Application Notes
(Selecting the Right Fan)

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Section 1
Determining the Airflow and Pressure Requirements

There are two electrical analogs of a cooling system that electrical engineers and designers should readily understand. The first analog (figure 1), assumes that the flow-resistance (R_S) of the entire cooling system is constant (that the system is linear). However, this assumption only yields approximate results because air is an expandable medium, and flow resistance is actually a complex non-linear function of flow rate for all but the simplest and most geometrically regular structures.

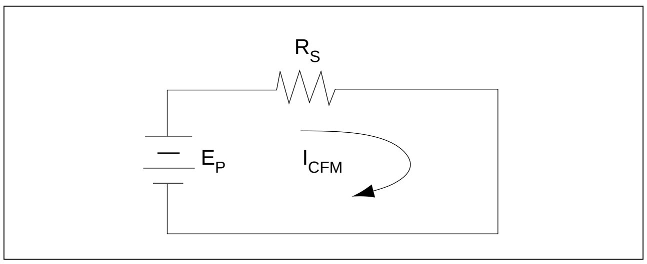

Figure 1. Electrical analog of pressure/flow relationship in a linear system.

In this first analogy:

Ep ≡ pressure head developed by fan, in equivalent inches of water column height.

I_{CFM} ≡ ratio of air flow through the system, in cubic feet per minute (CFM) of air at in-coming temperature.

R_S ≡ resistance to air flow (assumed constant) of the system, in inches/CFM.

Ep ≡ $I_{CFM} \cdot R_S$ (as in Ohm's Law). (eq. 1)

The flow resistance of a system is actually the sum of several resistances in series—the chamber, the filter, the fan itself, the exit ports and the entry ports (figure 2), and each of these varies with flow rate in a different manner. Finally, this analog assumes that the character of the flow-lines does not change significantly over the range of air-flows to be considered. The assumption is that the laminar-flow regions remain laminar, and the turbulent-flow regions remain turbulent, and that the extent of each region remains approximately constant.

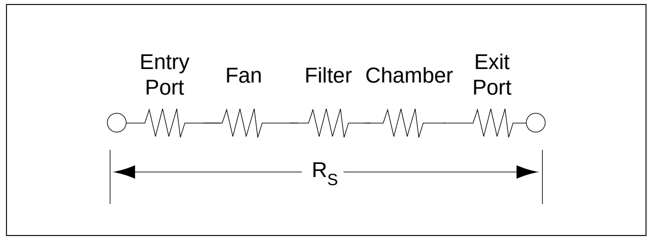

Figure 2. Series Flow Resistance

This "Ohm's Law of Air Flow" concept is a poor model of the actual situation in a practical system. Nevertheless, the concept proves practical for estimating the effect of small variations in pressure or air flow, around a point that has been verified independently. That is all we need from the concept.

The second analog (figure 3)is a more accurate model of the behavior it represents. This analog relates the allowable temperature rise (of the cooling air) to the airflow.

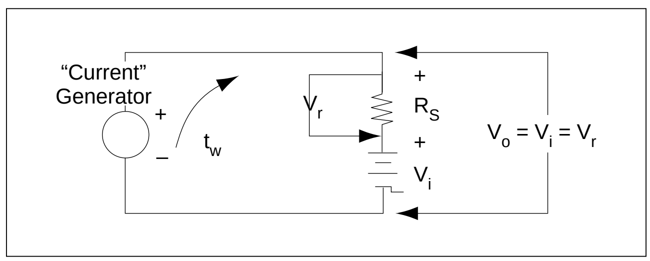

Figure 3. Electrical analog of temperature/flow relationship in a linear system.

I_W ≡ power transferred by the dissipating equipment into the airstream (Watts).

V_i ≡ temperature of incoming air (°C).

V_O ≡ temperature of outgoing air (°C)

V_r ≡ temperature rise of airstream (°C).

R_a ≡ thermal resistance of airstream at a particular flow rate (CFM) and for a particular heat-transfer geometry; assumed constant (°C/Watt).

V r = $I_W \times R_a$ (as in Ohm's-Law). (eq. 2)

This second analog is based on certain assumptions also, but these assumptions are neither as restrictive nor as difficult to justify as the first. They are:

- that the thermal capacity of the air stream remains constant, regardless of temperature (approximately true over moderate temperature ranges).
- that the rate of heat transfer from the dissipating elements of the system to the airstream remains constant regardless of temperature, for a given incoming airflow (nearly true for all conventional temperature ranges and airflow patterns).
- that we are justified in averaging the temperature of the various parts of the airstream, assuming “good mixing” — and this is not a dangerous assumption, if the system is well designed, without "pockets" or "starved areas."

One more equation is needed to equip us to apply the second analog: the one that relates the thermal resistance of the air stream to the magnitude of the airflow in CFM.

$$R_a = \frac{1.76}{CFM} (°C/W) \qquad \text{(eq. 3)}$$

Substituting equation 3 in equation 2, we get:

$$V_r = \frac{(1.76) I_W}{CFM} (°C/W) \qquad \text{(eq. 4)}$$

or to put equation 4 in words:

$$= \frac{1.76 \text{ (Watts transferred)}}{airflow_{CFM}} \qquad \text{(eq. 5)}$$

This relationship is illustrated in figure 4. Here is how to apply equations 4 and 1 to a practical example:

> An enclosure contains electronic equipment capable of dissipating 190 Watts, worst-case It is desired to maintain the temperature of the Outgoing air at no higher than 55˚ C. The highest input (ambient) air temperature will be 40˚C. The resistance to air flow of the cabinet is estimated at 0.015 inches/CFM.

Step 1: Calculate the required CFM, from equation 4. From the data, we get:

$$V_r - V_O - V_i = 55 – 40 = 15°C$$

$$I_w = 190W$$

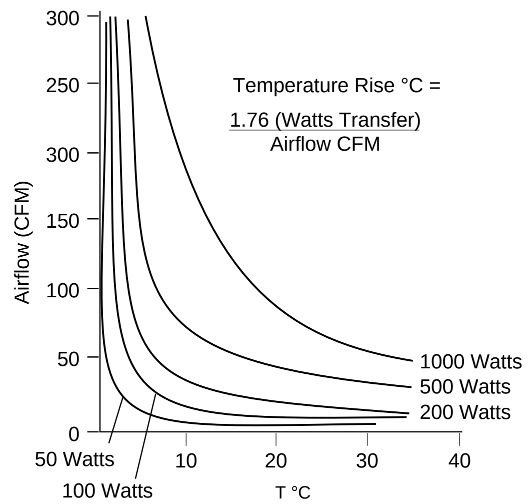

Figure 4. Air flow versus Temperature Rise for Various Power-Transfer Levels.

Substituting these into equation 4, we get:

$$15 = \frac{1.76 \times (190)}{CFM}$$

Step 2: Calculate the pressure drop at which the fan must deliver that minimum CFM.

From the data and Step 1, we get:

$$R_s = (0.015 \text{ inches})/(CFM)$$

$$I_{CFM} = 22.3CFM$$

Substituting these into equation 1, we get:

$$E_p = 22.3 \text{ x } 0.015 = 0.335 \text{ inches}$$

Thus, we need a fan capable of delivering 22.3 CFM against a pressure “head” of 0.335 inches of water— assuming, among other things, that the estimated resistance to air flow (0.015 inches/CFM) was reasonable.

Consult the fan manufacturers' pressure/flow rate curves for fans that will fit the enclosure and fit your ideas of satisfactory design (in terms of noise level, or voltage). Figure 5, 6, and 7 illustrate some typical curves.

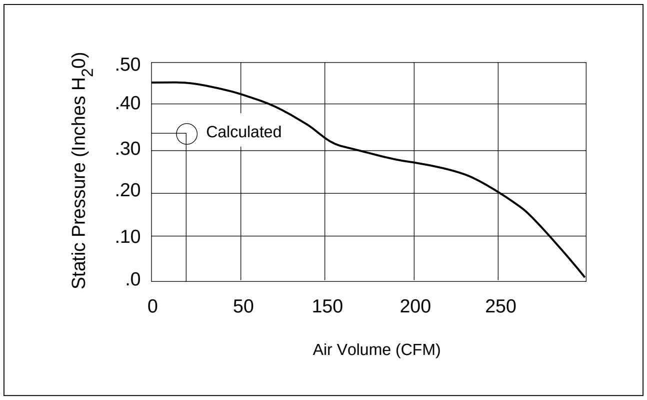

Figure 5. Fan A Performance Curve

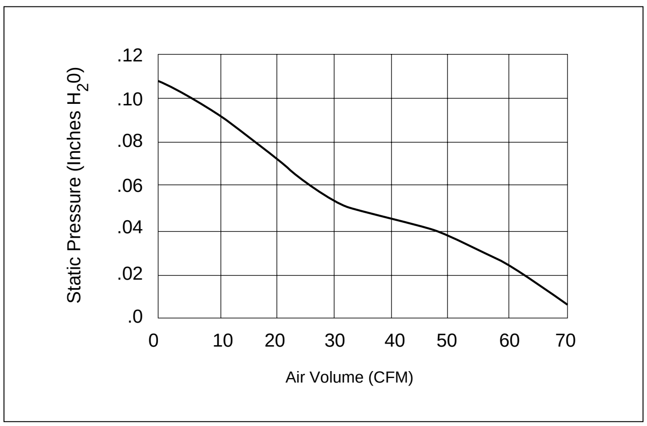

Figure 6. Fan B Performance Curve

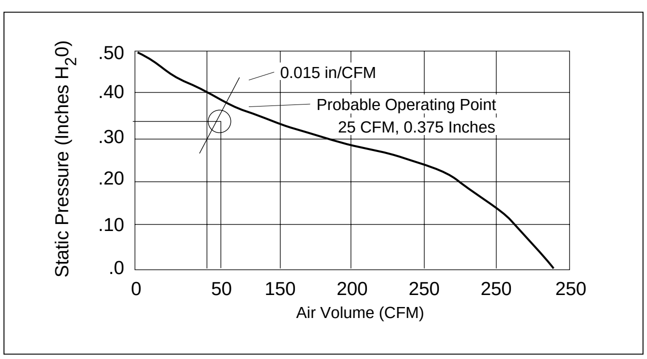

Figure 7. Fan C Performance Curve

Fan A will certainly do the job—but it is a large, powerful, relatively expensive device, much more fan than you need, for this application.

