AIR CONDITIONING AND REFRIGERATION
Journal
The magazine of the Indian Society of Heating, Refrigerating and Air Conditioning Engineers

Issue : July-September 2000

Selection, Design and Fabrication of Quieter Fans for HVAC Systems.

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The major source of noise in heating, ventilating and air conditioning (HVAC) systems are fans and blowers of the air handling units.

HVAC fans are mostly of the centrifugal type or the axial flow type. Centrifugal fans may be characterized by the type of blade used. Three types are shown in **Figure 1** - forward curved, airfoil, and radial. Many other designs fall between these examples[1]. The backward curved blades look exactly like the forward curved ones except that they are curved backwards. The forward-curved fan shown in **Fig. 1** is used primarily for heating, ventilating, and air-conditioning work where high volume flow rates and low-pressure characteristics are required in such equipment as package air-conditioning units, and small the three major subcategories shown in **Figure 2** - Vaneaxial, Tubeaxial and Propeller.

In this article, the selection and design of either of these two types of fans for quietness are discussed briefly.

Selection Criteria

The primary selection criterion for a fan is based not on its acoustical characteristics but on its ability to move the required amount of air against the required pressure. In addition, the fan must do so at a reasonable initial cost (it also may be required to handle dust-laden air, to resist abrasion and/or corrosion, to have a type of construction that can be repaired easily in the field, to withstand high temperature, etc) Once these requirements have been met, the type, size and speed of the fan are completely determined then the noise characteristics of he fan also are determined, In most cases it is not practical to substitute a fan which generates less noise, since a quieter design of he same type probably will not meet the other operating specification for the fan. Therefore, sound power levels generated by a correctly selected fan must be accepted as the sound power levels to be used in acoustical design calculations.

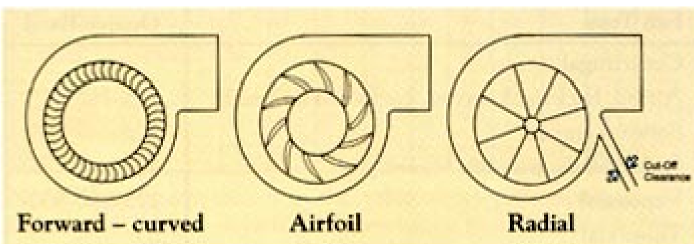

Figure 1 : Examples of centrifugal fans available for various applications; the type of blade is indicated

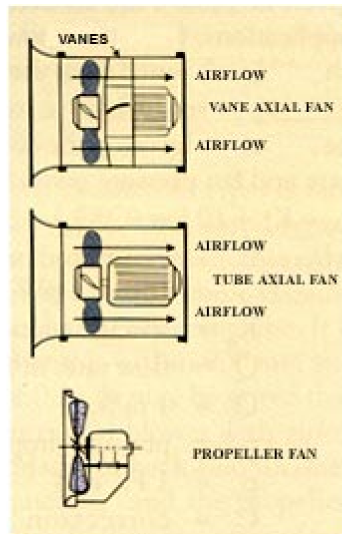

Figure 2 : Three major categories of axial-flow fans

Many engineers use rules of thumb, such as a certain maximum tip speed of the fan impeller or a certain maximum outlet velocity, to select a fan. Such practices are faulty inasmuch as they do not take into account the number of blades, the type of blades, and the inlet/outlet conditions. This is explained later in this article.

The noise characteristics of various types of fans are reasonably predictable, and if the fans are well designed, the noise characteristics are not significantly affected by minor changes in the fan geometry. If a well-designed particular application, it is generally useless to experiment with changes in the fan configuration. If the changes reduce the noise level, hey also will change the fan performance characteristics so that they no longer meet the specified conditions. The engineering approach to controlling duct transmitted noise is to provide sufficient sound attenuation in the duct installation to meet the specified acoustical requirements.

The performance requirements for fans used in heating, ventilating and air conditioning systems are relatively easy to meet, since none of the severe specifications mentioned above (such as corrosion) are a part of the selection criteria. Therefore, the selection can be made on the basis of noise characteristics. Incidentally correct installation of the fan is also very important and recommended inlet and outlet duct connections should be used.

