ENGINEERING LETTER | 12

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AN INTRODUCTION TO FAN ACOUSTICS

INTRODUCTION

Fan Acoustics is an important consideration in the industrial environment and with commercial ventilation systems. The sound generated by some fans can be a potential hazard to personnel in close proximity to the fan, and the sound can be transmitted, via the ductwork connected to the fan, to all areas serviced by the fan. Because of these concerns, fan manufacturers publish sound ratings for their products to serve as a guide for selecting fans to meet sound specifications, and to assist acoustical consultants in predicting the total noise levels in specific environments. This Letter provides basic information to help understand fan sound ratings and how to apply them.

WHY FANS MAKE NOISE

Like any mechanical device, fans generate sound, which emanates naturally from the turbulence of moving air, the mechanics of moving parts of the fan, and from vibration.

AIR TURBULENCE

Air turbulence within the fan increases the sound coming from the air movement. The noise resulting from air turbulence is a major factor in the sound levels of a fan in a specific application. Further, duct work can transmit this turbulent noise to all areas serviced by the fan.

Factors contributing to air turbulence include the resistance to flow, flow separation along fan surfaces, and shock related to abrupt changes in the direction of airflow, pressure, or velocity. The principal areas where such turbulence is encountered within a fan are shown in Figure 1.

A lower noise level can be achieved by reducing air turbulence. This can be done by considering several factors related to air movement when selecting fans.

The first factor to consider is the fan's *blade pass frequency,* which is a pure tone produced when the blades of the fan wheel (impeller) rotate past the housing cut-off sheet in centrifugal fans, or the turning vanes, in axial fans. The blade pass frequency is calculated by multiplying the number of blades times the rotating speed in revolutions per minute. If this frequency matches the natural frequency of the ductwork, it can excite the ductwork, which can cause it to resonate, thereby increasing the noise level.

Because of this possible increase in sound, and because certain pure tones are irritating to people, the sound output of the blade pass frequency should be investigated when sound reduction is desired.

The next factor to consider is the fan design. Generally a fan operating at peak mechanical efficiency will produce less noise, because high efficiencies result from minimal air turbulence within the fan.

There are four basic centrifugal fan wheel designs - forward curved, backwardly inclined, radial, and radial tip - and a variety of axial flow wheel designs (see Figure 2). Each wheel design has unique sound characteristics due to the way they handle air, and the efficiencies they can achieve.

Fan speed does not always determine which fan will be quieter. For example, centrifugal fans have higher amplitudes at lower frequencies, while axial fans exhibit higher amplitudes at the higher frequencies.

The amplitude of the blade pass frequency on an axial fan is higher and more pronounced than on backwardly-inclined fans, and commonly will have amplitude peaks at multiples of this frequency.