Fan B will not do the job. It's just not capable of overcoming the pressure drop, at any volume. (To put it differently, Fan B would deliver about 6.7 CFM, at 0.10 inches pressure in this application.)

Fan C is just right. It has a reasonable margin of safety at 22.3 CFM, for this enclosure. The probable operating point (obtained by drawing a line through the calculated operating point of 22.3/0.335, with a slope of 0.015 inches/CFM) appears to be about 25 CFM and 0.375 inches. Fan C is small, quiet, relatively modest in price. It will not overpower the system, so as to encourage excessive dust infiltration or airborne contamination, and it will not create an unnecessary noise problem.

Getting Practical

First of all, determining (or at least estimating) the resistance to airflow is the greatest practical difficulty you face. It's easy enough to say "test the system over a range of air flows," but usually you cannot do that conveniently—especially when the fan is being specified. Furthermore, there is no one value of resistance to air flow, as we have already noted. The curve of pressure drop versus flow rate for a typical complex enclosure is shown in figure 8.

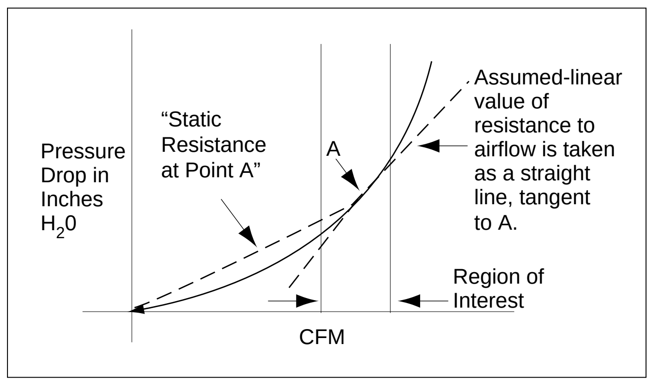

Figure 8. Experimentally determined pressure/flow curve

You can approximate this characteristic by a straight line over a short distance, but the slope of that line bears no meaningful relationship to the "static" pressure line drawn through the origin and the center of the "region of interest." What's to be done?

Here's one approach—essentially a rule of thumb, but a good way to estimate the pressure drop in many common enclosures. The graph of figure 9 illustrates approximate values of pressure drops vs. CFM for a variety of common structures. Pick the one closest to your application. Estimate the pressure-drop for the CFM of flow you need. Then use the following "rules" to scale up or down from that number.

For a given volumetric flow (CFM), the pressure drop through a given structure varies inversely as the square of the cross-sectional area perpendicular to the direction of flow. Thus, if we show a cross-section of 2 square feet, and you have the same configuration with a cross section of 4 square feet, the pressure drop for any given flow would be only 25 percent of the pressure drop shown on our graph.

For a given flow, the pressure drop varies directly as the length; except if the path becomes short, the fixed entry and exit drops become significant. For a given structure, the pressure drop varies directly as the square of the flow, unless the flow falls outside the ranges given by our curves.

In approximating the effect of differences between our "idealized" geometry and the actual configuration, remember that small differences near the entry or exit ports create larger errors than differences in mid-path, where the velocity is usually lower.

Another practical problem is presented by the extreme requirements—either unusually high pressure at low CFM, or unusually high CFM at low pressure. The high pressure case may often be solved by putting fans in series. Put one at the entrance port, pulling in, and one at the exit port, pushing out, thus doubling the pressure capability. The high volume case may often be solved by putting fans in parallel, with two or more located either at the entrance or the exit.

Perhaps the most challenging practical problem is not developing enough flow, but providing enough heat exchange. If you read the early part of this discussion again, you will see that we were careful to relate temperature rise to the power actually transferred to the air. Heat transfer by convection (radiation is generally trivial) depends upon the nature of the contact between the surface to be cooled and the cooling medium, upon the area of that surface, and the velocity of the airflow. You may need to use fins, deflecting vanes, and even constricting pipes, to direct blasts of high-velocity air over the heat-exchange surfaces. Remember, all calculations in this and any other text on this subject assume that you have successfully transferred the heat. Remember also that every vane or fin that you introduce into the air stream means more pressure drop.

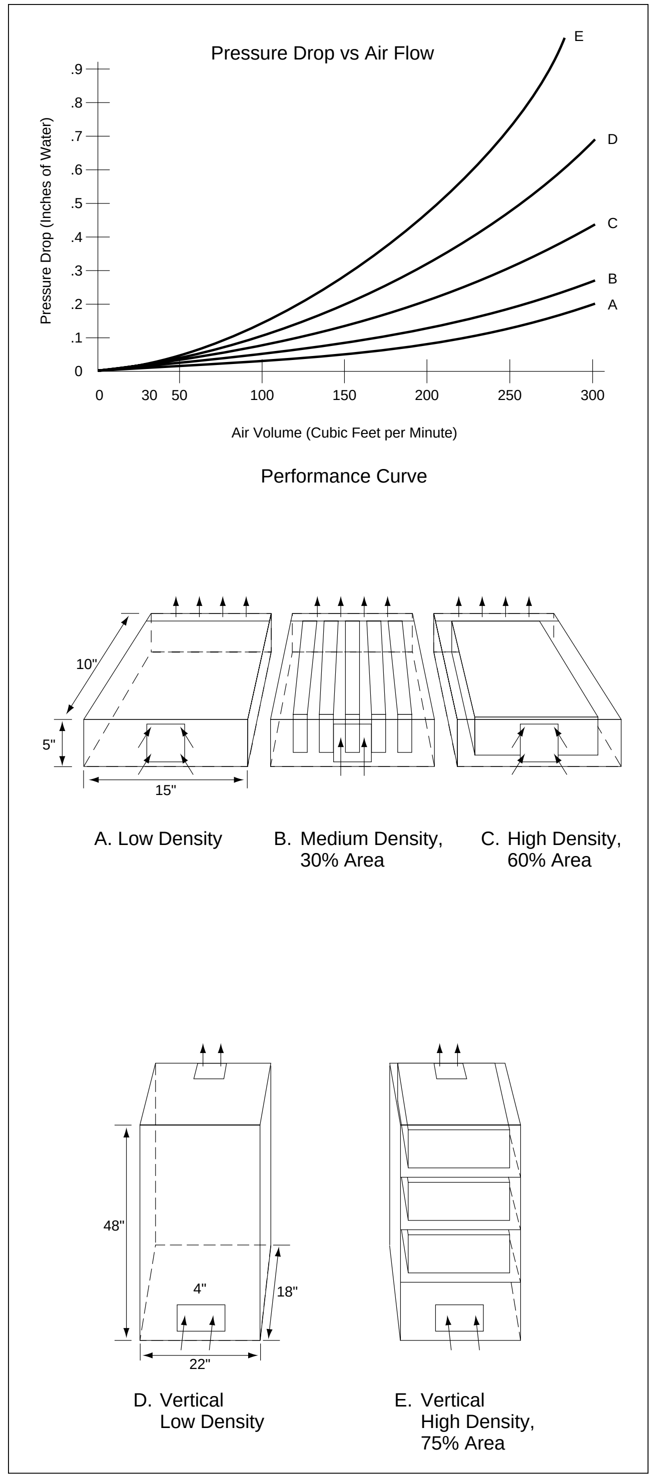

Figure 9. Examples of electronic enclosures

Section 2
Satisfying the Mechanical and Structural Requirements

Except in those rare applications in which one must allow for great variations in the equivalent airflow resistance, or in the viscosity and density of the air, it makes absolutely no difference what type of air moving device is used — just so long as it can produce the required CFM against the pressure drop (in inches of water) that it must overcome.