Prediction of Fan Sound Power

The sound power generated by fan performing a given duty is best obtained from manufacturer's test data taken under approved (ASHRAE Standard 68- 1986; also AMCA Standard 300-1986). However, if such data are not readily available, the octave-band sound power levels for various fans can be estimated by the procedure described here.

Fan noise can be rated in terms of the specific sound power level, defined as the sound power level generated by a fan operating at a specific capacity and pressure. The specific capacity chosen is 1m3/s & the pressure 1 Pa. By reducing all fan noise data to this common denomination, the specific sound power level serves as a basis for direct comparison for the octave-band levels of various fans and as the basis for a conventional method of calculating the noise levels pf various fans and as the basis for a conventional method of calculating the noise level of fans at actual operating conditions.

On a specific sound power level basis, small fans are noisier than large fans. While size division is necessarily arbitrary, the size division indicate are practical for estimate ting fan noise. At present, the sound power level data for forward curved fans varies widely; the specific sound power levels given in Table 1 are an average of that data. For critical applications, the sound power levels for a practical fan should be obtained from the manufacturer.

Sound power levels at actual operating conditions can be estimated by using the corresponding fan volume flow rate and fan pressure given by:

$$L_w = K_w + 10 \log (Q/Q_1) + 20 \log (p/p_1) + D \quad (1)$$

Where$_1$ L_w = estimated sound power level of fan, (dB re 1 pW),

K_w = specific sound power level (see **Table 1**),

Q = flow rate, m^3/s,

Q1 = 1 m^3/s

P = pressure drop, Pa,

P_1 = 1 Pa, and

C = correction factor for point of fan operation, dB

Values of the estimated sound power level are calculated for all eight bands, and the BFI (Blade Frequency Increment) is added to the sound pressure level in the octaves band in which the blades passage frequency falls.

Table 1 : Specific Sound Power Levels of Typical Fans

Fan Type	**Octave Band**								
	63	125	500	1K	2k	4K	8K	BK	BFI
Centrifugal									
AF,BC, or B1									
Wheel diameter									
Over 900 mm	40	40	39	34	30	23	19	17	3
Under 900 mm	45	45	43	39	34	28	24	19	3
Forward Curved									
All wheel diameters	53	53	43	36	36	31	26	21	2
Radial Bladed									
Low pressure									
(1 to 2.5 kPa)	56	47	43	39	37	32	29	26	7
Medium pressure									
(1.5 to 3.7 kPa)	58	54	45	42	38	33	29	26	8
High pressure									
(3.7 to 15 kPa)	61	58	53	48	46	44	41	38	8
Axial Fans Vaneaxial									
Hub ratio 0.3 to 0.4	49	43	43	48	47	45	38	34	6
Hub ratio 0.4 to 0.6	49	43	46	43	41	36	30	28	6
Hub ratio 0.6 to 0.8	53	52	51	51	49	47	43	40	6
Tubeaxial									
Over 10000 mm									
Wheel diameter	51	46	47	49	47	46	39	37	7
Under 1000 mm									
Wheel diameter	48	47	49	53	52	51	43	40	7
Propeller									
General ventilation	48	51	58	56	55	52	46	42	5

Blade Frequency Increment (BFI)

Fans generate a tone at the blade passage frequency, and the strength of this tone depends partly on the type of fan. To account for this blade passage frequency, an increase in sound pressure should be made in the octave band in which the blades frequency falls. The number of decibels added to the sound pressure level in this band is the blade frequency increment BFI. The blade frequency Bf is:

Bf = rps x No. of Blades or rpm x No. of blades / 60 (2)

The number of blades and the fan rpm (rps) can be obtained from the fan selection catalog. **Table 1** lists specific sound power levels and blade frequency increments. **Table 2** lists the octave band in which the BFI occurs.

The narrowband tones that a fan generates are extremely important in the noise control of a fan system. In many noise problems related to fans, the major or problem is the discrete frequency component contributed by the blade frequency. Special emphasis should be given to the octave and containing the blade frequency component, since the ear has the ability to identify a tone in a general noise background. Therefore, one tends to be annoyed by it. The attention characteristics of the system must be completely adequate in that octave band.