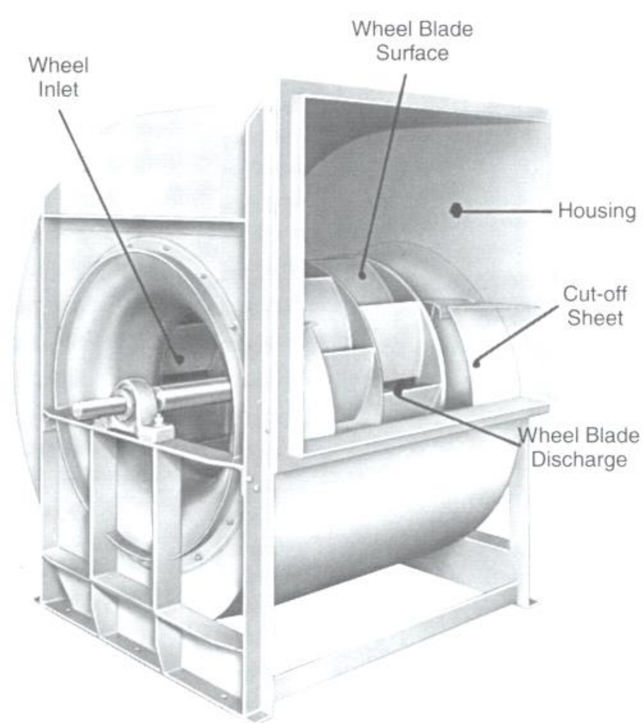


Figure 1 - Typical Areas of Turbulence

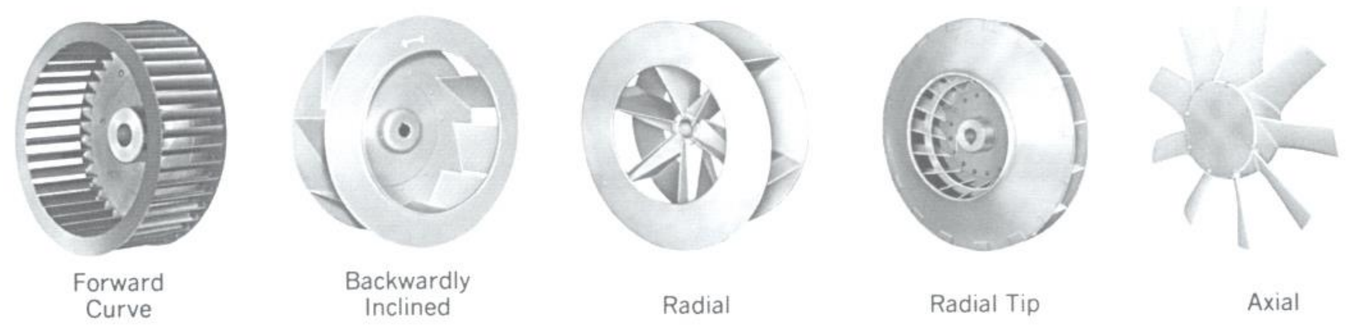


Figure 2 – Various Wheel Types

Of the four centrifugal designs, the backwardly-inclined fans are the most efficient, and therefore, the quietest. Those with airfoil-shaped blades offer the highest efficiencies, for clean air environments, while those with single-thickness blades can be used in applications where light dust or moisture is present, although the efficiencies are somewhat lower.

Certain types of axial fans offer the next highest efficiencies. An excellent example is the **nyb** Vaneaxial fan that uses airfoil shaped blades in an in-line flow design. This fan is used to handle high volumes of clean air at low pressures, which is a typical ventilation application.

Radial fans are typically low efficiency, open designs for special purpose applications, such as bulk material handling, or exhausting/supplying lower volumes of air at higher pressures. An exception to this is the **nyb** DH design (Figure 3), which has superior efficiencies for a radial wheel and relatively low sound levels. A radial fan will be much louder than a backwardly-inclined fan operating under the same volume and pressure conditions.

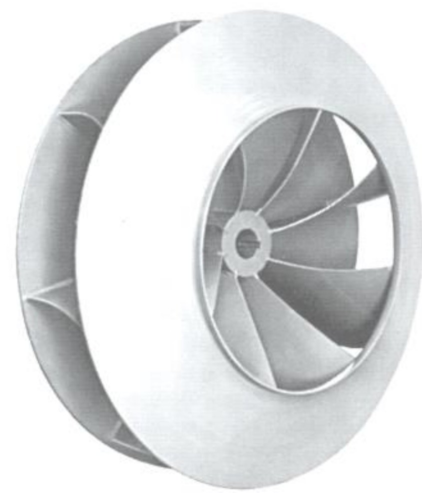


Figure 3 – DH Wheel

Radial-tip fans, commonly used to handle larger volumes of air that contains particles or material, exhibit sound characteristics similar to the radial fans.

The sound spectra of radial and radial-tip fans contain amplitude spikes at various frequencies, and a noticeable spike at the blade pass frequency.

The forward-curved fan design operates at speeds that are much slower than the other fan types, which results in lower noise levels from mechanical operation and vibration. However, because of its modest efficiencies, a forward-curved fan may be noisier than a backwardly-inclined fan when operating at comparable volume and pressure. The sound spectrum of the forward-curved fan shows a slower rate of reduction in amplitudes than the other centrifugal types, and because of the large number of blades, the blade pass frequency occurs much later in the spectrum and is not predominant.

MECHANICAL NOISE

The moving components of the fan - the motor, bearings, and drive - produce sound. This too can be transmitted through the system via the fan structure or shaft, or when these components are in the airstream. Motor sound will vary with speed, enclosure, electrical characteristics, and even the manufacturer. Antifriction bearings can be used to reduce bearing noise, and proper drive selection will reduce the likelihood of belt hop, or slap. Of course, proper maintenance must be employed to keep the moving parts running smoothly, and quietly.

