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HVAC SYSTEMS for Theatres, Auditoriums & Cinema Halls

The art and science of sysytem design

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The design consideration for any air conditioning system depends largely on its application. It is a known fact that Theatres, Auditoriums, Concert Halls or Motion Picture

Theatres (Cinema Halls) are high occupancy (persons per sq.ft) areas. Consequently, the fresh air requirement is high in order to dilute the CO_2 concentration. High occupancy also leads to a greater amount of latent load emitted by the human beings. All these considerations need to be factored into, by the designers while calculating the air conditioning load. The inside design condition is as per ASHRAE standard for human comfort. Typically, the Comfort band on Temperature varies between 70 to 76°F (21.1°to 24.4°C) with RH not exceeding 65%.

Another important criterion is the noise and vibration control. The desired Noise Criteria (NC) varies with the type and quality of the facility. The need for noise control is lower in Cinema Halls but it is important in Theaters, Auditoriums, and Concert Halls. In most cases, sound and vibration control is required both, for mechanical equipment and the air distribution system.

Table 1 summarises a list of important Theatres, Auditoriums and Cinema Halls air conditioned by the company, showing the plant capacity, occupancy, fresh air quantity, duct velocities and NC level maintained, and will give the reader a fair idea of existing practises in the industry.

Table 1 : HVAC design parameters of some theatres, auditoriums and cinema halls air conditioned by Voltas

Name	Area sq. feet	Height feet	Capacity tons	Occupancy	Fresh Air cfm	Inside Design Condition temp. (°F)	Inside Design Condition RH (%)	AHU capacity cfm	Fan Speed rpm	Fan Outlet velocity (fpm)	Duct Velocity fpm	Noise Level in auditorium	Filtration level (microns)
NCPA, Mumbai													
Auditorium	17700	30-45	110	1100	4000	73±2	55	52000	650	1800	SA< 900	NC 20	10
Stage+Orchestra Pit	2000	45	130	250	4000	73±2	55	52000	650	1800	RA< 600	NC 20	10
NIMHANS, Bangalore													
Main Auditorium	6000	28.5	100	745	2 Air Change	73±2	60	30000	635	1731	1100	25 TO 30 NC	20
Stage	3500	30.8	12	15	2 Air Change	73±2	60	30000	635	1731	-	25 TO 30 NC	-
Mini Auditorium	3500	9.8	41	207	2 Air Change	73±2	60	12000	635	1731	1100	25 TO 30 NC	20
Tagore Memorial Hall, Ahmedabad	8200	30	120	750	5600	73±2	55	53000	795	1800	1100	No Tender Spec.	10
Sardar Patel Smruti Bhavan, Surat	15280	30	130	1280	10000	73±2	55	44000	743	1800	1500	No Tender Spec.	10
PRASAD (IMAX), Hyderabad													
IMAX Theatre	13806	60	84	635	9500	73±2	55	22000	490	1595	1000	NC 25	10
Theatre 1	5385	20	34	350	3500	73±2	55	8000	720	1830	1000	NC 25	10
Theatre 2	5385	20	34	350	3500	73±2	55	8000	720	1830	1000	NC 25	10
Theatre 3	5385	20	34	350	3500	73±2	55	8000	720	1830	1000	NC 25	10
Theatre 4	5385	20	34	350	3500	73±2	55	8000	720	1830	1000	NC 25	10
Theatre 5	2875	20	28	225	3375	73±2	55	7000	723	1595	1000	NC 25	10
Projector Room (Theatre 5)	241	9	7	3	700	73±2	50	1900	1299	1860	<1500	N/A	5
Multiplex Projector Room	3028	9	24	10	2200	73±2	50	7450	891	1773	<1500	N/A	5
IMAX Projector Room	1396	9	14.5	5	1600	73±2	50	4000	655	1812	<1500	N/A	5
INOX, Pune													
Theatre 1	2800	19	50	250	2000	73±2	55	15000	700	1500	1000	NC 30	10
Theatre 2	4800	19	50	440	2000	73±2	55	15000	700	1500	1000	NC 30	10
Theatre 3	2800	19	60	250	3500	73±2	55	18000	700	1500	1000	NC 30	10
Theatre 4	4800	19	60	440	3500	.73±2	55	18000	700	1500	1000	NC 30	10
FAME ADLABS, Mumbai													
Main Auditorium (Typical) 5 Nos.	3636	34	29	300	1 Air Change (500)	73±2	55	9000	753	<1800	<1100	Not specified in tender	10
Common Projector Room	1840	9.8	9.15	5	0.5 Air Change (151)	73±2	55	5000	960	<1800	<1500	N/A	Room at -ve pressure.

Air Distribution

Generally, the seating area of any Auditorium or Cinema Hall is isolated from the exterior by lobbies, corridors, etc. In the seating area, people generally remain in one place throughout the show. A good air distribution system ensures that air reaches the entire seating area without creating drafts and noise. There are multiple options of locating the supply air ducting and return air (ducted or otherwise). The most common and possibly the best method is to locate the supply air duct at the ceiling level and distribute the air uniformly through ceiling diffusers, the design of which will depend on the height at which duct is located. The return air can be located below the seat and ducted back to the air handling unit room through trenches in the floor. This layout of air distribution would ensure uniform distribution of air without draft and noise. In a lownoise requirement, it is advisable to acoustically line the supply and return air duct completely with fiberglass wool of adequate thickness.