Below about one horsepower, there is no consistent and useful relationship between the type of air-moving device and its cost, efficiency, or capacity. Since almost all of the fans (and most blowers) used for cooling electronic equipment, data-processing devices, business machines, and industrial process controls or instrumentation are in the less than 200 Watt class (less than 1/3 HP), the only things that matter are:

• flow-rate versus pressure drop

• size and weight

• noise levels

• magnetic interference

• toughness and dependability

• cost - to buy and to use

Too much airflow may be almost as bad as not enough because excessive air flow means that a greater volume of dust, dirt, and other airborne contaminants are circulated through the enclosure—filter or no filter. That’s because no ordinary filter is better than 80 percent to 90 percent efficient at reasonable pressure drops. Accordingly, at double the airflow you increase the amount of contaminants settling on your exposed conductive paths and components by almost 50 percent. This result could harm system performance and require more frequent cleaning. Finally, too much airflow means that the fan is too big, too powerful, and, therefore, probably more expensive than necessary.

The constraints to respect in selecting a fan for a given application are:

• Fit

• Intrusion

• Safety

First of all a fan must fit. You may not always be able to use the lowest-cost, best-performing fan for a given airflow/pressure requirement, simply because it can't be accommodated in the enclosure. In this connection, don't overlook the possibility of mounting the fan on the top or bottom of a short enclosure (figure 1). A 4" x 4" fan won't fit on a rear panel, but it will fit as top-exhaust or bottom mounted unit and generally costs less, while cooling more efficiently.

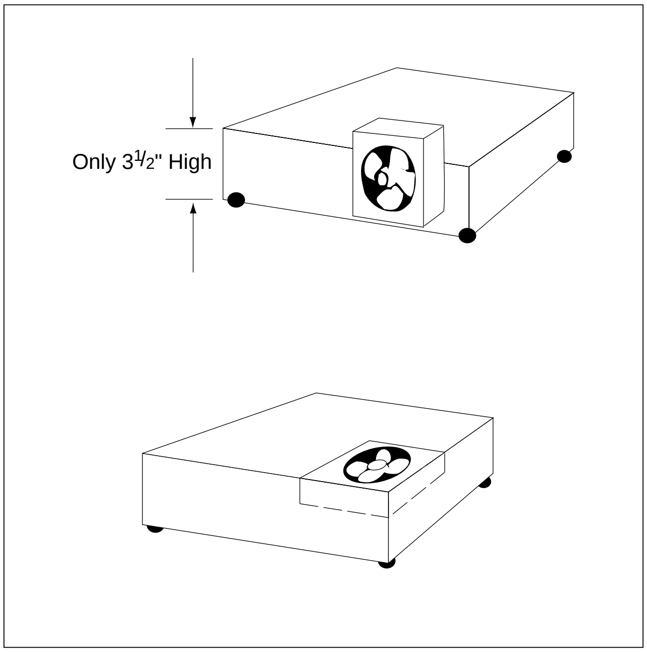

Figure 1. One way out of a common fit problem

Second, the fan must intrude as little as possible on the equipment within the enclosure, but it can't just be “tucked in” anywhere. The airflow path should be as direct as possible, and should route the cooling air past the important heat-exchange points in the equipment. Figures 2 and3 show what to look for in selecting a fan that will intrude the least and yet create a sensible airflow path.

Next, consider safety. Fingerguards are a minimum requirement in most cases, likewise the fan must have approved wiring. The wiring should be flexible if the fan is mounted on a hinged panel or door. But not all fans are easily equipped with guards, and some require special fixtures to achieve approved wiring. Safety considerations sometimes argue against the most convenient form of fan or mounting scheme, including locations illustrated in figures 2 and 3.

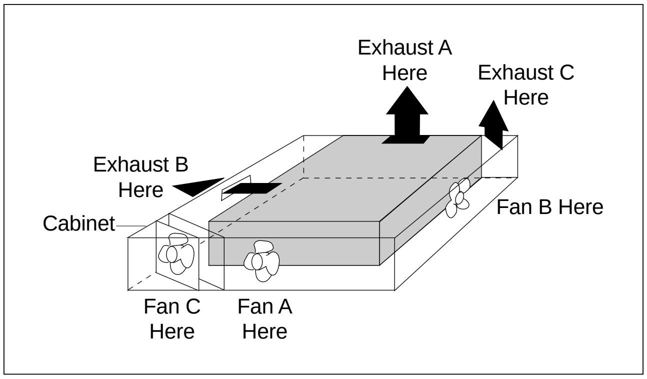

Figure 2. Short Cabinet Mounting Schemes

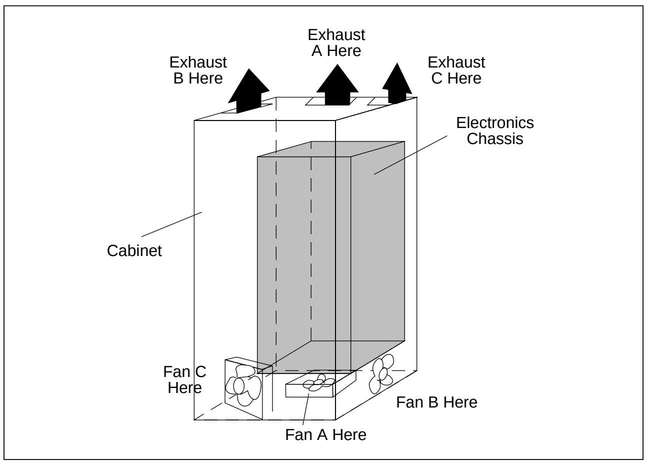

Figure 3. Tall Cabinet Mounting Schemes

For a given CFM, the average velocity of air through a structure is constant, regardless of the size, shape, or type of fan that moves the air. But the peak velocity of airflow is another matter. In some fans, the velocity of air leaving the tip of the blades is ten or more times as high as the average velocity. And the larger the fan, the higher the peak velocity. Furthermore, for a given design and size of fan, the higher the rotational speed, the higher the velocity.

Who cares about peak velocity? Your ears care, for one thing. Acoustical interference is an important consideration in fan selection—not just for human comfort and efficiency, either; noise means vibration, and vibration can cause all kinds of trouble in electrical and electronic equipment including some subtle kinds of failure.

Consequently, except in the regions in which the airflow is being guided past or through heat exchangers, velocity must be kept as near to the average velocity as possible.

Therefore, assuming you have a group of “possible” fans to choose from - choose the one that has:

- the smallest-diameter blades (down to 4")
- the slowest motor (lowest RPM)
- the most uniform velocity profile[1] and that will still deliver the CFM/pressure performance you need.
- Figure 4 shows a striking example of the advantages of choosing wisely, for low peak velocity.

Example: The cooling requirement is 200 CFM at 0.15" of water column. Two fans are under consideration. Their performance curves are illustrated (figure 4).

Fan A	Fan B
10" diameter	6" diameter
1600 RPM @ 60 Hz	3300 RPM @ 60 Hz
Axial design	Axial design

The prices of the two fans are within about 10 percent of each other, and both will fit (although the smaller lighter design fan B, is obviously somewhat easier and less expensive to accommodate structurally). Both fans will do the job - fan “B” having slightly more “margin” of safety - and neither could be considered “overpowered.” But here's the clincher - the noise level (Speech Interference Level) of fan B is almost 10dB lower than that of fan A. This is true despite the lower speed of fan A. Clearly, the 10:6 diameter ratio is more effective in this use in reducing noise than is the 2:1 speed ratio.

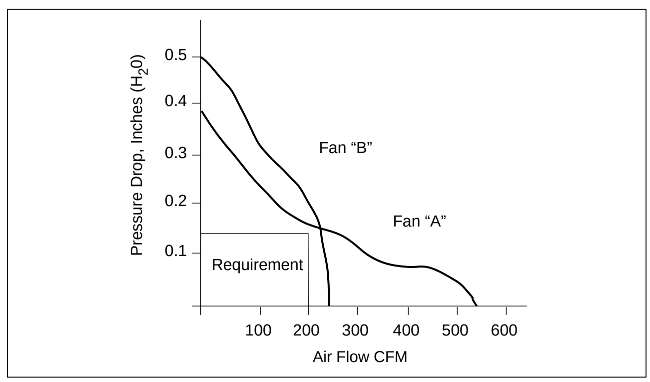

Figure 4. Performance curves of two “possible” fans for the example.

1. Almost always, the venturi-ported, impeller fan designs have the most uniform velocity profiles. Propellers are the worst of all.

Section 3
Selecting for Acceptable Noise Levels

The third step in optimum fan selection is verifying that the SIL (Speech Interference Level) and the corollary vibration and microphonic effects as well, of the available fans are acceptable in your application.

Basic Definitions

Sound is an oscillation in pressure, stress, or particle velocity, in an elastic or viscous medium. In addition, sound is the superposition of such propagated oscillations. Noise can be defined as any undesired sound.

The sound pressure level is defined by:

$$SPL = 20\log10 \frac{P}{0.0002} \qquad \text{(eq. 1)}$$

(Sound Pressure Level) in decibels (dB).

