11. FAN NOISE PREDICTION

The sound power produced by centrifugal and axial fans can be approximated by a simple equation (ref. ASHRAE Handbook)

$$L_W = K_W + 10 \log_{10} Q + 20 \log_{10} P + BFI + C_N$$

where:

L_W= sound power level (dB)

K_W = specific sound power level depending on the type of fan (see Fig 9-3), from empirical data provided by fan manufacturer

Q = volume flow rate (cfm)

P = total pressure (inches of H_20)

BFI = Blade Frequency Increment = correction for pure tone produced by the blade passing frequency (bpf) from Fig 9-3, add this correction only to the octave band whose center frequency is closest to the blade passing frequency.

bpf= blade passing frequency = #of blades $\times$ RPM / 60 (Hz)

C_N = efficiency correction (because fans that are operated off their optimum flow conditions get noisier)

$C_N = 10 + 10 \log_{10} (1-\eta)/\eta$ typical values:

η	Cn
90%	0
75%	5.2
40%	12.2

η= Hydraulic efficiency of the fan = $Q \times P / (6350 \times HP)$

HP = nominal horsepower of the fan drive motor

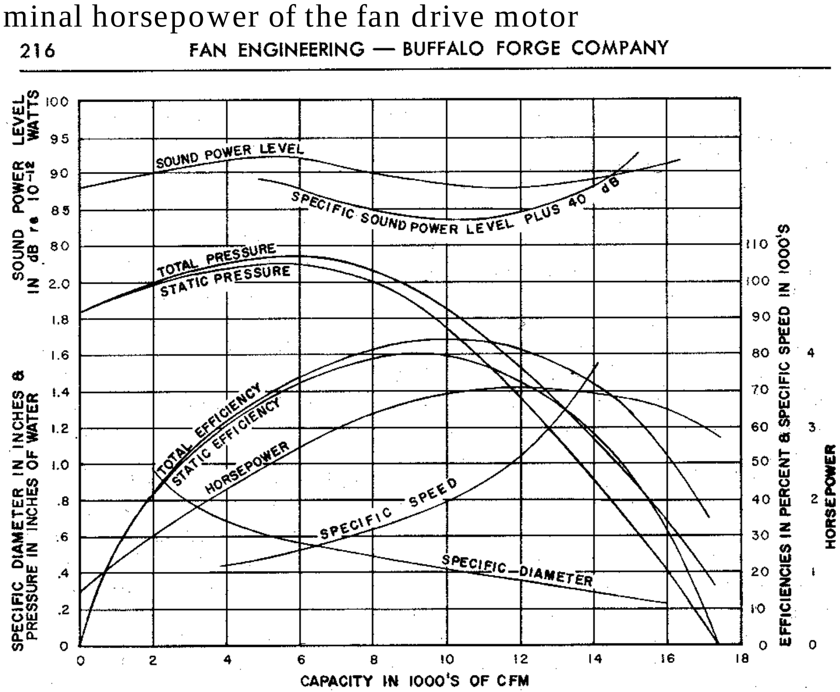


Figure 95 —Typical Constant Speed Performance Curves for Ventilating Fans

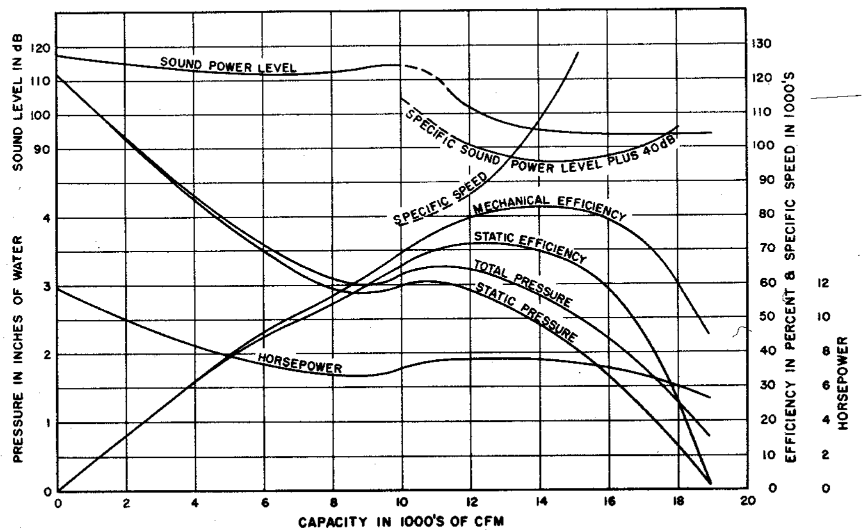


Figure 102 —Typical Constant Speed Performance Curves for Vaneaxial Fans

Fan Application

The choice of a fan depends on the desired ventilation requirements (volume, pressure, density, and speed) and other considerations including noise, initial cost, operating costs, environment, etc. Aerodynamic selection of type and size can be done with the aid of charts such as Figure 95 and 102 (ref. Fan