Another alternative is to locate the supply air duct along the sidewalls of the hall. The duct can be concealed in boxing and the supply air grilles can be either continuous or conventional. The grille is to be designed for adequate throw so that air reaches even the central portion of the seating area.

However, very large halls require larger throw from the sidewalls and such a requirement could vary between 50 to 150 ft. In such installations, selection of the grille is very important to meet the throw requirement. Generally, this results in a high air discharge velocity leading to noise as well as draft. It is therefore, recommended that supply air throw should be designed for a maximum of 30 ft throw from the side and to ensure it reaches the central portion of the hall, the return air should be collected below the seat.

Another alternative to take the return air in directly below the stage, and this method is quite commonly followed.

Recommended design velocity for supply air in the main duct should not exceed 1200 fpm and of return air 600 fpm. The recommended velocity at the outlet of each grille / diffuser is to be maintained between 150 to 200 fpm.

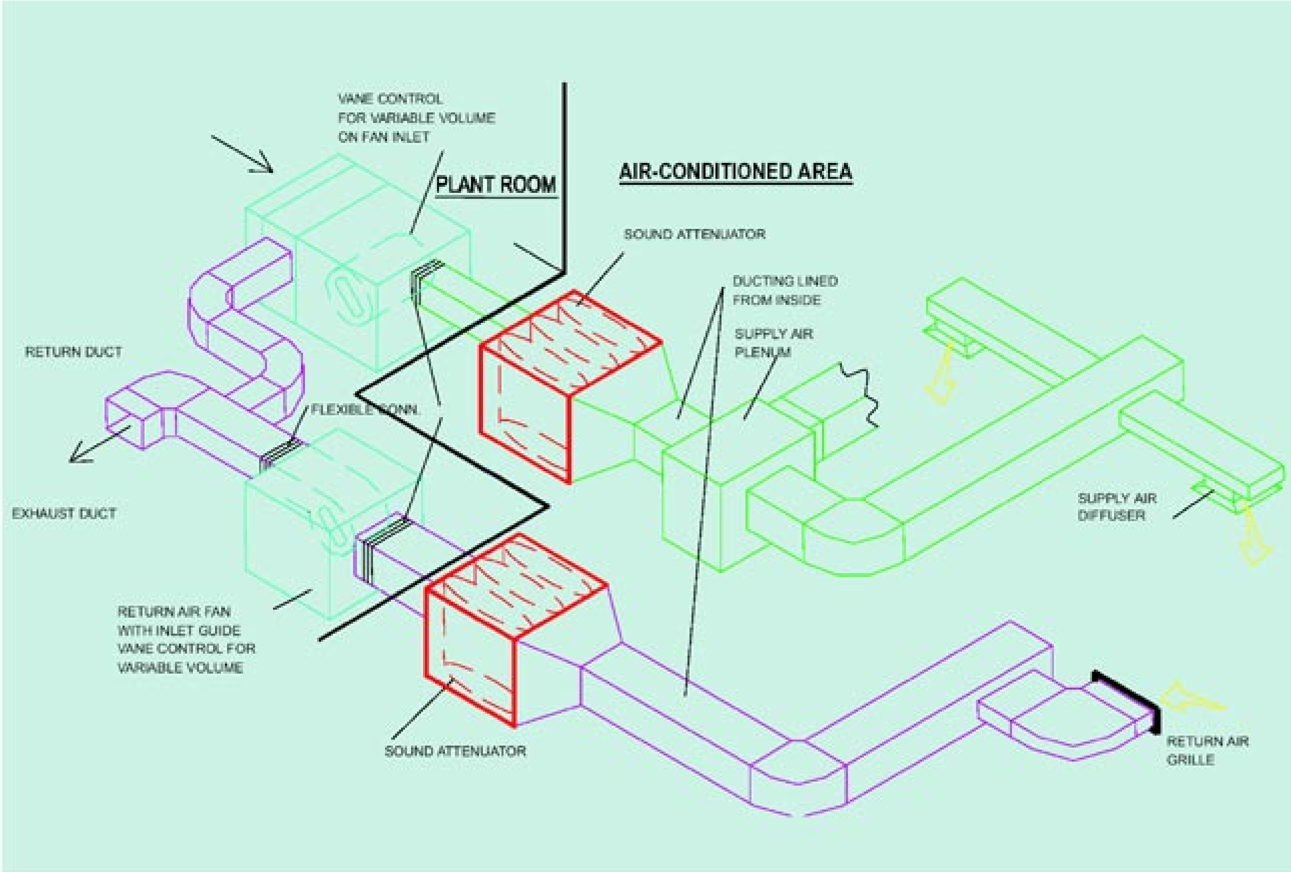

Figure 1 : A schematic drawing of an air distribution system

Mechanical Equipment

Location of mechanical equipment such as Chillers, Air Handling Units, and Pumps etc. is quite critical, particularly, where low noise criteria is a key requirement. It is advisable to have the AC plant room and the air handling unit room in a separate area, which is structurally isolated from the Theatre or Auditorium. The locational constraints sometimes force us to accept an air handling unit room within the same structure as that of the Auditorium and sometimes very near the Auditorium. It is essential to ensure, that in such installations, the air handling unit (AHU) is mounted on correctly selected vibration isolators and the chilled water pipe is provided with a flexible connection to avoid transmission of vibrations to the structure. The design of the AHU has to be for low rpm i.e. 650 rpm with outlet velocity not exceeding 1800 fpm. It is preferable to have AHU fans of backward-curved type and the bearing of nylon pillow block. A double-skin type of AHU is always preferred for low noise application. It is important to acoustically line the walls of the AHU room. Sound attenuators should be provided both in the supply air duct as well as return air path. Wherever return air is not ducted, it is recommended to have the return air attenuator on the return air opening of the Auditorium wall.