Table 2: Octave Band of Blade Frequency Increment

Fan Type	Octave Band
Centrifugal	
Airfoil, backward curved, backward inclined	125-250
Forward curved	500
Radial blade, pressure blower	125
Vaneaxial	125-250-500
Tubeaxial	63
Propeller	63

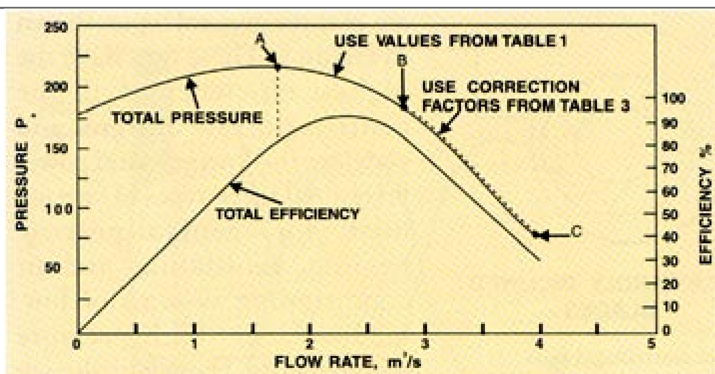

Figure 3 : Optimum Selection Zone for a Centrifugal Fan

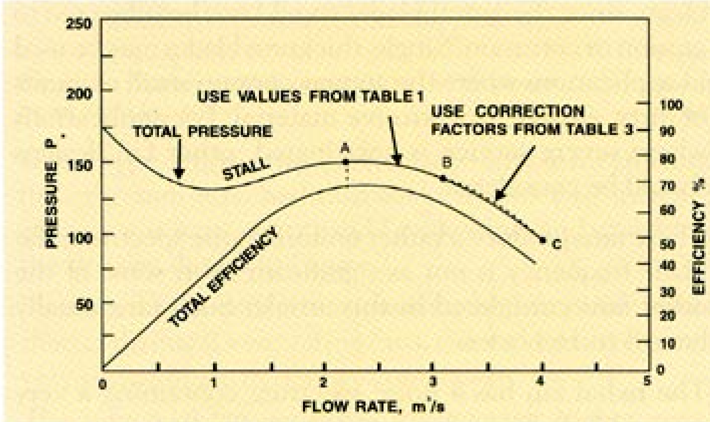

Figure 4 : Optimum Selection Zone for an Axial Fan

Table 3: Correction Factor C For Off-Peak Operation

Percentage of Peak state Efficiency	**dB Correction Factor**
90 to 100	0
85 to 89	3
75 to 84	6
65 to 74	9
55 to 64	12
50 to 54	15

Table 4: Comparative Sound Power Levels

Type of Fan	**Specific sound power level, K_w**									**Total Power Level**
	63	125	250	500	1k	2K	4K	8Khz	Total	Lw dB
Centrifugal AF,BC, or BI	45	45+3	43	39	34	28	24	19	51.1	121.1
Centrifugal, forward curved	53	53	43	36+2	36	31	26	21	56.3	126.3
Centrifugal, radial bladed	56	47+7	43	39	37	32	29	26	58.3	128.3
Vaneaxial	49	43	43+6	48	47	45	38	34	53.8	123.8
Tubeaxial	48+7	47	49	53	52	51	43	40	59.2	129.2
Propeller	58+5	51	58	56	55	52	46	42	62.8	132.8

Point of Operation

The specific sound power levels given in **Table 1** are for fans operating at or near the peak efficiency point. The fans performance curves for centrifugal and axial fans are respectively shown in **Figure 3 & 4**. The fans in general must operate at their peak efficiency. The conforms with the recommended practice of selecting fan size and speed so the operation falls at or near this point, it is advantageous for energy conservation and corresponds to the lowest noise levels for that fan. If, for any reason, a fan is not or cannot be selected optimally, the noise level produced will increase; correction factor C in equation (1) accounts for this increase. This correction factor should be applied to all octave bands. For off-peak operation, the corrections in decibels are shown in **Table 3.** In order to use this table first obtain the ratio of the efficiency at which it is operating and the responding peak efficiency at which it sis operating and the corresponding peak efficiency, using **Figures 3 and 4**. Then obtain the correction factor C from **Table 3**.