VIBRATION

Excessive vibration can significantly add to the overall noise level of an installation. This will occur if the fan or any of its components are not adequately balanced, if the fan is installed on an insufficient foundation, or if the fan is not properly isolated from other system components. For example, it is not uncommon for the fan's support structure or ductwork to have a natural frequency at the fan's operating speed or blade pass frequency, either of which can cause the system to resonate at that frequency, increasing the sound levels, and the possibility of damaging the installation. These risks can be eliminated by changing the speed of the fan, installing appropriate isolation, and/or detuning of the fan or affected system components.

NOISE MEASUREMENT

Overall noise levels can be measured at any installation using a variety of portable sound level meters, or more sophisticated equipment like a frequency analyzer (Figure 4).

Sound Pressure (Lp), is an atmospheric pressure change that is audible to the human ear, and is measured from a point in space where the microphone or listening device is located. The human ear can perceive a broad range of sound pressures,

from the threshold of hearing (2×10^{-7} microbar) to the threshold of pain (1 microbar). The threshold of pain is five million times louder than the threshold of hearing. The decibel is used in acoustical work to indicate sound pressure levels because it condenses this tremendous range of values to a workable range of from 10 dB to 130 dB. A *decibel* (dB) is a logarithmic ratio of some measured value to some reference value. It is standard international practice to use the sound pressure at the threshold of hearing as the reference value for the sound pressure level scale.

Figure 5 shows the relationship between the sound pressure measured in microbar, and the sound pressure levels measured in decibels.

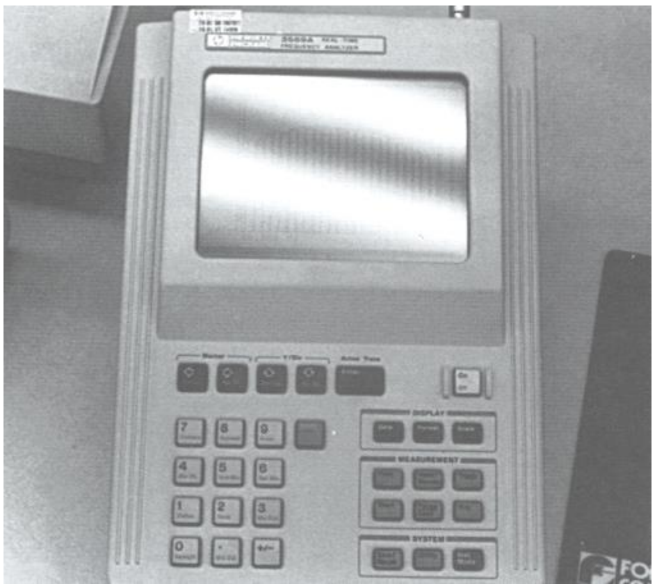


Figure 4 – Frequency Analyzer

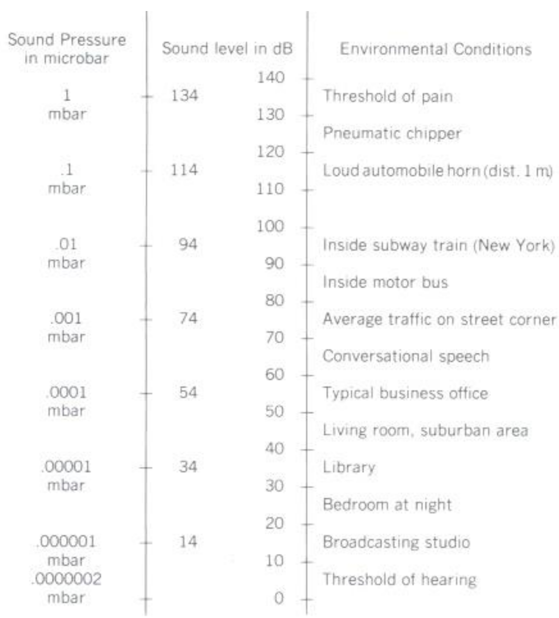


Figure 5 – Sound Pressure Measurements

Often a single sound pressure value is used to represent the total sound spectrum. This is expressed as dBA, indicating that the sound pressure, in decibels, has been adjusted to reflect a single number value for a sound pressure, weighted by the “A” scale. The “A” scale weighting reduces the effect of lower frequencies, with the intent to establish a value more proportional to the human ear frequency response. dBA is used by OSHA to set maximum allowable noise levels, prescribing a maximum dBA limit for an 8 hour exposure. dBA can be measured with a sound level meter, or calculated by applying the weighted values to the eight octave bands encompassing the range of hearing.