Where

P = the root-mean-square (rms) sound pressure, in microbars (µB), and

$$1\ \mu B = 1\ dyne/cm^2 = 1.45 \times 10^{-5}\ psi$$

Use the rms value because most sounds are complex signals, not pure sine waves, and the best way of describing them is in terms of their energy, which is measured by their rms values. The reference value of 0.0002 dB was selected somewhat arbitrarily, because this value represents the threshold of hearing (on average) for human beings of a 1000 Hz tone. When P = 0.0002 then SPL = 0 dB. (It is interesting to note that 0 dB is only about 3×10^{-9} psi—the Human ear is extremely sensitive.) The range of sound pressures discernible is quite large—from the hearing threshold level to about 10 million times greater in intensity. The use of a decibel (dB) scale condenses this range to a convenient scale from 0 to about 140.

But there is no such thing as “the” sound level. You must know over what frequency range the acoustic energy has been measured, and how the level at each frequency has been added into the total. The standard methods of doing this are given below, and related directly to fan specifications. Fan specifications describe, quantitatively and qualitatively, how much acoustic noise a fan generates. For example, a typical fan noise specification may read: “Noise Level is 61 dB (A) or 54:7 dB SIL at 117V/60 Hz (figure 1).”

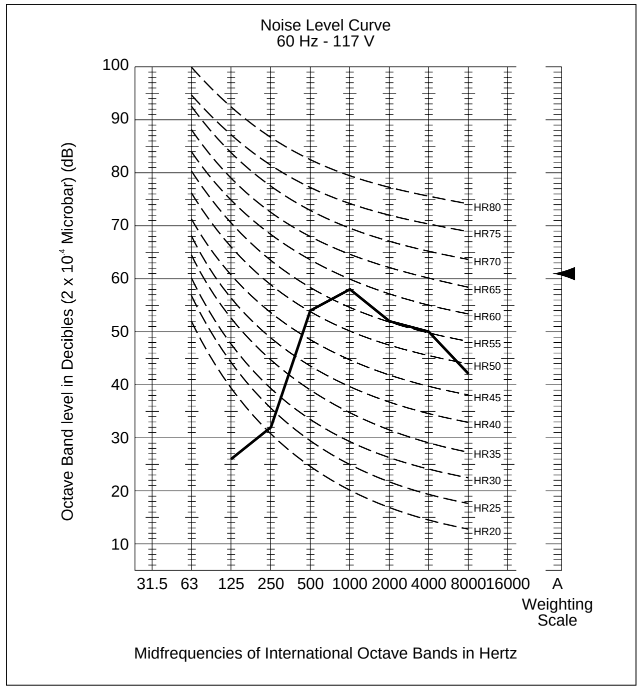

Figure 1. Typical noise-level curve for a fan.

The first two things we want to know about noise are loudness and frequency distribution. These facts are usually determined by use of a sound-level meter, an instrument consisting of a microphone, an amplifier, weighting networks (to be explained later), a calibrated attenuator, and an output-indicating meter. For accurate frequency analysis, the basic circuits and components of the sound-level meter should (ideally) have a uniform or flat response over the audible frequency range from about 20 Hertz to about 20 kilohertz. (The weighting networks are used to modify this uniform response in known, meaningful ways.)

One way to find the frequency distribution of the noise is to divide the audible range into ten octave bands as follows:

Effective Band Hertz	Center Frequency Hertz
22.1-44.2	31.5
44.2-88.4	63
88.4-177	125
177-354	250
354 - 707	500
707-1,414	1000
1,414-2,828	2000
2,828 - 5,657	4000
5,657-11,314	8000
11,314-22,628	16000

The nominal frequency range (ratio of high end to low end) of each band is 2 to 1, and the center frequencies increase at a 2 to 1 ratio. Now measuring a noise source, such as a fan, the meter can measure the sound pressure level for each octave band. Filters are used to remove or block energy components having frequencies above or below the band being measured. This information can be plotted as shown by the solid line on figure 1. The sound-level meter measured 26 dB for the octave band with 125 Hz center frequency, 32 dB for the next octave band, and so on — to 42 dB for the 8000 Hz center frequency band.

Because a curve is a cumbersome way of stating a specification, fan designers devised a single number to represent the noise intensity that the human ear hears. A simple average of the seven levels measured is one way to approximate this value, and this equals 44.8 dB for the curve of figure 1. However, sounds or noises at frequencies under 600 Hz do not sound as loud to human ears as equally intense (loud) noises at higher frequencies. The sound-level meter has provision to compensate for this human hearing “prejudice” - with three weighting circuits - A, B, and C - which discriminate against the lower frequencies. To insure uniformity among the different makes of sound-level meters of this kind, the U.S.A. Standards Institute (USASI, formerly The American Standards Association), has established a standard to which all sound-level meters should conform. This standard is shown as the frequency response characteristic of figure 2.

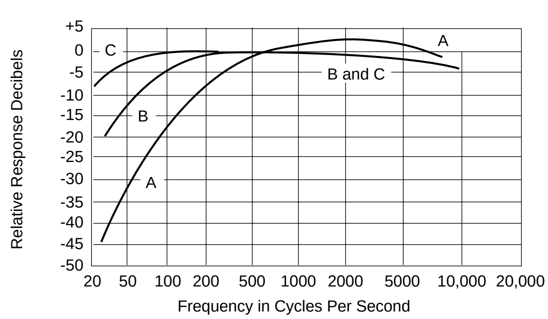

Figure 2. Standard frequency responses for weighting characteristics of sound-level meters.

Any sound or noise measurement should be identified as the weighting used: “53 dB (A),” or “the A-weighted sound level is 53 dB”. The reading is of sound level, not sound pressure level. Sound-pressure-level is read only when the sound level meter frequency response is flat (uniform) over the entire audible range, in other words not weighted at A, B, or C to fit the characteristics of the human ear.

A-weighted readings are used widely, and many noise-level graphs include an A-weighted scale. An A-weighted scale is included in figure 1 on the right-hand side, and an arrow indicates the reading of the sound-level meter when set at A-weighting—the specified noise of 61 dB (A) given by our specification.

A noise level may be loud enough to interfere with speech communication. Speech sounds are distributed over the frequency range from about 100 to 10,000 Hz, with most of the intelligence or information in the 200-6000 Hz band. If we measure the noise energy in that region only, we will have some measure of its ability to interfere with speech.

A three-band analysis of the octave bands centered on 500, 1000, and 2000 Hz will permit us to determine the arithmetic average of the sound-pressure levels in these three bands. The resulting numerical average, in decibels, is defined as the Speech Interference Level (SIL). From figure 1, we can calculate the SIL as follows:

Center Frequency Hertz	Sound-Level dB
500	54
1000	58
2000	52
TOTAL	164

Three-band Average: 54:7 dB SIL — as specified in our example. Interfan uses the more complete seven band average based on bands that have center frequencies of 125, 250, 500, 1000, 2000, 4000, and 8000 Hz.

In referring to figure 1 there are a series of parallel dashed-line curves. They are curves for the human ear, using a reference tone at 1000 Hz. These curves may be used as a basis for rating the effective loudness of a noise. For example, the NR5O curve shows that 54 dB level at 500 Hz sounds just as loud as a 50dB (NR5O) 1000Hz tone.

In the sound-level meter, the A-weighted curve characteristic is based on the NR-40 curve — in fact, it is the reciprocal of NR-40. There are limitations of the SIL rating - especially when taken alone, as the sole criterion for judging fan noise output.

Why Worry About Fan Noise?

First of all, noise is distracting and unpleasant and can adversely affect human performance (and therefore work output) in several ways. Noise can mask and interfere with speech communications, leading to misinterpretation of spoken orders or loss of information. For example, a softly spoken remark like "Don't touch that switch until I remove the short across the output," spoken from about 7 feet away, has a sound-level in the neighborhood of 40 dB (A). At the other end of the intensity scale, the average human threshold of sound induced pain is 144 dB sound pressure level. The U.S. Department of Labor regulations establish a limit of 90 dB (A) for 8-hour exposure. (Use caution if using more than one fan.) Remember that sound is measured logarithmically, and you cannot add two sounds arithmatically.

But loss of communication and safety hazard are only two of the problems created by noisy fans. Noise well below the "hazard" level can cause fatigue and errors. Prolonged discussions in an excessively noisy environment can be tiring, degrading judgement. Studies have shown that the effect of noise on work output depends greatly upon the nature of the work. The task requiring close attention for a long operation cycle is especially vulnerable to noise, with likely resultant higher rates of operator error and product rejects.

But acoustical annoyance is only a part of the story. The operating environment may not be able to tolerate noise. For example, sound studios operating recording equipment, or laboratories containing seismological-sensitive, delicate instrumentation. (Noise is received not only by human ears and conventional microphones.) Many devices are "microphonic." Components in a fan-cooled enclosure may function as vibration pickups and many devices become fatigued by vibration.

Fan-generated vibration can cause failure of plug-in connectors, especially on printed-circuit boards. There is persuasive evidence that under certain conditions failures in plated-through holes in printed circuit boards are actually caused (or made possible) by crystallization of the metal film due to sustained mid-frequency vibration. So, ironically, the fan you installed to ensure cool, reliable Operation may actually cause subtle, but not less catastrophic, failure in connectors or printed circuits. The noise output of a fan transmits three kinds of energy:

- Acoustical, transmitted directly from the fan through air.
- Acoustical, transmitted through excitation of the supporting structure.
- Inaudible vibration, transmitted through the supporting structure

Vibration is an oscillatory motion in a mechanical system. A fan is a source exhibiting torsional or twisting vibration, and, at rotational speeds of 60 to 120,000 rpm gives corresponding vibrations in the frequency range of 1 or 2 to 2000 Hz.