Cinema Halls or Motion Picture Theatres

Cinema Halls are the simplest of the auditorium structures. They have continuous operation of 8 to 12 hours in case of commercial Cinema Halls or 4 to 8 hours in case of Multiplexes. They operate between full occupancy to low occupancy and require plant performance to be controlled at low load. Noise criteria is not so important in this application. Lobby and passages have very dense occupancy in-between shows or during intervals. Lighting load is very low during the show and generally smoking is not permitted.

Projection Booths

Large theatres using high-intensity lamps require careful handling of design for projection rooms. Normally the projection equipment manufacturer recommends exhaust out of the projection room and thus the projection room is maintained at negative pressure. Exhaust is normally taken through the housing of the projectors. The other heat source includes sound and dimming equipment, which require controlled environment. A similar theatre using 16mm films is less complex in its requirement for the projection booths. It is good practice to air-condition the projection room giving a filtered supply air to avoid soiling of lenses.

Stage

In a performance theatre, the stage air conditioning has many complexities:

- Extremely heavy lighting load, which, at times, is mobile.
- Scenery on the stage, which could be delicate and intricate requiring careful air distribution.
- In case of very brisk movement of the artiste, as in a dance, the sensible load and latent load addition due to human occupancy has to be different.

Controlled Ozone Injection

Considering the increasing cost of energy and also the increasing awareness of Indoor Air Quality, controlled and engineered Ozone injection has been found to be an ideal solution to achieve both. Ozone, when injected in the indoor air through the supply air duct in controlled quantities, oxidizes various pollutants including volatile organic components such as carbon monoxide, toluene, formaldehyde, etc. which are commonly found in indoor furnishings. Ozone also removes all odours and eliminates fungi. It is important that injection of Ozone is within acceptable limit as per international norms to avoid illeffects of excessive Ozone.

In applications where Ozone injections are applied to improve indoor air quality, the fresh air requirement also gets considerably reduced from the ASHRAE standard of 15 cfm per person. This leads to a considerable reduction in air conditioning load thus saving capital and operating costs.

Regal Cinema in Mumbai had the problem of fish odour getting into the cinema hall, whenever a fish truck from the docks passed by. Regal Cinema then installed Ozone injection and since then the indoor air quality has improved and can be experienced as a testimony of Ozone's effectiveness. There are many installations in India and abroad, where the public areas, in particular have been treated with Ozone injection for improvement of indoor air thus reducing the air conditioning load and operating cost.

Noise

We will deal with this topic of 'Noise' and 'Noise Control' in great detail, since this is one of the most critical parameters in such applications. The major sources of noise in any air conditioning system are:

- **Supply and Return Fans**

Fans generate the maximum noise. Centrifugal fans produce noise across the entire audio frequency band having maximum value between 31.5 to 250 Hz. Thus, it is evident that these values are maximum in the low frequency band. The lowest noise levels are produced when the fan is operated in the region of its peak efficiency on its performance curve. The following formula can be used to calculate the efficiency.

$$\text{Static Efficiency} = \frac{\text{Flow Rate (CFM)} \times \text{Static Pressure (Inches)}}{K \times \text{BHP Input (HP)}}$$

where K = 6345 (a constant in the FPS system)

The sound power level of the noise generated by the fan can be obtained from the manufacturer for the entire octave band frequency 63 Hz to 4000 Hz. The increase in the sound power level on account of the fan not operating at its peak efficiency can be predicted using **Table 2**. Emperical data is available to predict the increase in dB based on efficiency of fans. **Table 2** shows how to determine noise power level for the fan selected for the application.

Table 2 : Increase in noise level based on efficiency

Aero Foil Centrifugal Fan & Vaneaxial		Backward Curved Centrifugal Fan		Forward Curved Fan	
Efficiency	Increase in dB	Efficiency	Increase in dB	Efficiency	Increase in dB
72 to 80	0	67 to 75	0	58 to 65	0
68 to 71	3	64 to 66	3	55 to 57	3
60 to 67	6	56 to 63	6	49 to 54	6
52 to 59	9	49 to 55	9	42 to 48	9
44 to 51	12	41 to 48	12	36 to 41	12

• Smooth Air Flow

- In case the system does not have good aerodynamic design and efficient operation of various components, the noise level of sources increases in the low frequency range, if fans operate at low efficiency and in an unstable region.
- Airflow at entrance and exit should be smooth to minimise generation of noise.
- Examples of good and bad airflow conditions are shown in **Figure 2**. Noise generation increases 10 to 30 dB