A feel of the relative noisiness of different types of fans can be had from Table 4 where the total sound power levels computed from **Tables 1 and 2** for a fan-delivering 10m^3/s (or 10,000 liters/s) of air against a static pressure of 1000 Pa (10 cm of water column) are compared. It is assumed that the wheel diameter is within 900 mm and the hub ratio is small 9 (about 0.3) It may be noted that for this typical case, the centrifugal blower with airfoil blades, or backward curved blades or backward inclined blades, is likely to be the quietest, and he propeller type fan may be the noisiest fan. In the axial types of fan, vaneaxial may be noted to be the quietest as well as most efficient from the aerodynamic point of view.

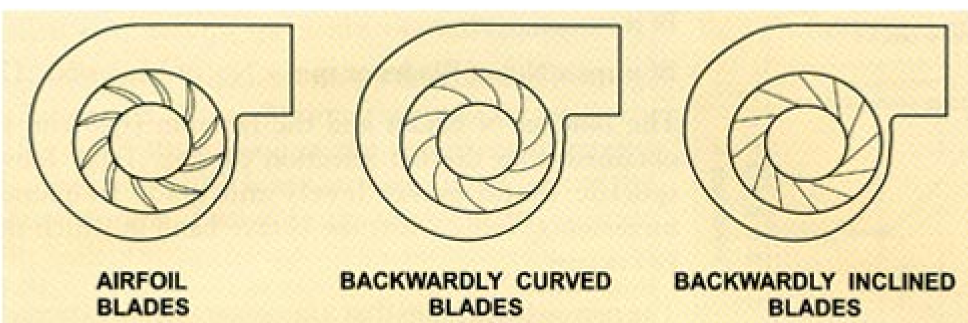

Figure 5 : The three principal types of high-efficiency, low-noise-level centrifugal fans

Centrifugal Fans

A centrifugal fan moves air and creates pressure partly by centrifugal action and partly by rotating velocity. The centrifugal forces are created by a rotational air column which is enclosed between the blades of the fans.

The lower value of the net flow velocity for the backward curved fan is a desirable characteristic both from an efficiency standpoint and from a noise standpoint. However, it is not always possible to use this type of fan. Each type of fan has different pressure and power characteristics which make it the logical choice for a given application.

The scroll of a centrifugal fan may take the form of an involute where the axial clearance increases directly in proportion to the angle traversed, reckoned from the neck or cut-off point (see **Figure 1**) This is required for a constant mean velocity of flow around the scroll, under the assumption of a uniformly distributed air supply from the blade-discharge area. Such a scroll is desirable for minimum noise generation. If the scroll clearance increases more rapidly, the velocity at the exit is reduced and the pressure is increased. While increased pressure may be desirable for certain applications, it causes an abrupt pressure change at the cutoff and thus serves to a increase the noise at the blade frequency. In fact, increased velocity at any stage, or across a part of the cross-section, increases overall aerodynamic noise generation as a consequence of anti-logarithmic addition of noise levels.

The cutoff clearance also is critical. In general, the smaller the clearance, the greater the rotational noise that is generated at the cutoff. A clearance of 5 to 10 percent of the wheel diameter generally is considered o be the optimum range; a lesser clearance would increase the rotational noise while a larger clearance would affect the aerodynamic efficiency.

Centrifugal fans of the backward-inclined blade design are of three general types, shown, in **Figure 5**. These fans have the highest efficiency of all the centrifugal fan designs and generate the lowest sound power levels. All three types of fans are used for general-purpose heating, ventilating, and air conditioning systems of low, medium, or high-pressure requirements. These fans also are used for clean-air industrial applications. The airfoil must be used only in those applications where the air is quite clean, since the airfoil blades should not be subjected to erosion or corrosion. Single thickness blades may be used of dirty, erosive, or corrosive material. For applications where severe service is anticipated, other fan designs should be considered.