Fan vibration may come from three sources. First there is the pneumatic pulsation of the frame caused by the pressure variations on each part of it, as the impeller blades go by; second there is some combination of electromagnetostrictive vibration and electromagnetic lamination "buzz"; third there is mechanical dynamic unbalance vibration produced by the fan rotor in its bearings, and a consequent flexure of the housing and its supports. How does the fan designer minimize the noise from these sources?

- Pneumatic Pulsation—Blade-angle design and the rotor material are the key to low noise at a given speed. The quietest fans, all other factors being equal, seem to be those that are all metal, including both the housing and the impeller blades.[1] Metal blades are stiffer than plastic blades, and therefore exhibit less elastic deformation for a given pulsation pressure. They also transmit less vibration because they may be mounted to the structure to create a stiffer, higher mass, having less resonance in the audio region.
- Electromagnetic and Magnetostrictive Effects—Low induction levels greatly reduce the noise energy output of the motor. (This tradeoff may increase the motor drain by a Watt or two out of 20, but is the reduction in electrical noise worth the sacrifice?)
- Unbalanced Vibration—Near-perfect balancing is the answer to this problem, and all-metal fans are much more stable after balancing. They won't creep, warp, or otherwise distort with age, so they stay balanced. Given a low vibration level, the rigidity of the housing structure is the key to avoiding resonance and transmission of the vibration.

1. If you doubt this compare nylon guitar strings to metal strings — or a nylon-string tennis racket to a steel-string racket of the same gauge.

If speech interference is the only noise effect you worry about, then SIL ratings are not only useful, but conclusive, If you worry about fatigue, errors, distractions, safety, or morale, then you should use the A-weighted reading and the peak Octave reading (highest point on the curve of figure 1), and you should consider the possibility of sympathetic resonances in surrounding structures. If you worry about reliability of the equipment cooled by the fan, or about poor signal-to-noise ratios due to microphonics, then you should study both the rigidity of the housing structure and the low-frequency noise output levels.

Finally, whatever criteria you combined in order to select a quiet fan, remember that fans are normally tested in an isolated condition, not rigidly mounted to the enclosure. Without going through all the implications of this test technique, a fan can always be made quieter by proper mounting on a structure that loads the acoustical and vibration sources.

Section 4
Selecting for Minimum Magnetic-Field Interaction

This step in the design sequence involves verifying that the magnetic field radiation of the fan will be tolerably low, or that the field orientation is benign, for your application

Because any magnetic device, such as a power transformer or a fan drive motor, has an external magnetic field, you must consider the effects of the fan motor's magnetic field radiation on circuit components, modules, cabling, and on other circuits and circuit elements in the enclosure and perhaps out side of it, too. Magnetic interference can appear, for example, on the display of a cathode-ray oscilloscope, causing at least two undesirable effects. First, it can interfere directly with the CRT-user's interpretation of test signals displayed; and second, it can cause the operator to fret over the question: Is what I see caused by the circuit under test, by the test equipment, or by something else (figure 1)?

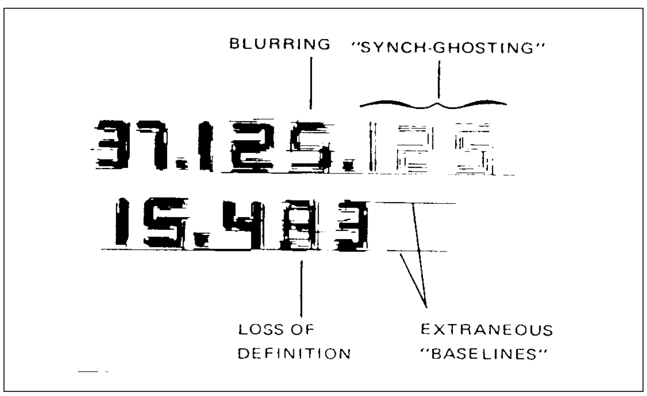

Figure 1. Display of CRT with Typical Test Signal and Magnetic Radiation Noise.

If the magnetic radiation is not appearing as noise directly on the CRT display, perhaps there is enough low-level pickup in, say, a circuit amplifier, to change the circuit characteristics, and maybe that will puzzle the electronic circuit designer. Or perhaps, there is enough voltage induced in the circuit cabling to alter the net signal enough to produce characteristics different from those designed or expected. Here are the factors that determine the magnitude of magnetically induced voltage signals.

$$e_{ind} \;\alpha\; \Phi \times f \times N \qquad \text{(eq 1)}$$

In equation 1, Φ is the quantity of magnetic flux, represented by lines called lines of induction. By convention, the number of these lines per unit area at right angles to their direction is defined as the magnitude of the induction, and the induction is expressed in "lines per unit area". Lines of induction are called Maxwells, or Webers (1 Weber = 10^8 Maxwells), and magnetic induction is expressed in Maxwells per square centimeter, or in Webers per square meter. One Maxwell per square centimeter is called one Gauss.

The total number of lines of induction threading through a surface, the magnetic flux Φ, in general is:

$$\Phi = \int B \cos\theta \, d A \qquad \text{(eq. 2)}$$

where B is the magnetic induction or flux density in Maxwells per square centimeter, or Webers per square meter, and A is the area. In the special case in which B is uniform and normal to a finite area A, then

$$\Phi = BA \qquad \text{(eq. 3)}$$

Generally speaking, however, we can only say that the voltage induced into a closed path, (the input circuit loop of an amplifier, or a wire loop in a cable wire), depends on the effective area, A $\cos\theta$. Thus, if $\theta = 60^\circ$ then $\cos\theta = 0.5$ and the area effectively coupled to the field flux is reduced by 1/2 from the theoretical maximum voltage.

In equation 1, "f" represents the frequency, usually of the power line driving the fan, for example 50 Hz, 60 Hz, or 400 Hz; plus the harmonics (usually odd) thereof. The value "N" is usually one turn, since most closed loops into which the fan field will couple are single-turn configurations.

For areas measured in cm^2, and N = 1 turn, and B in Gauss, the induced voltage is given by:

$$e_{ind} = 4.44 \; f \, AB \cos\theta \times 10^{-8} \; Volts \qquad \text{(eq.4)}$$

The graphs of figures 2 and 3 show the field strength "maps" made by a search coil carried around a fan at a distance of 2.5" from the shaft center at the blade end. The field is small, because the fan uses the preferred "inside/out" (or "inside/outside") motor design. Figure 2 illustrates the flux-density distribution in the plane of the blade rotation. , and the other the flux-density distribution in the plane of the rotor shaft.

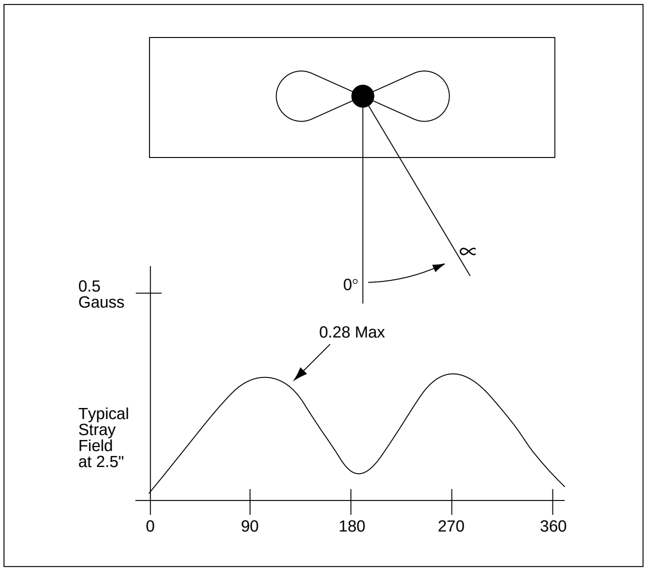

Figure 2. Flux Density in Plane of Blade Rotation

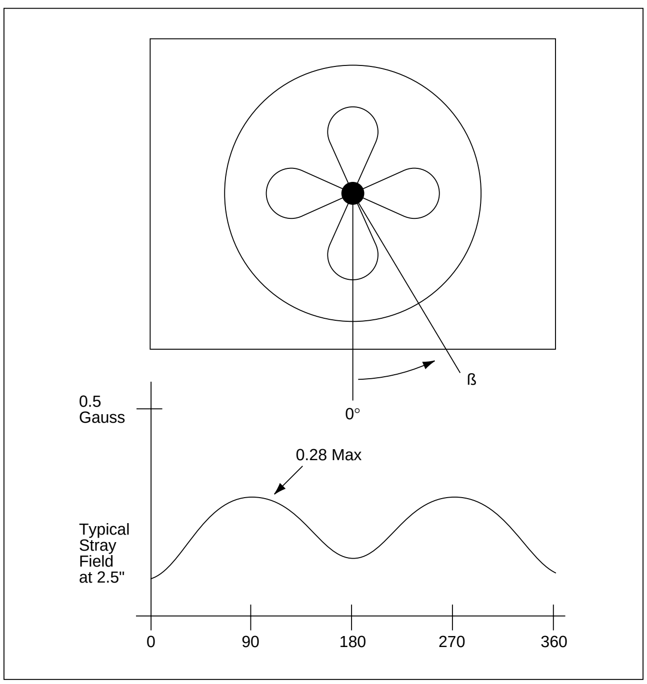

Figure 3. Flux Density in Plane of Rotor Rotation

To get some idea of the effect of magnetic induction on a practical application, even with this low-field-strength, inherently self-shielded design, consider the following example:

A = 20 cm^2 (loop from amplifier input “high” to ground, through input resistor, and back from ground to “low”)

$$B = 0.16 Guass \quad \alpha = 150° \ \beta = 90°$$

$\theta = 30°$ (Orientation of loop with respect to field B)

N = 1 turn, of course

f = 60 Hz (harmonics assumed negligible for this example).