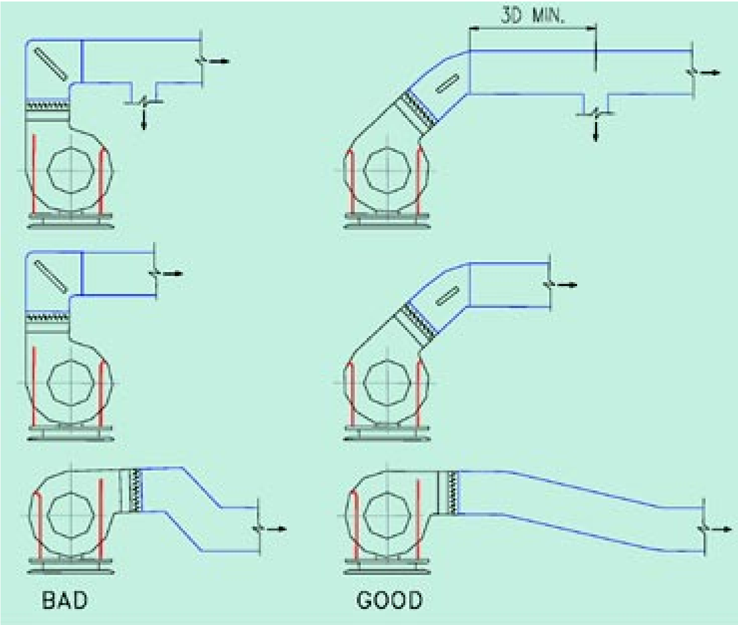

Figure 2 : Examples of good and bad fan outlet conditions

• Ducting, Dampers, Air Terminal Devices

Deflectors such as vanes, fins, etc used to deflect air for even distribution of the same throughout the airconditioned space leads to some generation of noise. The greater the deflection, higher is the noise level produced.

The sound power level (Lw) generated by air flowing through diffusers increases by 16 dB for octave bands around 500 Hz and 16 to 24 dB for octave bands between 500 to 1000 Hz, in case the velocity of the air passing through diffusers is doubled.

Dampers, which are integral parts of grilles or diffusers, to control the volume of airflow, increase/ radiate the noise frequency in the range of 1000 to 8000 Hz. More the restriction to the airflow, higher the noise level. Thus, if the airflow is reduced by half using dampers, there will be an increase in the sound power levels by 10 to 20 dB depending on the design of the air outlet.

It is therefore, recommended not to use dampers in the grilles for such sensitive applications. Instead dampers can be installed in the duct for achieving the same objective.

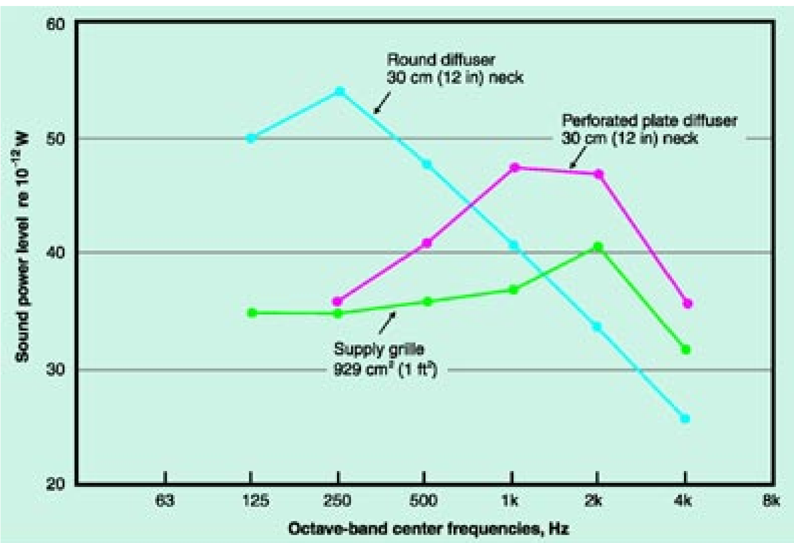

Figure 3 : Sound power level of different air outlets

Figure 3 predicts the noise power level generated using different types of grilles and diffusers at various frequency bands. This table can be used to predict the noise power level of a single grille or diffuser. The next step is to predict the noise pressure level (Lp) from noise power level (Lw). The formula below can be used for calculating the sound pressure level. This formula is applicable only for Auditorium and Lecture Hall applications. The volumes of such spaces are in excess of 20,000 ft3 and are usually characterised by many seats, people and sound absorptive materials.

$$Lp = Lw - 5\log_{10}X - 28\log_{10h} + 1.3\log_{10\,N} - 3\log_{10}f + C$$

Hence,

$$Lp - Lw = -5\log_{10}X - 28\log_{10h} + 1.3\log_{10\,N} - 3\log_{10}f + C$$

Where: -

- Lp - average sound pressure level for total diffusers at height of 1.5 m above floor

- Lw - sound power level of single outlet in the array
- X - ratio of floor area served by each outlet to square of ceiling height
  (eg X = 1 if area served h)
- h - Ceiling height
- N - No. of diffusers in room (N should be at least 4)
- f - Octave band frequency Hz
- C - Proportionality constant 31 in FPS

A misaligned diffuser can create noise *eg* misalignment of 1 dia in 2 dia length can increase the sound power by 15 dB

Figure 4 illustrates the correct installation of a diffuser with respect to the duct outlet. The duct connection shown in the sketch is a flexible duct but the same is true also for a rigid duct.