Engineering, Buffalo Forge, 1970). Figure 17 (ref. Handbook for Mechanical Engineers, Baumeister and Marks) compares data for various commercial fan types. Specific diameter (D_S) and efficiency vs. specific speed (N_S) are shown. p=pressure drop (inches water), D=fan wheel diameter (ft), d= density of air (.075 lb/ft^3 at standard temperature and pressure), Q = flow volume (cfm), RPM = fan rotational speed (rpm).

$$N_S = \frac{RPM\sqrt{Q}}{(p/d)^{3/4}} \qquad D_S = \frac{(p/d)^{1/4} D}{\sqrt{Q}}$$

Generally, efficiency increases and fan size decreases as specific speed increases. This figure can be used to determine the most efficient size and type of fan for a particular application.

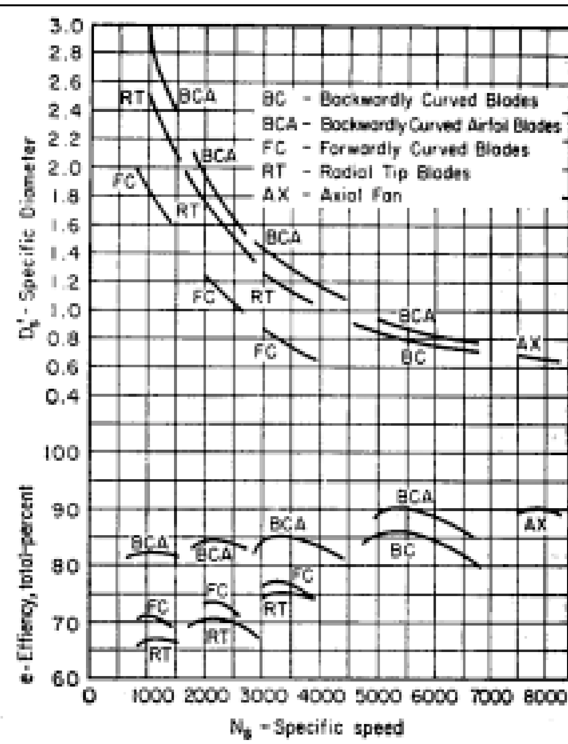


FIG. 17. Specific speed, specific diameter and efficiency for commercial single-inlet fan types.

BC—backwardly curved blades
BCA—backwardly curved airfoil blades
FC—forwardly curved blades
RT—radial-tip blades
Ax—axial fan
$D_s' = D_s \times 12$

Noise Comparison

For lowest noise output, fans should always be operated near their peak efficiency point. A common mistake is to use a fan that is too small (or too large) for the application, so that it cannot be run at its most efficient point. Variable airflow applications can also cause noise problems. The cheapest way to achieve variable volume (and the noisiest) is with VAV (Variable air volume) units, which basically throttle the flow with louvers. A better way from a noise standpoint (but more expensive), is a variable speed motor drive. The typical noise characteristics of various fans are compared in Table 1 below.