From which we calculate:

$$e_{ind} = 4.44 \times 60 \times 20 \times 0.16 \times 0.866 \times 10^{-8}$$

$= 7.4 microvolts$ or approximately 10 microvolts peak

In a low-level amplifier input circuit, this could be disturbing, indeed.

For a conventional fan motor, the field might have been 10 - 100 times as large, and the orientation would most certainly have created a peak near the loop in question. Forewarned is forearmed.

So far, we have discussed the malign effects of magnetic radiation and the factors that proportion magnetically induced voltages. Here is how to reduce them. Short of removing the fan entirely, presumably impossible, here are some ways to reduce the amount of magnetic radiation.

- Rerouting —Either reposition or reorient the fan or the circuit-element into which the undesired and intolerable field has been induced. Perhaps moving a component a short distance, or turning it slightly will remove the induced signal, or reduce it to an acceptable level. Often, the position and orientation of the fan is fixed or restricted by packaging or airflow considerations. Unfortunately, the magnitude of magnetic flux is, in most conventional fans, at a maximum in the path and direction of airflow, where, usually, the components, modules, and cabling we wish to protect from flux are most likely to be located. (More on this later. Recheck figure 2, however, and you will see that directly in front of the fan, the stray field is at its minimum. This is true only of fans using the inside/out motor design.
- Shielding—Another effective way to eliminate or reduce induced magnetic noise is by shielding. This may mean “protecting” the component or module by a metal casing. Perhaps an entire section can be “cordoned off”. Sometimes, low-noise performance may be achieved by the judicious use of shielded cable.

Why, don't we shield the motor itself? Indeed, the inside/out design (figure 4) and all-metal housing and rotor effectively constitute a self-shielded motor. This reduces the electromagnetic radiation to low intensities so that circuitry in the equipment being cooled is much less affected by proximity to the fan. But shielding of a conventional motor is generally not possible, because the shielding blocks the airflow.

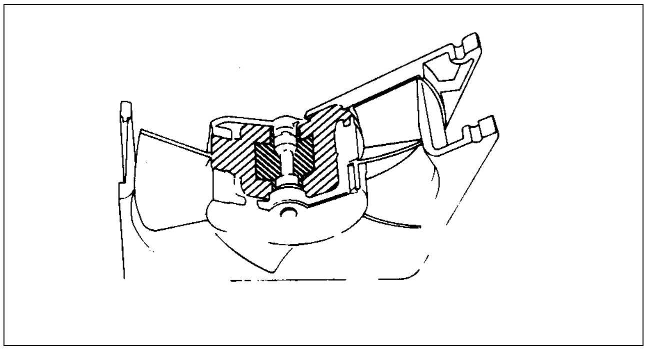

Figure 4. Inside/outside Motor Design

- Reshaping the Magnetic Field—This technique uses shorted-turn coils, usually single turns, known as shading coils, which when properly placed, set up a magnetic field in opposition to the one created by the motor. The overall effect is to reshape the motor radiation field and move the field away from circuit elements. A continuously looped copper ribbon, or a copper washer, properly placed, can achieve a 20 dB reduction in pickup at the sensitive point.
- Using a Lower-Intensity Source—As noted earlier in the discussion of acoustical noise, conservatively designed motors operate at lower flux densities, hence have less saturation—greatly reducing the leakage field intensity

As described re-routing or repositioning are the best ways to avoid electromagnetic interference. Here are a few reminders and suggestions that you may find worthwhile, at least as a checklist.

- Power Supplies usually need the most direct access to the airstream, and can usually tolerate electromagnetic radiation better than most components — but remember that even power supplies have regulating ("error") amplifiers, and the signal level in them is of the order of 100 to 500 Usually, however, this amplifier has great tolerance for 60 Hz pickup, and is well "buried" in the center of a steel chassis.
- Bus cabling—particularly flat "ribbons" of paralleled cable conductors - are best kept far from the fan plane. Indeed, all cards facing a fan should be wired at the opposite edge, if possible. If cards must be "back-wired," consider a front-entry fan location.
- Grounding may be more of a trouble than a help if the grounding "bus" forms a long, high-area pickup loop, as it so often will. Watch out for "induced" ground loops. Avoid them by running all grounding paths in planes perpendicular to the main flux field.
- One way to reduce all magnetic pickup, regardless of source, is to reduce the area of the pickup path. This sketch (figure 5) is worth a thousand words:

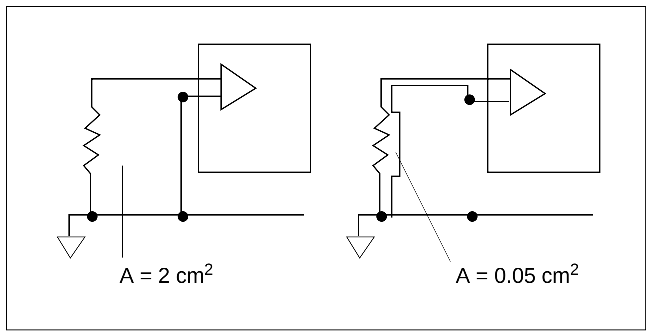

Figure 5. A pickup reduction of 40:1, merely by rerouting one lead.

- Start with a low-field fan, and you may save hundreds of man-hours of troubleshooting, redesign, rework and even repackaging.

Section 5
Evaluating the Performance of the Complete Cooling System

By now, you have narrowed the selection of a suitable fan for your application - perhaps down to only one design. Or you may even have designed yourself into a corner, and have rejected all the available standard designs—each for one or more good reasons.

Regardless of your situation at this point in the design, the time has come to re-examine it from a systems viewpoint, and check its validity. (If you've run out of available fans, or practical combinations of available fans, you may want to look for primary relief—a new set of assumptions that will re-open the lists to some nearly acceptable designs.)

The most likely place to attack a stubborn design is in the temperature-rise assumption. Why did you decide that the temperature of the exhaust air could not be more than 25˚ higher than the highest incoming ambient? If you trace this decision back to its source (in your mind or someone else's) you will usually find that one or two motivations were at work.

One motivation might be fear that the outgoing air would cause discomfort to personnel, or “overcome the room air conditioning,” or “add to an already high ambient.” These fears are groundless, as we shall see, if the application is at all practical, regardless of the fan or cooling system used. A second motivation might be concern over the highest “hot-spot” temperature that will be reached, over and above the highest cooling-air temperature. This may or may not be a rational cause for concern, but usually some-thing can be done to improve the situation.

First, dispose of the room-heating and air-conditioning questions. If the system must move 900 Watts out of the cabinet into the room it does not matter (on an energy basis) whether this wattage comes into the room at 200 CFM with only an 8˚C temperature rise, or at 50 CFM with a 32˚C temperature rise—900 Watts is 900 Watts. The air conditioning (natural or mechanical) must accomodate this load.

Your design should not direct a blast of hot air at persons in the room, but that problem is easily handled by simple deflecting vanes, or a primitive dispersal duct, or even a stovepipe. Therefore, as long as you can mix the hot exhaust air with the rest of the room air before humans (or other sensitive mechanisms) are exposed to it, the exhaust temperature can't affect the average room temperature — either way.

The second reason for limiting temperature rise may be more difficult to dismiss. To understand the relationship between "hot-spot" and air stream temperatures we shall use the simple model of figure 1, and the corresponding electrical analog, figure 2.

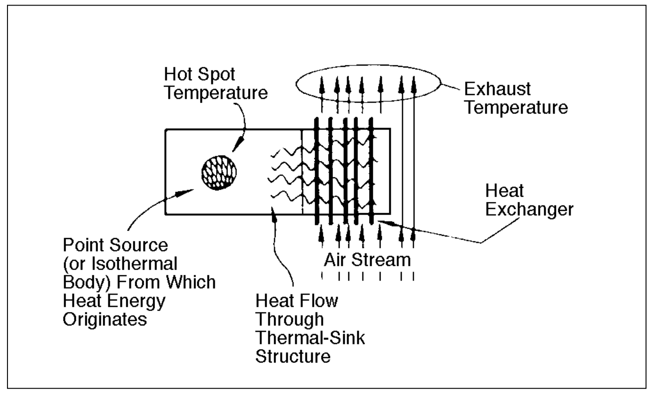

Figure 1. Representation Of Relationships Among Heat Source, Heat Exchanger, And Cooling Air Stream.