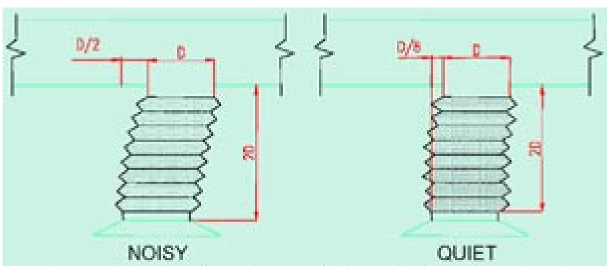

Figure 4: Correct and incorrect installation of a flexible duct connection between main duct and a diffuser

Noise Control

The control of noise in a duct air distribution system can be achieved through attenuation of sound in ducts using sound attenuators or sound absorptive plenums. The systematic approach to attenuation is to reduce the noise level in the various components of the air distribution system.

Sound is attenuated in the duct system by insertion losses in:

- Unlined ducts
- Lined ducts
- Lined elbows
- Branch points resulting from division of power
- Result of end reflection

Unlined ducts. Attenuation provided by unlined rectangular sheet metal ducts in dB/metre is shown in **Table 3**.

Table 3 : Attenuation provided by unlined rectangular duct

Perimeter/Area ratio (cm/sq.cm)	Hz		
	63	125	250
0.31	0.3	1.0	0.3
0.13-0.31	1.0	0.3	0.3
Under 0.05 to 0.13	0.3	0.3	0.3

If the ducts are externally lined, values must be doubled.

Transmission losses depend on the level of sound within a duct, cross sectional area of the duct and length of duct in occupied space. The same is represented in **Figure 5** and in **Table 4**.

Table 4 : Attenuation provided by unlined ducts

Rectangular Duct (unlined)

Dimensions		Perimeter / area ratio								
cm x cm	(in x in)	cm/cm²	in/in²	dB/metre						
10 x 20	(4 x 8)	0.30	(0.75)	0.7	2.1	3.3	8.2	22.4	21.2	8.4
15 x 30	(6 x 12)	0.20	(0.50)	0.6	1.9	2.6	6.3	17.3	15.7	7.6
20 x 30	(8 x 12)	0.17	(0.42)	0.6	1.8	2.4	5.8	15.9	16.3	8.7
25 x 41	(10 x 16)	0.13	(0.33)	0.5	1.7	2.1	4.9	13.5	13.3	7.9
30 x 30	(12 x 12)	0.13	(0.33)	0.6	1.7	2.3	5.4	14.8	14.9	8.8
30 x 61	(12 x 24)	0.10	(0.25)	1.2	0.8	1.8	4.1	11.1	9.5	6.5
30 x 91	(12 x 36)	0.09	(0.22)	1.2	0.8	1.6	3.6	9.8	5.8	4.2
38 x 38	(15 x 15)	0.11	(0.27)	1.2	0.9	2.0	4.7	12.7	12.6	8.3
38 x 76	(15 x 30)	0.08	(0.20)	1.2	0.8	1.6	3.5	9.5	8.1	6.1
46 x 71	(18 x 28)	0.07	(0.18)	1.2	0.8	1.5	3.4	9.3	8.8	7.0
46 x 91	(18 x 36)	0.07	(0.17)	1.2	0.7	1.4	3.1	8.5	7.1	5.8
61 x 91	(24 x 36)	0.05	(0.14)	1.1	0.7	1.3	2.9	7.8	7.3	6.6
61 x 122	(24 x 48)	0.05	(0.13)	1.1	0.6	1.3	2.6	6.3	5.8	5.5
76 x 76	(30 x 30)	0.05	(0.13)	1.1	0.7	1.4	3.0	8.2	7.6	7.0
76 x 152	(30 x 60)	0.04	(0.10)	0.4	0.6	1.3	2.2	4.4	4.9	5.1
91 x 91	(36 x 36)	0.04	(0.11)	0.4	0.7	1.3	2.7	6.6	6.7	6.7
91 x 182	(36 x 72)	0.03	(0.08)	0.4	0.6	1.0	2.0	3.3	4.3	4.9
107 x 107	(42 x 42)	0.04	(0.10)	0.4	0.6	1.2	2.4	5.2	5.9	6.5
122 x 122	(48 x 48)	0.03	(0.08)	0.4	0.6	1.1	2.2	4.3	5.4	6.2

Circular Duct (unlined)

Diameter cm (in) :							
15.2 (6)	0.7	1.6	3.3	5.9	7.2	7.2	6.6
30 (12)	0.5	1.0	2.3	4.9	7.2	7.2	4.9