Better definition of sound pressure levels is gained by breaking the sound spectrum into discreet ranges. The standard practice is to divide the audio spectrum into eight octave bands identified by the center frequency of each band. Figure 6 shows the octave bands of the audio spectrum as defined by the American National Standards Institute (ANSI) Standard S 1.6, series 2.

Series 2 ANSI S1.6			
From (Hz)	**To (Hz)**	**Center Frequency (Hz)**	**Band Number**
45	90	63	1
90	180	125	2
180	355	250	3
355	710	500	4
710	1400	1000	5
1400	2800	2000	6
2800	5600	4000	7
5600	11200	8000	8

Figure 6

Fan manufacturers generally test and rate fan noise according to Air Movement and Control Association (AMCA) Publication 300 - Test Code for Sound Rating Air Moving Devices, and Publication 301 - Methods for Calculating Fan Sound Ratings from Laboratory Test Data. This testing procedure requires a reverberant or semi-reverberant room with a calibrated reference sound source to determine the room characteristics, and is known as the substitution method.

Sound data is acquired in the octave bands shown in Figure 6.

The measured sound pressure of a test fan is mathematically converted to a sound power level using predetermined microphone locations.

MEASURING FAN NOISE

dBA is a useful measurement for evaluating the overall noise level at a particular location, but this measurement takes into account all of the sound sources affecting that particular location, which include the sounds from all equipment in the area, natural sounds of the environment, and from other environmental factors. Some of these factors are the current physical properties of the air such as temperature, humidity, and pressure, whether the location is outside or inside, the size and material of the room. All of these affect the sound pressure experienced by the listener, and recorded by the sound level meter. Because of this, **it is impossible for the fan manufacturer to guarantee sound pressure levels or dBA values.**

For several years fan manufacturer,s and other makers of industrial equipment, have used Sound Power (Lw) values to test and rate fans. Sound power has been chosen because it is independent of the acoustical environment in which the fan is installed. It is the only value that is specific to the particular fan.

Sound power is the total energy emitted from a fan which is a function of the fan's speed and point of operation, and is independent of the fan's installation and surrounding environment. A sound power level is the acoustical power expressed in dB radiating from a sound source. It is defined as:

$$\text{Sound Power (Lw)} = 10 \log \frac{(\text{Watts})}{(10^{-12})}$$

Sound power levels can be converted into predictable sound pressure levels once the acoustical environment surrounding the fan is defined.

Sound pressure for a given fan changes with a change in air volume, pressure, or efficiency. Because of this, fans must be tested at several speeds and efficiency points. After a fan's sound power level has been determined at different speeds and points of operation, it is important to remember that these levels will always be the same unless the fan is physically altered. If a fan line is geometrically proportional, the sound power for other fan sizes can be accurately projected from the base fan. AMCA Publication 301 defines methods for acquiring such data.

FAN SOUND RATINGS

The sample table shown in Figure 7 shows a listing of total sound power for a particular fan size and type at several speeds in each octave band. Sound power ratings can also be presented graphically.

Fan RPM	Octave Bands							
	1	2	3	4	5	6	7	8
1100	73	72	68	62	59	58	51	45
1300	79	74	75	67	63	62	57	50
1500	85	77	78	71	67	66	62	54
1700	90	80	81	75	70	69	66	58
1900	93	83	83	78	73	71	69	61
2200	96	86	86	82	76	74	72	66
2600	99	90	86	88	80	77	75	70
3000	101	93	88	90	83	79	78	74
3400	105	98	90	94	87	82	81	78
3800	107	102	94	96	90	85	83	81
4200	109	106	97	98	93	88	86	84
4600	111	109	99	99	97	91	88	86
5000	113	112	102	100	101	93	90	88

Figure 7

Total sound power can be broken up to inlet sound power and outlet sound power. For all functional purposes, the sound power that is radiated from the inlet and outlet of a fan is equal to each other. Because a fan manufacturer can present its sound information in the form of inlet, outlet, and total sound power, it is important to clarify the identity of the rating before any comparisons and calculations are made.

In general, a fan manufacturers' sound ratings are at peak point of efficiency as shown in Figure 8. As stated earlier, fan efficiency and air turbulence contribute to changes in noise levels. Consequently, if a fan is operating at a point of operation outside its maximum efficiency range, the user will have to correct the manufacturers' sound ratings as shown in the table on page 5.