Table 1. Comparison of noise from various fan types

Fan Type	**Noise (broad band)**	**Blade passing tone**	**Flow**
Centrifugal			
Airfoil blades	Lowest	Moderate	Very efficient
Backward Inclined Blades	Lower	Moderate	
Forward Inclined Blades	Moderate	Lowest	Low pressure drop applications
Radial Blades	High	High	
Axial			
Vane	Higher than centrifugal	Can be high, depends on flow obstructions	Very efficient
Tube	More than vane	“	
Propeller	Highest	“	

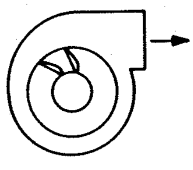
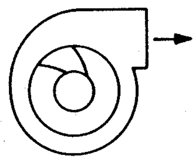
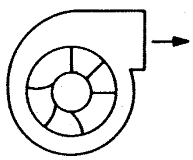
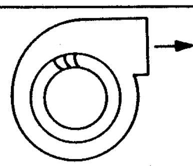
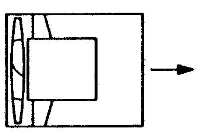
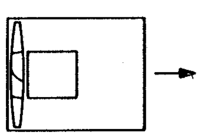
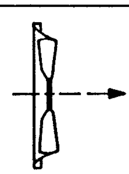
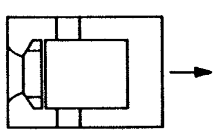
	TYPE	DESIGN	SPECIFIC SOUND-POWER LEVEL, K_w — CENTER FREQUENCY–Hz (dB re 10^{-12} watt and 1 cfm at 1 inch ftp)								BFI	APPLICATIONS
			63	125	250	500	1000	2000	4000	8000		
CENTRIFUGAL FANS	AIRFOIL		35	35	34	32	31	26	18	10	3	Highest efficiency of all centrifugal fan design contains 10 to 16 blades of airfoil shape. Used for general heating, ventilating, and air-conditioning systems, usually applied to central station units where the horsepower saving will be significant. Can be used on low, medium, and high-pressure systems and will operate satisfactorily in parallel. Is also used in large sizes, for clean-air industrial applications where power savings will be significant. Can be used on industrial exhaust systems, where the air-cleaning system is of high efficiency.
CENTRIFUGAL FANS	BACKWARD INCLINED BACKWARD CURVED		35	35	34	32	31	26	18	10	3	Efficiency is only slightly less than that of the airfoil fan. Contains 10 to 16 blades. Used for the same general applications as the airfoil fan. Can be used in industrial applications where the gas is essentially clean, but does not meet the standards required for airfoil fan selection.
CENTRIFUGAL FANS	RADIAL		48	45	43	43	38	33	30	29	5–8	Simplest of all centrifugal fans–relatively low efficiency, usually has 6 to 10 blades; includes both radial blades (*R*), and modified radial blades (*M*). Used primarily for industrial exhaust, including dirty gas fans and recirculating gas fans. This design also used for high-pressure industrial applications.
CENTRIFUGAL FANS	FORWARD CURVED		40	38	38	34	28	24	21	15	2	Efficiency less than the airfoil and backwardly curved fans, this fan is usually fabricated of lightweight and low-cost construction. It may have from 24 to 64 blades. This design will be the smallest of the centrifugal fan types and operates at the lowest speed. Used primarily in low-pressure heating, ventilating, and air-conditioning applications, such as: domestic furnaces, small central station units, and packaged air-conditioning equipment.
AXIAL FANS	VANEAXIAL		42	39	41	42	40	37	35	25	6–8	High-efficiency axial flow fan with airfoil blades and high pressure capability. Blades may be fixed or adjustable and the hub diameter is usually greater than 50 per cent of the fan tip diameter. There may be from 3 to 16 blades. This fan design has guide vanes downstream from the wheel which permits good air flow pattern on the discharge side of the fan. Used for general heating, ventilating, and air-conditioning applications in low, medium, and high-pressure systems. May also be used in industrial applications such as: drying ovens, paint spray booths, and fume exhaust systems.
AXIAL FANS	TUBEAXIAL		44	42	46	44	42	40	37	30	6–8	This fan is more efficient than the propeller fan design and can develop a more useful pressure capability. The number of blades may vary from 4 to 8 and the hub is usually about 50 per cent of the fan tip diameter. The blades may be of airfoil or single thickness cross-section. The fan is built without downstream guide vanes. Used in low- and medium-pressure ducted heating, ventilating, and air-conditioning applications where the poor air flow pattern downstream from the fan is not detrimental. This fan is also used in some industrial applications such as: drying ovens, paint spray booths and fume exhaust systems.
AXIAL FANS	PROPELLER		51	48	49	47	45	45	43	31	5–7	Low efficiency wheels are usually of inexpensive construction and are limited to very-low-pressure applications. Usually contains 2 to 8 blades of single thickness construction attached to a relatively small hub. The housing is a simple circular ring or orifice plate. This fan is used for low pressure, high-volume air-moving applications such as air circulation within a space or as exhaust fans in a wall or roof.
AXIAL FANS	TUBULAR CENTRIFUGAL		46	43	43	38	37	32	28	25	4–6	This fan usually has a wheel similar to the airfoil or backwardly inclined wheel, described above, which is built into an axial flow type housing. This results in lower efficiencies than the centrifugal fans of similar wheel design. The air is discharged radially from the wheel and must change direction by 90 degrees to flow through the guide vane section. Used primarily for low-pressure return-air systems in heating, ventilating, and air-conditioning applications.