The radial fan has a noise spectrum; the blade frequency is not as significant as in some of the other fans considered in this article. Such fans usually have 6 to 12 blades.

The forward-curved fan shown in **Figure 5** produces a higher noise level than the backward-inclined bladed fans, but the blade frequency component is low. Such fans usually have 36 to 64 blades.

There are many different wheel designs within the forward-curved classification, which results in a broad range of specific sound power levels. Therefore, it is advisable to obtain actual fan noise data on the design under consideration.

Fans with forward-curved blades are used in packaged air-conditioning equipment where space is at a premium. In many cases this results in poor fan inlet and/or fan outlet flow conditions. The fan performance is degraded, and a higher noise level is generated. The unit must be very carefully designed, and sufficient space must be allowed for good airflow to make these devices acceptable. Extra space is needed for ensuring uniform inflow and unconstrained outflow as also for housing acoustic attenuators.

Axial Fans

Axial-flow fans (see **Figure 2**) impart energy to the air by giving it a twisting motion, as shown in **Figure 6**. This vortex type of flow is not particularly suitable for efficient flow through attached ductwork; in order to increase efficiency, it is necessary to add guide vanes and straighten the flow, as shown in **Figure 7**.

Vaneaxial fans generate somewhat higher noise levels than centrifugal ventilating fans; their spectra contain a very; strong blade frequency component.

The tubeaxial fan generates a somewhat higher noise level than the vaneaxial fan; its spectrum contains a very strong blade frequency component.

The noise levels of the propeller fan are only slightly higher than those of the tubeaxial and vaneaxial fans, but such of the noise is at low frequencies and therefore is difficult to attenuate.

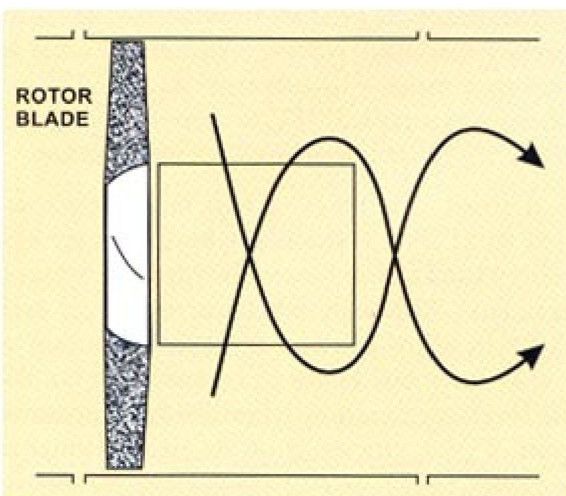

Figure 6 : Vortex motion imparted to the airstream by an axial fan rotor

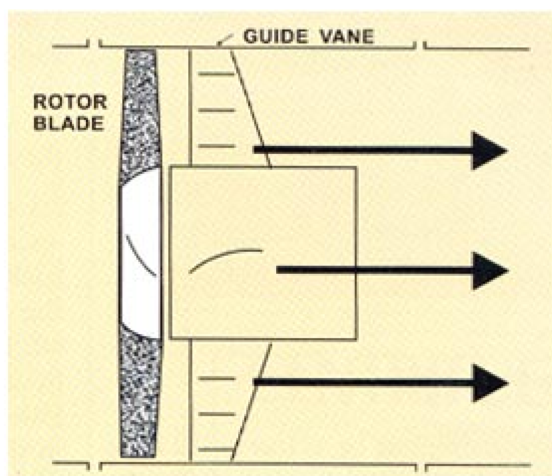

Figure 7 : Guide vanes straighten the vortex flow and increase fan efficiency, which result in lower noise levels

Sound Attenuators for Fans

If lower sound power levels are required than are generated by well-designed fans it is necessary to add attenuation to the system. This may be provided by sound attenuators (also called sound traps) installed as separate unit in the field or by sound attenuators installed as an integral part of the fan assembly.

