	1700	3400	6900
1	Sound Power Output of: Fan	From Ex. 3	89	89	79	74	69	64	59
2	Power Divisions to Terminal, 5% Area	Table 12	13	13	13	13	13	13	13
3	Natural Attenuation, 24 × 36 in. Duct, 20 ft long	Table 13 × Length	4	2	1	1	1	1	1
4	Attenuation of 36 in. Elbow, with Vanes	Tables 14 and 15	2	4	4	3	3	3	3
5	Elbow Attenuation of 18 in. Take-Off, with Vanes	Tables 14 and 15	0	2	5	3	3	3	3
6	Natural Attenuation, 12 × 18 in. Duct 20 ft long	Table 13 × Length	4	3	2	2	2	2	2
7	End Reflection for Size 12 × 16 in. $Q = 4$	Figure 13	5	1	0	0	0	0	0
8	Total Natural Attenuation in Branch	Sum of Lines 2 to 7	28	25	25	22	22	22	22
9	Sound Power Level per Terminal without Sound Treatment	Line 1 Minus Line 8	61	64	54	52	47	42	37
10	Permissible Sound Power Level per Terminal, Allowing 3 db for Return Duct	Line 18 of Step 2, Minus 3 db	52	45	39	34	33	31	31
11	Min. Attenuation of Sound Treatment to be designed for 12 × 18 in. Duct	Line 9 Minus Line 10	9	19	15	18	14	11	6
12	Sound Power Output of: Vanes of Take-Off (800 FPM)	From Tables 16 + 17	44	39	36	32	28	22	15
13	Power Divisions to Terminal, 25% Area	Table 11 or 12	6	6	6	6	6	6	6
14	Natural Attenuation, 12 × 18 in. Duct, 20 ft long	Table 13 × Length	4	3	2	2	2	2	2
15	End Reflection for Size 12 × 18 in. $Q = 4$	Figure 13	5	1	0	0	0	0	0
16	Total Natural Attenuation in Branch	Sum of Lines 13 to 15	15	10	8	8	8	8	8
17	Sound Power Level per Terminal without Sound Treatment	Line 12 Minus Line 16	29	29	28	24	20	14	7
18	Permissible Sound Power Level per Terminal, Allowing 3 db for Return Duct	Line 18 of Step 2, Minus 3 db	52	45	39	34	33	31	31
19	Min. Attenuation of Sound Treatment to be designed for 12 × 18 in. Duct	Line 17 Minus Line 18	−23	−16	−11	−10	−13	−17	−24
20	Total Attenuation Required	Line 11 Added to Line 19 Per Table 2	9	19	15	18	14	11	6

required for the return duct system should be calculated following the procedure illustrated in *Example 4*.

To prevent excessive noise radiation from the walls of high velocity ducts, all fittings should be smooth and designed to avoid abrupt changes of direction or velocity. Whenever possible, high velocity ducts and terminal units should be located in noncritical areas, such as above corridors. Plaster ceilings or tightly sealed window stools are good sound barriers, but perforated ceiling tiles are not.[53] The space above the acoustical tile ceiling of a quiet room should not be used to install a group of high velocity terminal units serving surrounding rooms.

Cross Transmission Between Rooms

Radiator enclosures, etc., should not extend through partitions, and all openings around pipes and ducts should be sealed tightly to prevent sound leakage between rooms. Barriers should be provided in suspended acoustical ceilings above the partitions.[53]

To eliminate cross transmission through the duct system, sound attenuation should be provided in ducts between air outlets in adjacent rooms.[54] The required duct attenuation will depend upon the degree of sound isolation (transmission loss, TL) for which the wall is designed. Normally, the sound coming through the duct should not add more than 1 or 2 db to the sound coming through the wall.[9] On this basis, the required duct attenuation can be calculated from Equation 15.

$$\text{Attenuation, db} = (TL) + 6 - 10 \log_{10} S_p/S_o \quad (15)$$

where

(TL) = transmission loss of the partition, decibels.
S_p = area of the partition, square feet.
S_o = area of outlet, square feet.

Transmission loss ratings for typical wall constructions in the speech frequency range (600–4800 cps) may be obtained from References 9 or 55. The factor $10 \log_{10} (S_p/S_o)$ can be found from Fig. 1 for any ratio of the partition area S_p to the outlet area S_o. *Example 5* illustrates the use of Equation 15.

Example 5: Calculate the attenuation required in a duct between two rooms, each having two 12 × 12 in. air outlets. The partition between the rooms has an area of 200 sq ft and an average transmission loss of 40 db in the speech frequency range (600 to 4800 cps).

Solution: From Fig. 1:

$$10 \log_{10} (S_p/S_o) = 10 \log_{10} (200/2) = 20 \text{ db}$$

From Equation 15, the required attenuation is 40 + 6 − 20 = 26 db.

To provide the required duct attenuation, duct linings, splitters, cells, outlet absorbers, or prefabricated silencers may be used. In high-velocity supply systems with separate terminal units for each room, the combined attenuation of two terminal units in series is adequate for practically all cross transmission problems, even in music practice rooms.