61 (24)	0.3	0.7	1.6	3.3	5.6	3.0	1.6
122 (48)	0.1	0.3	1.0	2.0	2.0	1.6	1.6

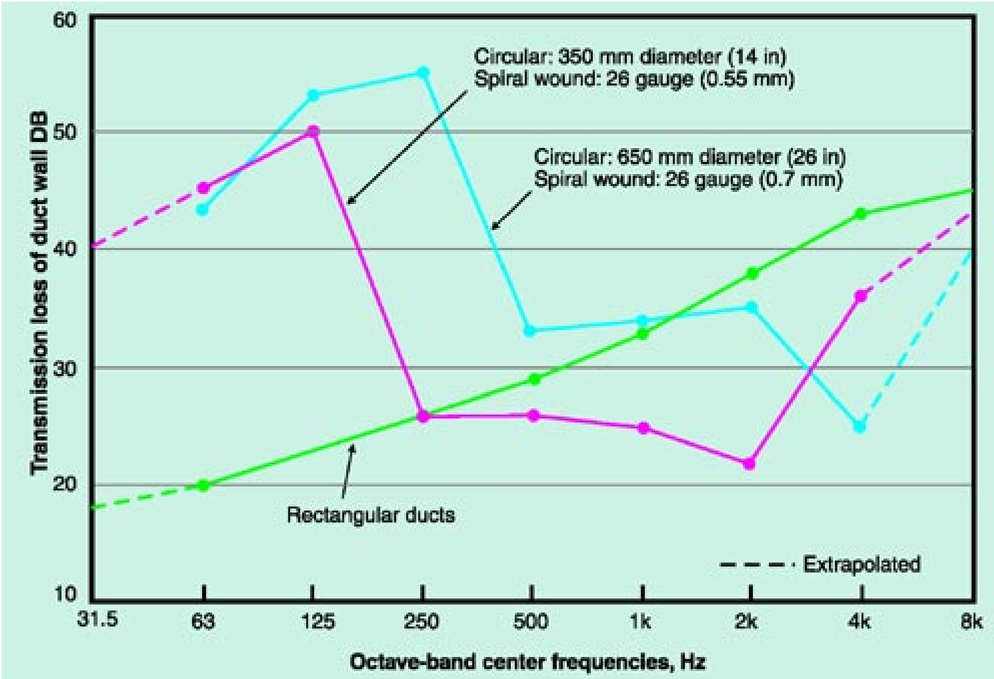

Figure 5: Transmission loss for sound propagated from the interior of a rectangular duct through the duct wall to the exterior

Table 5 : Attenuation of lined elbows

Width of duct in plane of turn in inches	**63**	**125**	**250**	**500**	**1000**	**2000**
	--------------------Hz------------------------					
5 to 10	0	1	1	2	3	4
11 to 20	0	1	2	3	4	6
21 to 40	1	2	3	4	5	6
41 to 80	2	3	3	5	6	8
Unlined elbows all sizes	6	1	2	3	3	3

Lined duct. Attenuation of sound provided by lined duct with 1" and 2" thick insulation, is shown in **Figure 6**.

Lined Elbows. Attenuation of lined elbows, 90deg rectangular with/without fiberglass, 25 mm lining, is shown in **Table 5**.

Branch Points. Division of sound power at a duct branch results in attenuation that can be expressed as :

$$\text{Attentuation} = 10 \log_{10}\left\{\frac{\text{area of branch}}{\text{area of main}}\right\}\text{db}$$

and is shown in **Table 6**.

End Reflection. Attenuation of sound in ducts resulting from end reflection loss in dB is shown in **Table 7**.

Table 6 : Attenuation at a lined duct branch due to division of power

Ratio area of branch/main duct	0.01	0.02	0.04	0.06	0.08
dB to be subtracted	20	17	14	12	11
Ratio area of branch/main duct	0.1	0.2	0.4	0.6	0.8
dB to be subtracted	10	7	4	2	1

Table 7 : Attenuation in ducts from end reflection

Hz	63	125	250	500	1000
6	18	12	8	4	1
12	13	8	4	1	0
24	8	4	1	0	0
48	4	1	0	0	0

Table 8 : Attenuation/self noise generation for face velocity 1000 fpm and pressure drop 0.4"wg

Hz	63	125	250	500	1000	2000	4000	8000
Insertion loss db	4	11	20	37	44	44	38	22
Self noise Lw db generation	48	42	39	35	36	37	35	30

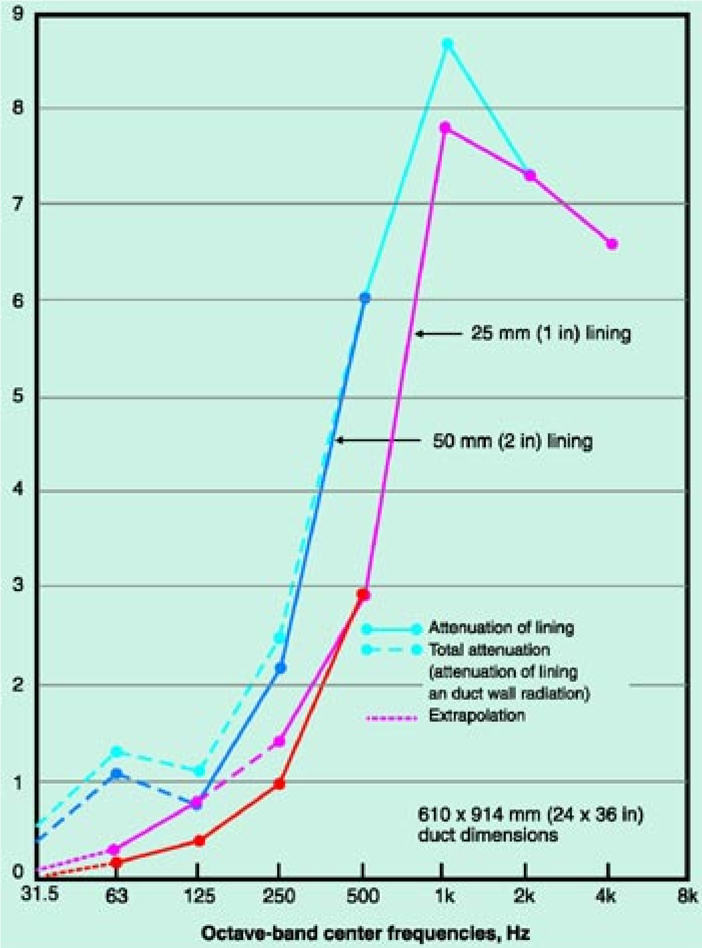

Figure 6 : Graphical representation of sound attenuation by lined ducts

Attenuation Due to Sound Attenuator or Duct Silencer

A sound attenuator has a cross section equal to or greater than the duct in which it is installed.