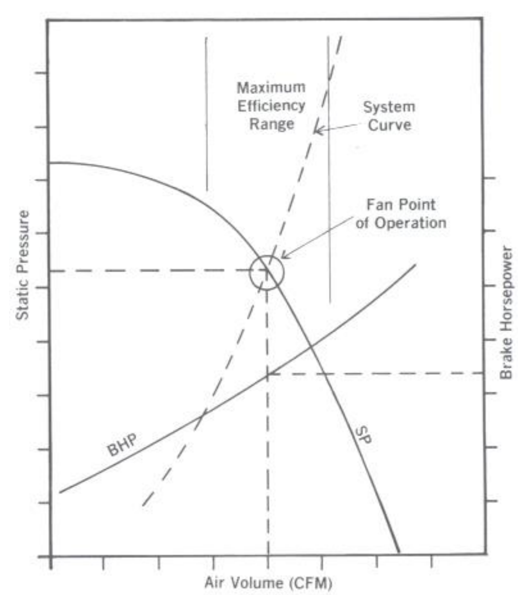


Figure 8

Figure 8 shows a fan's point of operation at the intersection of the static pressure and volume range on the curve. Since air volume can be defined by a velocity or velocity pressure through the fan's outlet area, the fan's point of operation can be defined by stating the ratio of velocity pressure to static pressure, or VP/SP. By using a chart such as the one shown in Figure 9, the user can make the necessary sound corrections for fan operations outside the maximum efficiency range.

Published fan sound power ratings and corrections only reflect noise created by air turbulence within the fan. Because of the infinite variables, mechanical noise and vibration noise are impossible to accurately predict, and are not included in the rating.

Another rating method is described in AMCA Publication 302 - Application of Sone Ratings for Non-Ducted Air Moving Devices. A *Sone* is a ratio of loudness between two sounds. The Sone scale is linear, ranging from soft to loud. Unlike the decibel, two Sones are twice as loud as one Sone. This method will produce reasonably accurate estimates of sound pressure in a free-field condition, and is used by manufacturers of roof ventilators and other non-ducted commercial ventilation products, but is not suitable for analytical purposes.

Fan Speed	VP/SP	Point of Fan Operation	Octave Bands							
			1	2	3	4	5	6	7	8
up to 2500	0 to .03	Peak SP	5	3	0	1	1	0	-3	-1
	.03 to .10	Peak ME	0	0	0	0	0	0	0	0
	.10 to .30	1/2 Peak SP	4	2	0	0	2	1	2	2
	.30 and up	Near Wide Open	2	2	2	1	3	3	2	3
over 2500	0 to .03	Peak SP	3	4	5	4	0	0	0	-2
	.03 to .10	Peak ME	0	0	0	0	0	0	0	0
	.10 to .30	1/2 Peak SP	4	2	0	3	3	3	2	1
	.30 and up	Near Wide Open	3	3	1	3	4	5	3	4

Figure 9 - Typical dB Corrections for Point of Operation

APPLYING SOUND POWER

When the sound power for a fan has been calculated at a fixed speed and known point of operation, the sound pressure can be estimated. It should be remembered that sound pressure or dBA predictions are only estimates based on certain known conditions or assumptions regarding the location of the fan and the physical installation.

The Short Form for Sound Calculations shown on page 8 is one way to calculate sound pressure. This is a step-by-step method for estimating sound pressure levels or dBA for a specific installation.

The short form only applies to outdoor installations or to indoor installations where the listener is relatively close to the fan and the room is relatively large. Such installations may be termed “free field.” Even given these assumptions, reflecting surfaces, inadequate support structures, high-loss ductwork, or flexible duct connections could seriously alter the outcome.

For example, the fan corresponding to Figures 7 and 9 might be required to operate at 1500 RPM:

Octave Band	1	2	3	4	5	6	7	8
Center Frequency	63	125	250	500	1000	2000	4000	8000
1. Fan Total Sound Power @1500 RPM	85	77	78	71	67	66	62	54
2. VP/SP Correction	5	3	0	1	1	0	-3	-1
3. Fan Sound Power (1) + (2)	90	80	78	72	68	66	59	53
4. Correction for Installation (inlet or outlet)	-3	-3	-3	-3	-3	-3	-3	-3
5. Corrected Sound Power at Fan (3) + (4)	87	77	75	69	65	63	56	50
6. End Reflection Values	14.5	9.0	4.5	1.5	0	0	0	0
7. Corrected Sound Power (5) - (6)	72.5	68	70.5	67.5	65	63	56	50
8. Conversion for Sound Pressure, Q=2	20	20	20	20	20	20	20	20
9. Sound Pressure at 15 feet	52.5	48	50.5	47.5	45	43	36	30

Line 1 - enter the published sound power for each octave band corresponding to the required speed.