Figure 8 shows a centrifugal fan, with sound attenuators on both the inlet and the outlet, used in the supply system of a central station ventilating system. The outlet of the fan is fitted with a sound attenuator to reduce the flow of acoustic energy from the discharge of the fan to the supply air ductwork. The required attenuation of this attenuator is determined by using the sound power levels of the fan (without attenuator) to calculate the sound level that would result in the room with the most critical noise criterion. The amount by which the calculated sound level exceeds the allowable sound level determines the required amount of attenuation than the fan attenuator must finish. There is no "rule of thumb" for selecting such attenuator must be selected on the basis of actual requirements as described in Reference. The sound attenuator on the fan intake system shown in **Figure 8** is used to reduce noise radiated on the inlet sides of the fan (outsides the fresh air intake) to an acceptable value. This attenuator is selected in the same way as the attenuator used in the fan discharge. However, here, a different criterion must be used, often such a criterion specifies the A-weighted sound level that must not be exceeded at either the property line of the building or at the nearest adjacent building or residence.

Figure 9 shows attenuators attached directly to both the upstream and downstream sides of an axial flow fan; the sound attenuators and the fan are treated as a single unit which is separated from both the inlet and discharge ductwork by flexible collars. The entire unit is supported on an inertia block which in turn, is isolated from the building structure by vibration isolators.

Most sound attenuators for fan noise are of the absorption type because this type (see **Figure 10**) has broad band attenuation characteristics. Generally d=h=50 mm. In special designs, however, d/h may be as low as 0.2 or as larger as 5.0. In general, the larger the value of d/h, the wider will be the attenuation spectrum, making the attenuator applicable over larger frequency bands. Resonator-type attenuators are sometimes used in special installations but are less widely used than the absorption type attenuators, since they must be custom-designed and may require tuning under field conditions.