A sound attenuator has three important characteristics, which are a function of air velocity at the face of the attenuator :

• Insertion loss

• Pressure drop

• Self noise

Figure 7 shows a typical attenuator. **Figure 8** shows a comparison of the insertion loss provided by a 2.1m long sound attenuator :

• A low-pressure drop unit

• A standard pressure drop unit

• Duct of the same length and cross section lined with a fiberglass blanket 25 mm thick

Note: In areas like concert halls/auditoriums, where air velocities are less than 1000 ft/min, the insertion loss is far greater than the self noise propagated at frequencies less than 500 Hz.

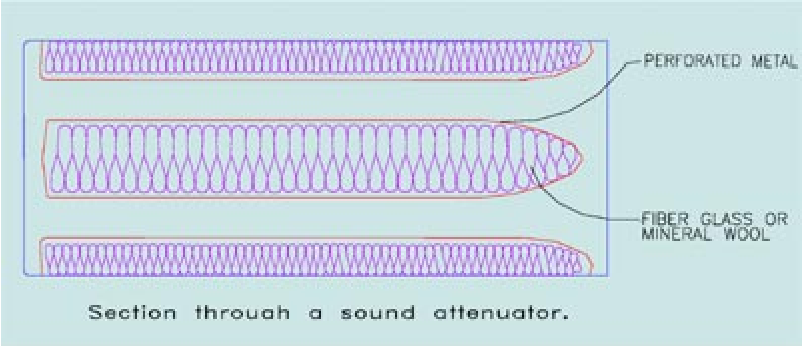

Figure 7 : Section through an attenuator

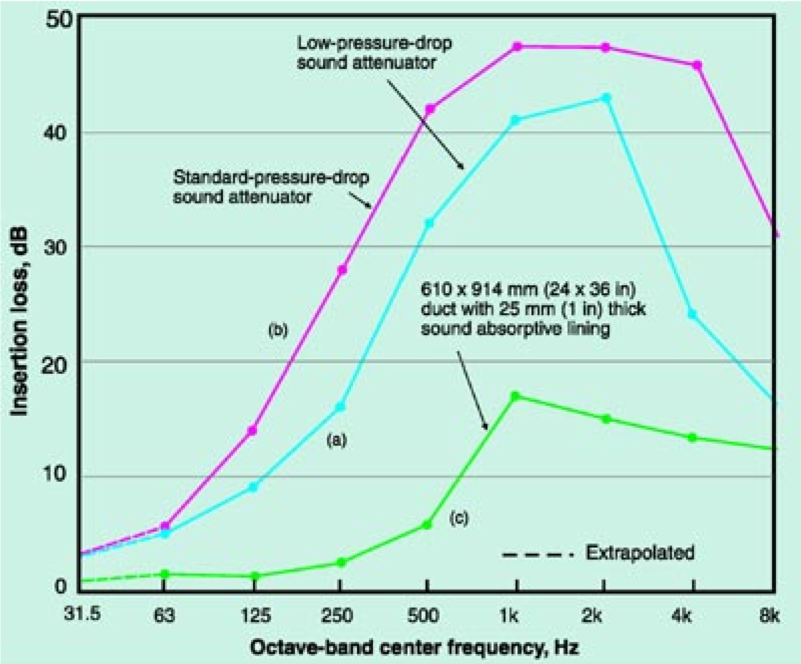

Figure 8 : Comparison of insertion loss provided by an attenuator of two different types and a lined duct

Determination of Extent of Sound Attenuation

Attenuation of noise can be done by considering:

Primary Attenuation. Deals with the location of the source of noise (eg fan). It also considers the aspects of how the noise can be controlled from the source by reducing the rpm, outlet fan velocity, considering the materials in the bearing, and whether backward or forward curve. Fan type should be used depending on the critical nature of the noise criteria to be maintained within the air-conditioned space.

Secondary Attenuation. This considers the noise within the ducting work and air terminal devices and the design of these items has to be done in order to reduce the noise level. Additionally, whatever dB level is actually attained has to be matched to the criteria desired within the air conditioned area by the client, which would therefore call for additional attenuation to be carried out viz., by inserting a sound attenuater.

Step I:

We have obtained the sound power level of the fan Lw from the fan manufacturer.

Step II:

From this, we have to account for the insertion losses to arrive at the net sound power level.

Step III:

We have also obtained the value of Lw-Lp applicable for Air Terminal Devices. The actual sound pressure level thus can be arrived at using the following expression:

Lp = Lw – (Lw – Lp)

Step IV:

Using NC chart, we can arrive at the actual NC value without use of a sound attenuator. Hence, the difference between actual and desired NC levels has to be attenuated in the sound attenuator.