Line 2 - enter the appropriate VP/SP correction. For this example, assume VP/SP = .025.

Line 3 - enter the algebraic sum of lines 1 and 2.

Line 4 - enter the appropriate correction for the type of fan installation. If neither the inlet nor outlet are ducted,

no correction is necessary. If either the inlet or outlet is ducted away from the listening location deduct 3 dB. This 3 dB reduction accounts for the assumption that the amplitude of inlet and outlet noise is approximately equal and half the noise is ducted away. Figure 10 provides a graphic depiction of the effects of adding or subtracting noises of similar or like amplitude.

Figure 10

If the inlet and outlet are both ducted away from the listening location, only the sound power radiated through the fan housing will remain. The appropriate reduction will vary from one fan to another depending upon the specific housing thickness and reinforcements and their attendant transmission loss. Refer to the manufacturers’ rating tables for the appropriate reduction for a specific fan type.

For this example, assume only the outlet noise is ducted to the listening location.

Line 5 - enter the algebraic sum of lines 3 and 4.

Line 6 - End reflection is a phenomenon that takes place when a sound wave reaches an abrupt expansion such as the end of an open duct. At this point some of the sound waves are actually reflected back into the duct so that the resultant sound power level is reduced. The effects are more pronounced in lower frequency ranges and in smaller duct diameters as shown in Chart III, page 8. For applications where noise level emitted from the inlet or outlet duct concerns the listening location, the duct diameter must be determined and the appropriate values subtracted from the fan sound power.

For this example, assume only outlet ducted noise is available at the listening location and the duct is 15" in diameter. (See Chart III on page 8.)

Line 7 - enter the difference between lines 5 and 6.

Line 8 - enter the correction for directivity and distance.

As mentioned previously, the amplitude of a noise level will vary depending upon the installations and the distance between the source and the listening location. The number of reflecting surfaces also determines the sound wave radiation pattern. These patterns are known as directivity factors (Q) and indicate the type of radiation from the number of reflecting surfaces.

AMCA Publication 303 - Application of Sound Power Level Ratings describes Q = 1 as having spherical radiation with no reflecting surfaces. An example would be an axial fan located in a stack. Q = 2 is used for hemispherical radiation where one reflecting surface is present such as a fan on the floor in the middle of a room. For each additional reflecting surface, the directivity factor is doubled. For example, a fan mounted on the floor directly adjacent to a wall would have a Q = 4 factor.

The appropriate directivity factor must be used in conjunction with the distance from the noise source to the listening location to obtain the reduction factor (Lw - Lp) to convert sound power to an estimated sound pressure. Using Chart I on page 8, the listening distance from the source must be plotted on the bottom horizontal graph and a vertical line should be drawn at that point. A horizontal line drawn from this vertical line at its intersection with the appropriate directivity line will indicate the (Lw - Lp) reduction.

These estimates apply to a listener's position from the noise source and do not consider outside influences from other machinery or unpredictable obstructions, but produce reasonably accurate estimates of sound pressure in a free field condition or outside installations.

For this example, assume a Q = 2 directivity factor at a distance of 15 feet. (See Chart I on page 8.)

Line 9 - deduct line 8 from line 7 and enter the result.

The sound power levels represent the final estimate based on all the stated conditions. The one remaining step is to determine the proper dBA value.

The dBA value is the sound pressure level corrected to the "A" weighting network. This is accomplished by deducting the proper "A" weighting value from each of the eight octave bands, then using the graph from Figure 10 to combine the results to obtain a single number dBA value that represents the fan and its particular installation. *Because decibels are logarithmic values, simple addition cannot be used.*

A simpler method of approximating dBA values can be found on Chart II on page 8. Using the scale on the left hand side of the graph, plot the sound pressure levels from line 9 directly on to the graph for each octave band. Then the maximum dBA can be derived by finding the band number (center frequency) that exceeds the highest octave band level by the most decibels. In our example, band number 5 (1000Hz) exceeded the octave band level 40 dBA by 8 dB. This was greater than any other band number. Therefore, the dBA level for this fan would be approximately 48 dBA at 15 feet based on a Q-2 directivity.