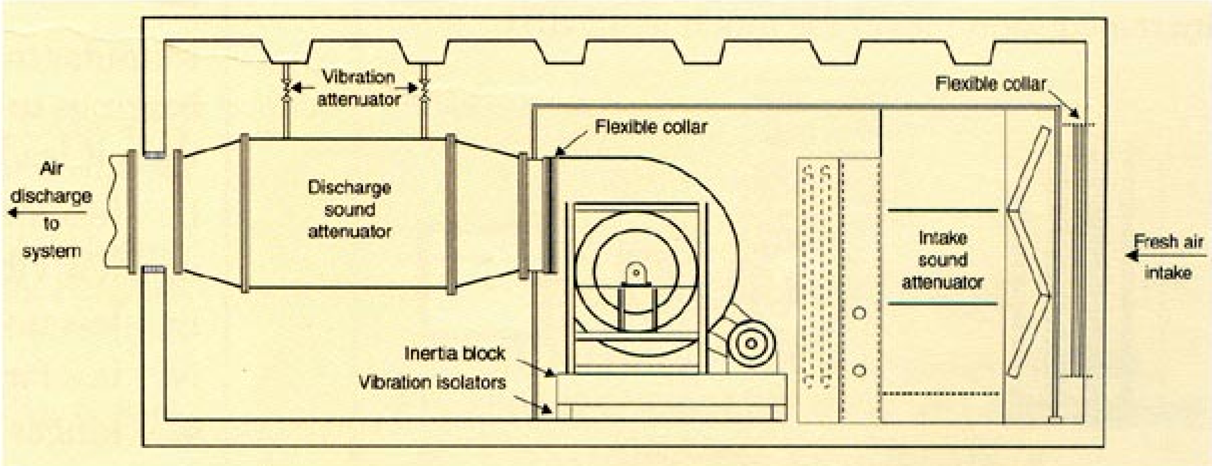

Figure 8 : Sound attenuation system for a centrifugal fan installation

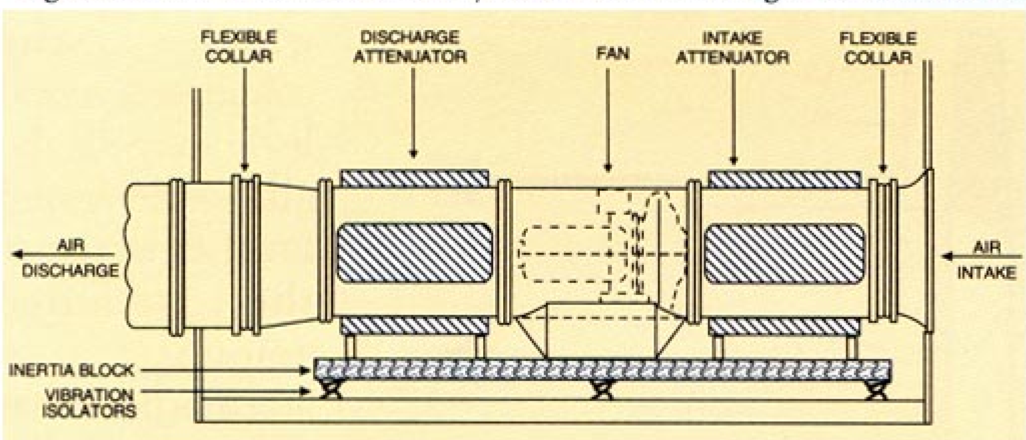

Figure 9 : Sound attenuation system for an axial-flow fan installation

Designing and Operating for Quietness

It is especially important that both centrifugal fans having forward-curved blades and axial-flow fans operate in the optimum selection zone. It is essential to select a fan so that its operating point does not fall on the stall region; operation in the stall region may result in a noise level 8 to 10 dB higher than the level that results from operation in the optimum selection zone. Furthermore, there is a possibility of "surging" which will generate a varying noise level that is very objectionable. Surging may also result where fans are connected in parallel land operate in the stall region; then the load a will shift back and forth between the two fans, and a varying noise level will result.

In addition to selecting a fan whose performance falls within the optimum selection zone, it is important to provide good inlet and outlet conditions for the operation of a fan. Fan inlet conditions are very critical-the penalty for poor inlet conditions is reduced performance and increased sound level (as much as 10 dB)

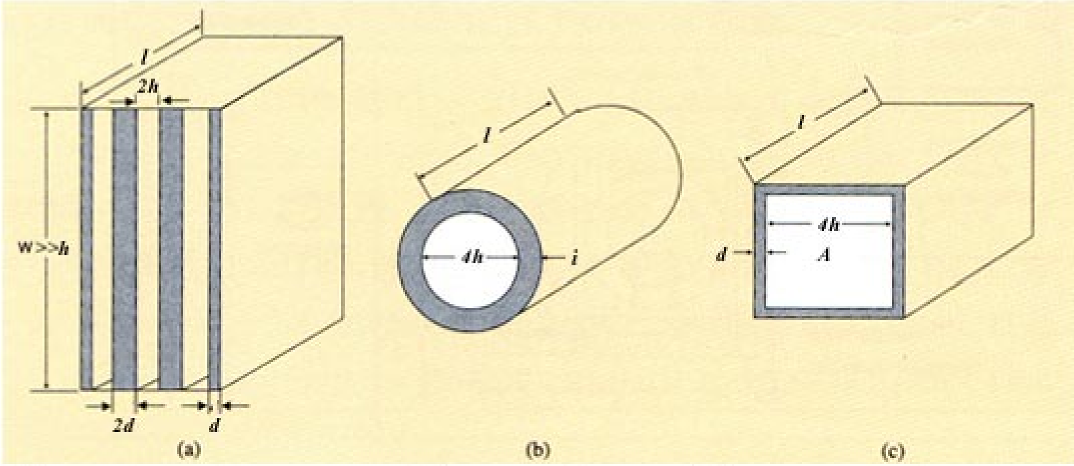

Figure 10 : Geometry of frequently used dissipative silencer types : a) parallel baffle b)round; c) lined ducts

Centrifugal fans generally produce noise across the entire audio-frequency range - usually from 31.5 to 250 Hz. As a result, fans in a typical HVAC system usually set the low-frequency noise limits in an occupied space.

As discussed above, the types of fans most frequently encountered in HVAC system for buildings are the backward-curve and forward-curved centrifugal fans and the vaneaxial fans. Although the characteristics of each fan type differ in terms of noise level as a function of frequency, they have one factor in common; by far, the lowest noise levels are produced when the fan is operated in the region of peak efficiency in it air performance curve (see **Figure 11**).