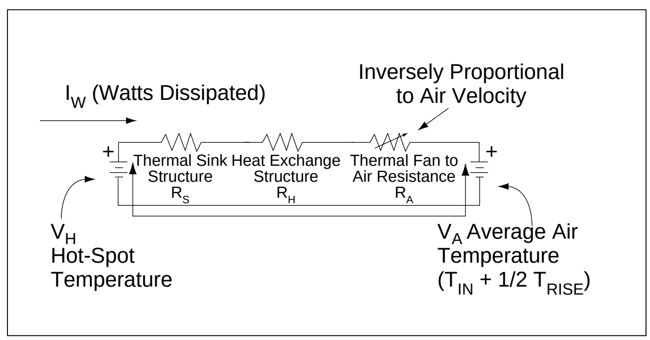

Figure 2. Electrical Analog of Figure 1

In figure 1 the temperature of the energy source (the “Hot-Spot” temperature) is higher than that of the exhaust air-stream — but how much higher? Examination of figure 2 shows that the hot-spot temperature can be minimized (held as close as possible to the exhaust air temperature) by minimizing three thermal resistances (measured in ˚C/Watt):

1. The resistance of the thermal path from the heat source to the far end of the thermal sink.

2. The net resistance of the thermal paths from the heat sink body to the fin surfaces.

3. The net resistance of the thermal path from the heat-exchange fins to the air sweeping over them.

Of these three thermal resistances, the last is the largest by far and it is inversely proportional to the velocity of the air passing through the fins. Therefore, the really significant factor in holding down the Hot-Spot temperature is attaining a high enough velocity of air through the heat exchanger. How can we increase that velocity? There are many ways -ranging from the simple expedient of relocating the heat-exchange structure nearer to the main airstream, through the use of deflector vanes or constricted ducts, to the "last-ditch" addition of an internal, local fan mounted on or near the heat exchanger. These and other configurations are suggested by the sketches of figure 3.

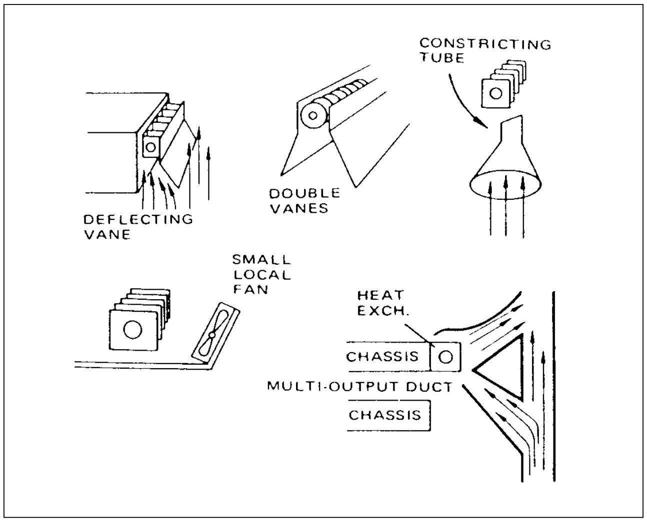

Figure 3. Increasing the air velocity through the heat exchanger.

Having increased the velocity, we can allow ourselves a higher temperature rise, which permits a lower airflow rate (CFM is inversely proportional to temperature rise, you will recall) and that means, generally, that more fans will be capable of satisfying our requirement.

The one exception to this sequence of events is the case in which the vanes, ducts, or other expedients used to increase the velocity through the heat exchangers have increased the pressure drop to so marked an extent that the usable standard fans are once again few, if any. This rarely happens, fortunately.

Of much greater concern than vanes or ducts (which usually intrude on only a part of the total air-stream) are devices that intercept all of the airflow. These include:

- Filters
- Safety Guards of all kinds
- Decorative Grills of all kinds
- Exit Ports - escutcheons, venturis, simple-cutouts, etc.
- Entry Ports - same as Exit Ports

Every one of these interposed objects exacts its toll in pressure drop. Here, in figure 4, we show a typical pressure-volume relationship for a cooling system. Note the relative contributions of the individual components, and note particularly the fact that the basic enclosure is less than one-half of the total pressure drop!

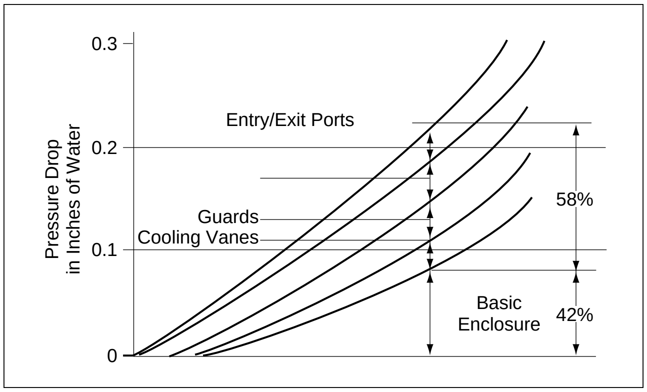

Figure 4. Typical Pressure Drops Related to Accessories

What can be done about these necessary but unpleasant features of an airflow system? Once again, intelligent design can help significantly to reduce the effect of these mainstream pressure drops. Here are some observations that will suggest practical remedies to the alert designer:

- The lower the velocity through a filter, a guard, or any other obstruction, the lower the pressure drop. Clearly, for a given airflow in CFM, the velocity is inversely proportional to the area transverse to the airflow. If you are feeding air through a filter, the larger the filter, the lower the pressure drop through it. (And here's a bonus - at lower velocities, the filter efficiency is much higher. You may elect to use one large filter, or two or more small ones in parallel.
- The further you locate a guard away from the blades, the "looser-mesh" may be its design. If you set the guard back 2", you may be able to use a few thin wires or slats, instead of a finer mesh. If you are determined to let nothing larger than a flea through the safety guards, however, you might consider an oversized guard (see sketch in figure 5) which is the equivalent of the filters "in parallel," mentioned above.

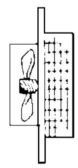

Figure 5. Oversize Guard Has Lower Pressure Drop.

- Lowering the velocity of the airstream through or past an obstruction reduces the noise it generates.
- An abrupt transition of any kind (direction, area, cross-sectional profile) from one part of an airstream to another creates far more pressure drop than a smooth, gradual transition. Look at the venturi in an axial fan, if you want a good example of good practice. Surprisingly, figure 6b has lower pressure drop at moderate to high CFM than figure 6a, although its average cross-section is markedly smaller, and the mean airflow path is longer.

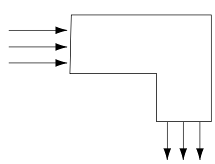

Figure 6a. Abrupt transition in direction and cross section.

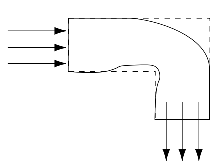

Figure 6b. Gradual Transition in Direction and Cross Section.

- Sheet metal cutouts, unless preceded or followed by venturi-like air guides, are sharp transitions. Figure 7a has more pressure drop than figure 7b, despite larger effective area.

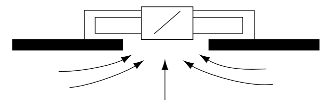

Figure 7a. Abrupt entry cutout.

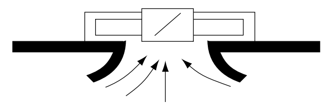

Figure 7b. Gradual entry port (venturi-like).

- Decorative grilles should be treated like guards or filters, and the principles of figures 5-7 should be applied. Take a look at the slat-grille designs on modern air-conditioning units - especially the small window designs. They can't spare pressure drop either, and they minimize it in the necessary decorative trim by following the design practices suggested.

If your back is to the wall and you've tried everything, but the fan you want to use just won't do what you want, you might need to try for a new or modified enclosure design that achieves optimum utilization of the fan air-moving capabilities. Alternatively, you might want to know how to select a fan that is being fully loaded.

This analysis, although based on an "idealized" fan characteristic, is a close approximation, for most common axial and similar fans. The analysis begins by assuming that the pressure-versus-air flow relationship is linear between intercepts (figure 8).

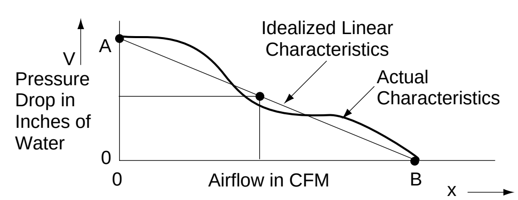

Figure 8. Straight-line approximation to actual fan characteristic

The general equation for this straight line is:

$$x = B\left(1 - \frac{Y}{A}\right) \qquad \text{(eq. 1)}$$

The work done by the fan (the energy output) is proportional to the area xy.