Fig. 9-3. Acoustic properties of various fan types.

Fan Laws: Size and Speed

Fan performance can be predicted over a wide range of sizes and speeds using basic scaling relations (ref. Handbook of Acoustical Measurements and Noise Control, by C. Harris, 1991).

$$Q_a = Q_b(d_a/d_b)^3(n_a/n_b) \quad (41.3)$$

$$P_{ta} = p_{tb}(d_a/d_b)^2(n_a/n_b)^2 \quad (41.4)$$

$$P_a = p_b(d_a/d_b)^5(n_a/n_b)^3 \quad (41.5)$$

$$L_{Wa} = L_{Wb} + 70 \log_{10} (d_a/d_b) + 50 \log_{10} (n_a/n_b) \quad (41.6)$$

where Q = volume flow rate, m^3/s
p_t = total pressure, kPa
P = fan power, kilowatts
L_W = sound power level, decibels re 1 picowatt
d = rotor diameter, meters
n = rotor speed, number of revolutions per minute

Note the additional subscripts:

a = data at required performance conditions
b = data at base curve performance conditions

Although Eq. (41.6) is less accurate than the equations which predict other performance characteristics, it is sufficiently accurate for estimating purposes.

The fan laws express mathematically that when two fans are both members of a similar series, their performance curves are similar, and at the equivalent point of rating on each performance curve, the efficiencies are equal. The fan laws can be applied only to one point of rating on the fan curve for each calculation; this point can be used to calculate only the equivalent point on any new speed curve. To define the new speed curve accurately, it is necessary to use enough individual base curve data points to calculate the new curve so that the new curve will be defined with a minimum of interpolation error between the chosen points.

To apply the fan laws, it is necessary to have actual test data for one fan in the same series. The use of the fan laws is restricted to cases where the linear dimensions of the larger or smaller fans are all proportional to the fan for which there are test data.

A misunderstood parameter in fan noise generation is the speed of the fan. *For a given type of fan, volume flow rate, and pressure, there is one particular size that is more efficient than all other sizes.* Neither a larger size nor a smaller size can be more efficient. It is a common misconception that if a larger fan is used which runs at a slower speed, the noise will be reduced. This is in error; the larger fan does run at a slower rotational speed, but it does not operate at peak efficiency. Therefore, the noise is greater. While the larger fan is running at a slower speed, the outer diameter of the larger rotor must run at the same velocity as the outer diameter of the smaller rotor in order to develop the required pressure. The specified pressure cannot be developed at lower velocities. Therefore, the noise-generating mechanism is the same, and the noise produced by the larger fan at lower speed is not less than that produced by the smaller fan. The change in efficiency will result in higher noise levels for the larger fan. *Once the optimum size of fan for a particular application is determined, it is not possible to reduce the noise levels by using a larger and slower-speed fan.*

An exception to the above statement may occur if the speed change is sufficient to lower the blade frequency significantly; since the human ear responds more poorly at lower frequencies (see Fig. 17.1), there is a reduction in loudness