Thus the airflow at the entrance and exit of a fan should be as smooth as possible to minimize the generation of turbulence; turbulence results in the generation of noise and an increased static pressure drop in the system. For this reason, fittings (such as elbows and transitions) should not be placed closer than 3 to 6 duct diameters downstream from a fan. This is illustrated in **Fig. 12**, which shows examples of good and bad airflow conditions a the outlet of the fan; the noise generation in the latter case may be 10 to 30 dB higher than the former case.

For an HVAC system having a constant volume of airflow, if the airflow conditions are uniform, the fans generally are operated at close to their maximum efficiency. However, for a variable-air-volume system, where volume controls are installed at the fan intake, the operating efficiency often is well below the maximum static efficiency2.

Finally, in a fan or blower installation, it is very important to mount the fan so as to isolate it dynamically from the ducting as well the building structure. Otherwise, the latter would be set into vibration. This structure-borne sound can travel long distances and radiate lot of noise thereby acting as a mechanical loudspeaker.

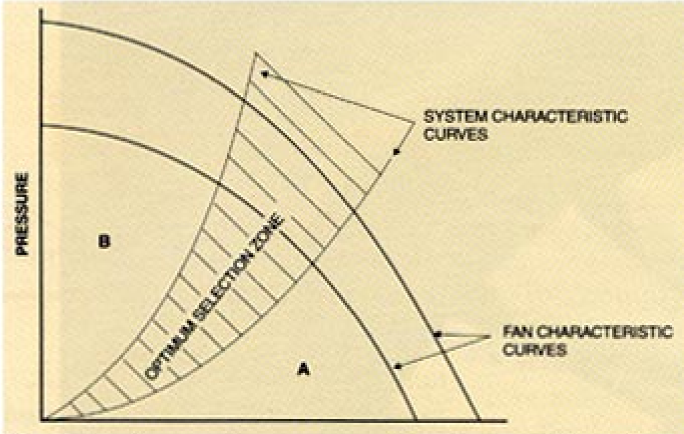

Figure 11 : A curve of pressure vs. volume flow rate, illustrating an optimum selection zone of a fan with changes in speed. A larger fan (which operates in area A) or a smaller fan (which operates in area B) is less effecient and radiates a higher noise level

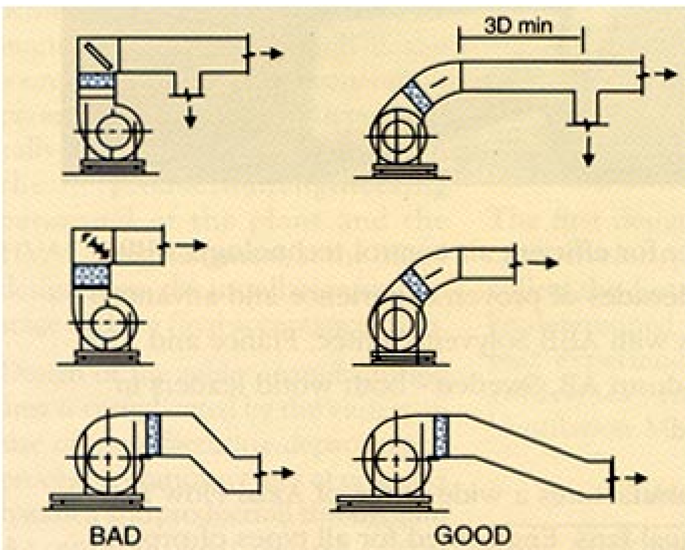

Figure 12 : Examples of good and bad fan outlet conditions

Manufacturing for Quietness

Manufacturing the mating parts or components like bearings to closer tolerances is costlier and difficult, but then it leads to

i. Less impacting of mating surfaces,
ii. Less vibration
iii. Less noise radiation,
iv. Less fatigue stresses,
v. Longer intervals between breakdowns,
vi. Longer life for mating components.

And therefore it is more cost-effective in the long run.

Particular care also needs to be taken in balancing the impeller assembly.

For further reading, the reader may refer to Trane's Application Engineering Manual4.

References

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