Another method is to combine decibels such that a logarithmic addition can be employed in lieu of the tabular method shown in Chart II. Logarithmic addition involves calculating the antilog of each decibel to be added, summing the antilogs, finding the logarithmic sum, and multiplying by 10. This method and the formula are given in AMCA Publication 303.

TROUBLESHOOTING

To avoid undesirable noise levels in the final installation, the system designer needs to consider many factors. First, an acceptable noise level criteria must be established, based on the activity in the area, the nature of the noise, the relationship of the listening location, noise-criterion curves, and the OSHA permissible noise exposure regulations.

Properly selecting a fan type and operating it at peak mechanical efficiency will assure the quietest possible operation. It is not always possible to select a fan that does not exhibit a predominant blade pass frequency, but an awareness of this will help in selecting acoustical attenuation when necessary.

Octave Band	Sound Pressure From Line 9	Correction For "A" Weighted Network	dBA Value by Octave Band	Diff.	Factor From Ill. #10	Factor + Higher Value	Diff.	Factor From Ill. #10	Factor + Higher Value	Diff.	Factor From Ill. #10	Single Number dBA Value
1	52.5	-26.2	26.3	-5.6	1.1	33.0						
2	48	-16.1	31.9				-13.3	.2	46.5			
3	50.5	-8.6	41.9	-2.4	2.0	46.3						
4	47.5	-3.2	44.3							-1.6	2.3	50.4
5	45	0	45	-.8	2.6	47.6						
6	43	+1.2	44.2				-9.6	.4	48.1			
7	36	+1.0	37	-5.9	1.0	38.0						
8	30	+1.1	31.1									

Location of the fan with respect to the listener is very important. The greater the distance, the lower the noise level. The use of absorptive and reflective materials as well as isolation usually control excessive noise.

If the final installation seems excessively noisy, an octave band sound analyzer should be employed to measure the noise level. Because it analyzes the spectrum by octaves, it is helpful in isolating components within the spectrum that are major contributors to the noise problem.

Often, the fan is not the major source of the noise; many times it is nearby machinery or the surrounding environment that is louder than the fan. After identifying the noise source, its reduction can be approached from two directions:

1. Reduce the noise at the source.
2. Reduce the noise at the listening location.

The first approach is usually the most cost effective. To reduce fan acoustical noise, a reduction in sound energy is important. Lining ductwork with sound absorbing material or adding duct silencers will reduce airborne noise within the duct system. Flexible connectors between the fan inlet, outlet, and connecting ductwork will aid in reducing both vibration noise and mechanical noise that may be transmitted through the entire system.

Fan noise produced by vibratory forces can be induced by a number of components. Sometimes the source is easily detected from experience and at other times measuring instruments are required. The solution to vibratory noise will depend on where it occurs. Reducing the amount of the vibration, eliminating it by substitution, isolating it, or changing the frequency are all possible solutions.

For example, unbalance is a chief cause of vibratory noise. Consequently, balancing the rotor will reduce the vibration caused by imbalance. Replacing a noisy bearing or drive component will eliminate the source. Installing rubber or spring isolators will prevent transmission of the noise to the mounting structure. Detuning natural frequencies of a structure by changing the fan speed or the natural frequency may eliminate this problem.

Using the second approach, the noise level at the listening location can be reduced by increasing the distance of the sound path. This can be accomplished by moving either the fan or the listener or by rotating the fan so that the noise is directed away from the listener. Changing the characteristics of the room by adding sound absorbing material will help reduce noise However, the effectiveness of sound absorbing material drops off rapidly at frequencies below 250 Hz.; consequently, this approach is somewhat limited. Enclosing the fan in a sound absorbing room, for example, will aid in reducing noise transmitted from the fan structure but will do nothing about noise within the duct system. Erecting sound barriers or employing some type of ear protection are also alternative solutions.

These troubleshooting tips only cover a few possible alternatives. Volumes of reference material are available on the subject, and acoustic consultants are available to assist in the areas of noise abatement and acoustical control. Fan manufacturers can provide assistance in resolving noise issues related to the specific fan but normally do not perform overall acoustical engineering consulting.