Rating System for HVAC Noise

Single number rating systems in common use are

• Weighted sound level

• NC curves

• RC curves

These rating systems for HVAC are valid for continuous steady state noise that exhibits no fluctuations in sound level with time.

Failure to recognise the limitation leads to a serious error in rating noise produced by the system.

Fluctuating noise levels may occur as a result of unstable fan operation OR beats between 2 or more fans OR other rotating equipment which operates at the same speed.

Sound level meters are designed to measure ‘A’ weighted sound level of noise expressed as dB (A)

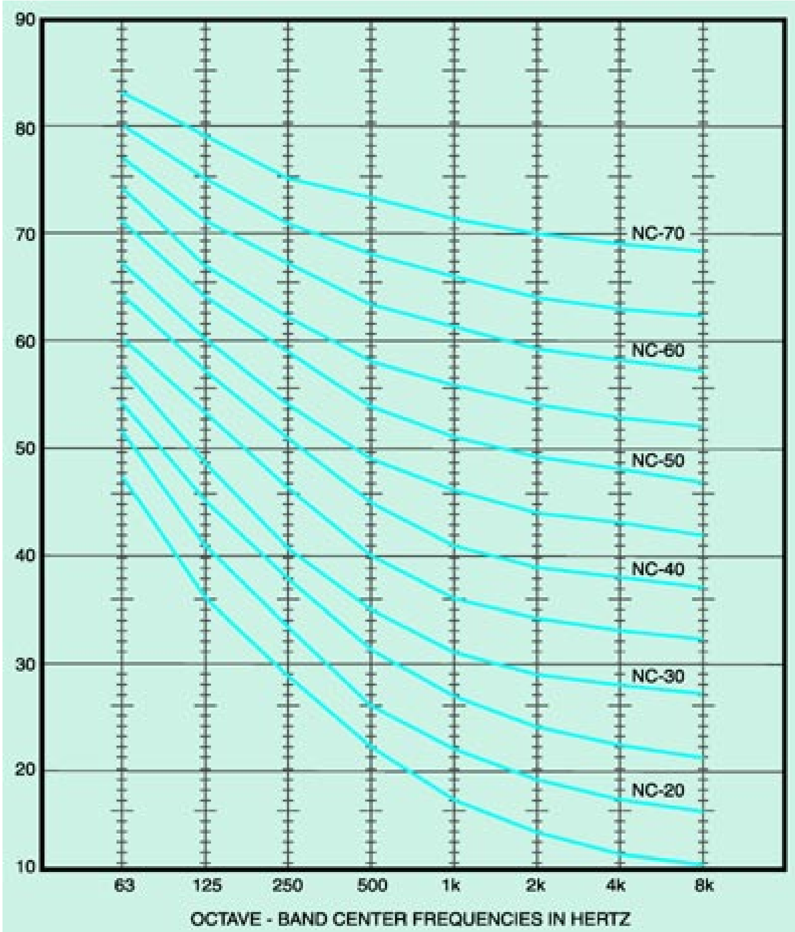

Figure 9 : NC curves

Table 9 : Noise Control Criteria for HVAC systems

Acceptability Criteria For Steady Background Noise In Unoccupied Rooms

Type Of Room	**Recommended Criterion** **Preferred**	**Alternative**	**Approximate A-weighted sound level** * **dB**
Recording Studios	RC 10-20(N)	NC 10-20	18-28
Concert and Recital Halls	RC 15-20(N)	NC 15-20	23-28
TV Studios, Music Rooms	RC 20-25(N)	NC 20-25	28-33
Legitimate Theaters	RC 20-25(N)	NC 20-25	28-33
Private Residences	RC 25-30(N)	NC 25-30	33-38
Conference Rooms	RC 25-30(N)	NC 25-30	33-38
Lecture Rooms, Classrooms	RC 25-30(N)	NC 25-30	33-38
Executive Offices	RC 25-30(N)	NC 25-30	33-38
Private Offices	RC 30-35(N)	NC 30-35	38-43
Churches	RC 30-35(N)	NC 30-35	38-43
Cinemas	RC 30-35(N)	NC 30-35	38-43
Apartments, Hotel Bedrooms	RC 30-35(N)	NC 30-35	38-43
Courtrooms	RC 35-40(N)	NC 35-40	43-48
Open-plan Offices & Schools	RC 35-40(N)	NC 35-40	43-48

Libraries	RC 35-40(N)	NC 35-40	43-48
Lobbies, Public Areas	RC 35-40(N)	NC 35-40	43-48
Restaurants	RC 40-45(N)	NC 40-45	48-53
Public Offices (Large)	RC 40-45(N)	NC 40-45	48-53

* A-weighted sound levels are not recommended for use in the design of HVAC systems

Note

Loudness of sound varies both, with sound pressure and frequency.

Noise induced annoyance is not directly proportional to loudness. 'A' weighted scale is unreliable because specified sound level in dB (A) is not necessarily acceptable to an occupant of a room, supplied by the system.

'A' weighted is not used as a diagnostic tool in analysis of noise complaints about an operating HVAC system as the problem may not only be loudness related. Knowledge of the spectrum of noise is required to discover and correct the reason for complaints.

Hence NC curves are used as shown in **Figure 9** and recommended criteria for both NC and RC are shown in **Table 9** for a variety of applications.