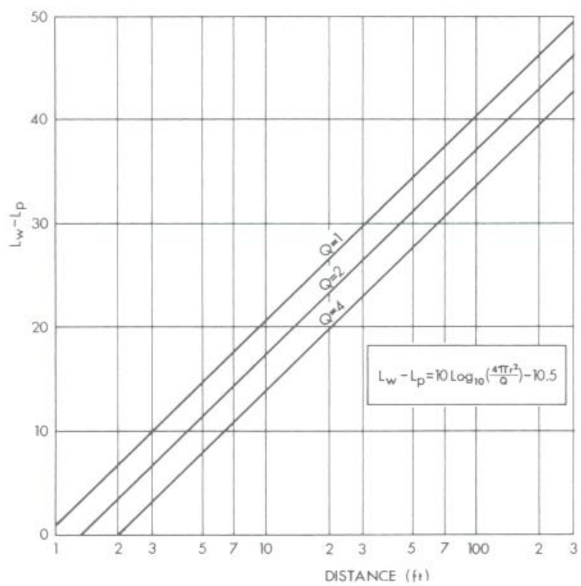
SHORT FORM FOR SOUND CALCULATIONS

This form is to be used for the approximate sound pressure level calculation of a fan, assuming that the listener's position is in the dominant free field. In most cases this can be considered no more than 5 feet in an enclosed room, or an outside installation free from reflecting surfaces.

OCTAVE BANDS	1	2	3	4	5	6	7	8
CENTER FREQUENCIES	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
1. Fan Sound Power Rating at __________RPM								
2. VP/SP Correction								
3. Fan Sound Power (1) + (2)								
4. Correction for Installation (Inlet, Outlet)								
5. Corrected Sound Power at Fan (3) + (4)								
6. End Reflection Value (Chart III)								
7. Corrected Sound Power (5) - (6)								
8. Conversion to Sound Pressure (Chart I)								
9. Sound Pressure at___________ ft. (7) - (8)								

The estimated dBA value is _______ at _______ ft. (Chart II)

CHART I
DIRECTIVITY/DISTANCE REDUCTION



[Given directivity and distance, Sound Power is converted to Sound Pressure.]

Q-1 UNIFORM SPHERICAL RADIATION with no reflecting surface. Example: Stack discharge.

Q-2 UNIFORM HEMISPHERICAL RADIATION with one reflecting surface. Example: Floor mounted fan.

Q-4 UNIFORM RADIATION over 1/4 SPHERE with two reflecting surfaces. Example: Fan mounted on floor near interior wall.

CHART II
SOUND PRESSURE TO DBA CONVERSION

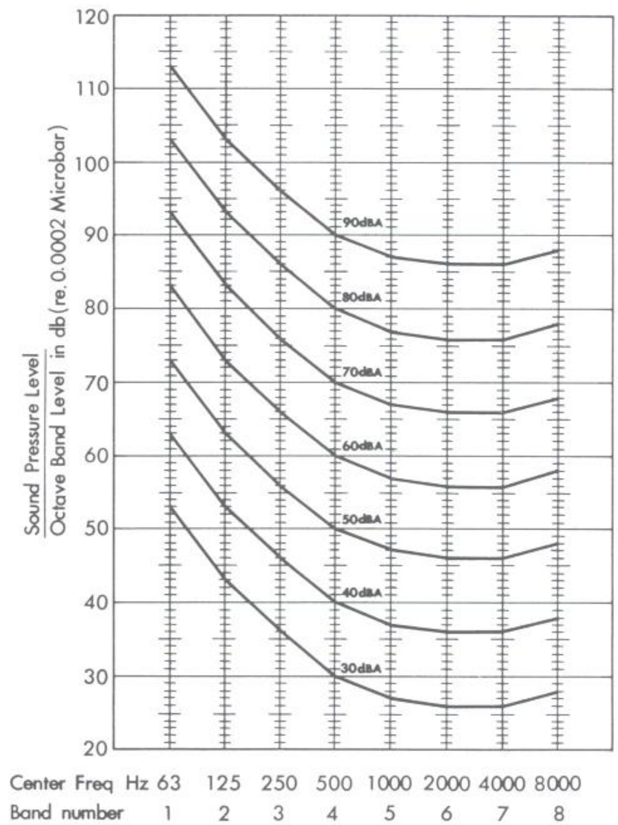


CHART III
END REFLECTION VALUES (Decibels)

Octave Band		1	2	3	4	5	6	7	8
Hz		**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Duct Diameter Inches	5	23.5	17.5	12.0	7.0	2.5	.5	--	--
	10	17.5	12.0	7.0	3.0	1.0	--	--	--
	15	14.5	9.0	4.5	1.5	--	--	--	--
	20	12.0	7.0	3.0	1.0	--	--	--	--
	30	9.0	4.5	1.5	.5	--	--	--	--
	40	6.5	2.5	1.0	--	--	--	--	--