Multiplying equation (X) by y, we get:

$$xy = By - \left(\frac{B}{A}\right)y^2 \qquad \text{(eq. 2)}$$

To find the values of x and y (the operating point) at which xy is a maximum, we first differentiate equation (2) with respect to y:

$$\frac{d(xy)}{dy} = B - 2\left(\frac{B}{A}\right)y \qquad \text{(eq. 3)}$$

and set the derivative equal to zero:

$$\frac{d(xy)}{dy} = 0 = B - 2\left(\frac{B}{A}\right)y \qquad \text{(eq. 4)}$$

and solve for y:

$$y = \frac{A}{2}$$

then we substitute this value into equation (1), and find that:

$$x = \frac{B}{2}$$

Therefore, the maximum air-moving energy delivered by the idealized fan occurs at:

$$airflow = \frac{B}{2} CFM$$

$$pressure\,drop = \frac{A}{2} \; inches\ of\ water$$

pressure drop = A inches of water

This not very startling result is exactly analogous to the conclusion that a generator delivers maximum power when the load resistance equals the internal resistance.In fact, if you call the “internal resistance” of the fan B/A, and find a cooling system with a resistance to airflow of exactly B/A inches per CFM, then you can load the fan to maximum capability.

But our problem is of an opposite nature. We know the airflow resistance—what we need to find is a fan having intercepts B and A that yield a matching ratio.

Well, that would be lucky indeed — but you can actually change the required airflow resistance over wide ranges, by altering the internal velocities, and thereby allowing a higher temperature rise, if necessary. Therefore, you can almost adjust the cabinet airflow resistance to B/A.

In the limiting case, the allowable temperature rise is nearly from ambient to the "hot-spot" temperature. Therefore, the “analytical” approach to forcing an optimum design is as follows:

1. Take the temperature rise as only about ten degrees less than hot-spot minus ambient (highest ambient, of course).

2. Calculate CFM from the total Watts of dissipation and that optimistic temperature rise.

3. Calculate or estimate the pressure drop of the enclosure, including whatever velocity-boosting things are needed to ensure good heat transfer.

4. Find a fan in which B/2 is close to or slightly larger than the CFM calculated in (2) above, and A/2 is close to, or slightly larger than the pressure drop estimated or calculated in (3) above. That's the fan that will be optimally matched to this requirement.

You may not need (or want) to live quite so “dangerously,” but at least you will know exactly how small the fan's power could safely be. Larger fans, properly “loaded” (matched) will then be safer for this application.

Remember that the higher the temperature rise of the air in the system, the more efficient the design. You will probably want to trade efficiency for added hot-spot margin, but that should not be done until you are certain that you know where the margin begins.

Section 6
Ensuring Optimum Reliability in the Complete Cooling System

Following the advice in preceeding sections, your fan candidates should be narrowed to those that meet:

- Required airflow and back pressure
- Size, shape and style
- Noise levels
- Magnetic radiation levels
- Compatibility with the complete cooling system.

The final step is to pick the most reliable fan of the candidates remaining. (Not simple, but essential.) Normally, to do this, you would check the first standard gauge of reliability rating that comes to mind: the published ratings of the eligible fans for Mean Time Between Failure (MTBF). And when you do, you will realize that MTBF is not enough of a distinction.

Although MTBF ratings are a necessary element in measuring reliability, they are not in themselves a sufficient condition of dependability. To be so, an MTBF computation would have to measure all likely design and manufacturing characteristics that affect dependability—and they do not. To be sure, they do measure a few important characteristics, such as projected life of winding insulation, expected lubricant life, and hot-spot temperature rise in the motor windings. With our experience in the application of fan material and configurations, Interfan has designed fans that have reassuringly high MTBF ratings.

Our high MTBF ratings were achieved through an investment of man-years of attention to lubrication design and to quality assurance in bearing manufacturing. On the design level, noteworthy developments have included new synthetic lubricants, provision of extra-large reservoirs, extra large bearings, and design of leak-proof end-seals. Beyond design improvements, several other factors have helped to eliminate bearing failure in Interfan fans —assuming rated conditions. These include extremely tight tolerances in all shaft and bearing surfaces, and multiple inspection characteristics, during which each dimension is subject to at least two independent verifications. With respect to bearing failures caused by dynamic unbalance Interfan has unique automatic balancing equipment, with the capability of micro-inch dynamic balancing. Our normal manufacturing standard is balancing to within 5 micro-inches on both end-planes.

Rotor-shaft freezing is a common failure mode in poorly constructed fans. Such failures are probably due to dry or misaligned bearings. But bearing failure does not have to be catastrophic to be adverse. Just listen to (and feel) the noise and vibration output of the rumbling, poorly designed fans of some competitors that were so quiet and smooth when new, less than a year ago.

We are convinced that our clear-cut superiority is traceable to our design and manufacturing conservatism in the use of superior materials - conservatism in the establishment of margins of safety - conservatism in the insistence on tight tolerances. And yet, we have not rested on our enviable record of reliability in service. We continue to develop new, improved structures and materials.

What does all this admittedly self-serving information mean to the designer faced with selecting the optimum fan for his application? Just this. What you will probably find, after surveying all of the technical material literature on fans, is that 90 percent of all your applications point to a small group of fans, and this group will include two or three Interfan fans. In this 90 percent of your applications, our fan will give you most, if not all, of the following clear-cut advantages:

- Interfan fans will usually exhibit lower noise.
- Interfan fans will exhibit substantially less magnetic field radiation, and the orientation of that field will be more tolerable than that of other fans.
- Interfan all-metal fans will be stronger and more rigid than most of their counterparts.
- Despite all this, our fans will be competitive in real cost:
- If we are stating the case correctly, you won't need our assurances of extraordinary dependability. But if you do, we want you to know that we sincerely believe that our fans have the best reliability record of any in their class.
- On further note- You have the right to ask: To do all this — to get a better preforming fan at less money, where did they cut corners? And the simple answer is that we did not. Our superiority was achieved through superior design, manufacturing and tooling efficiency, and pride - in the integrity of our designs, and the consistent perfection of our products.

Looking Back . . .

We have now examined, in some detail, all six topics in this series. Each is a logical step in the design procedure we recommend. There are six steps altogether and each step has been explained in a major paragraph of this application note:

1. Determine the required airflow and back pressure.

2. Select a size, shape and style of fan that will fit, will be easy to assemble, and will direct the air flow properly.

3. Verify that the SIL (Speech Interference Level), and the corollary vibration and microphonic effects as well, of the available fan will be acceptable in your application. If not, repeat steps 1, 2 and 3, until a fan is found that satisfies all three.

4. Verify that the magnetic field radiation of the fan will be low enough, or that its orientation is benign, for your application. Methods of altering field orientation and of reducing field strength, are among the alternatives considered previously.

5. Design the tentatively selected fan into a complete cooling system, making it as elaborate as need be. Then, check your estimated back pressure maximum (see step 1). If necessary, repeat steps 1-5, using a fan capable of higher (or lower) back pressure.

6. Consider the acceptable available fans -the ones that pass all the tests in steps 1-5. Then pick the most reliable one.

Interfan Glossary of Terms

AC (Alternating Current) Has a frequency associated with it and does not require polarity connection.

Auto Restart After either a locked rotor or a “brown-out” power failure, the fan automatically restarts. In older DC fans, the fan would not restart until you disconnected the power and reconnected it once again, or grounded out the input leads. Auto restart fans are considered standard today.

AWG (American Wire Gauge) Relates to the size of a wire. The larger the number, the thinner the wire.

Axial Flow See Tube Axial Flow.

CFM (Cubic Feet per Minute) Cubic Feet per Minute air flow. The measure of the volume of air flowing through a given cross-sectional area per minute of time.

CSA (Canadian Standards Association) An independent Canadian product safety testing agency.

dB (Decibel) The standard unit of measure on a log scale used to express the ratio of two levels. Used here, it represents the ratio of a sound pressure to that of the threshold of human hearing for a given frequency.

dB(A) The amount of noise based on frequencies the human ear can pick up. The frequency response is weighted to discriminate against the lower frequencies because the human ear perceives these as less intense. This is the “A” scale.

DC (Direct Curent) Has no frequency assoicated with it and must have polarity considered in connection.

EMI (Electromagnetic Interference) Disturbances caused by electromagnetic waves that can impair the reception of the desired signal. (Shielding materials for the entire EMI spectrum are not readily available.)

Frequency Usually refers to the AC voltage applied in fans. It is usually 50 or 60 Hz.

Humidity Ratings Represents the maximum humidity a fan should be employed in. Most Interfan fans are rated for 90% relative humidity.

Impedance Protected For AC fans, this is a way of protecting the fan from an over voltage or over current.

Impeller The “propeller” or moving portion of the fan.

Lead Wires/Pigtails Electrical wire connections to the fan circuit.

L-10 Life Also known as Weibel Analysis Curve. The theoretical life of a fan based on the life of the bearings with respect to temperature. Represented as a curve wrt. temperature. The higher the temperature, the lower the viscosity of the lubricant on the bearings, therefore, the lower the life of the bearings.

MTBF (Mean Time Between Failure) Mean time between failure. The probability that a device will fail with respect to time in operation. This data is usually gained by comparing the number of failures that come back from the field to the number of fans actually employed in systems originally. Not the same as life expectancy.

Pascal Metric unit of pressure = $1N/m^2$.

Polarity Protected For DC fans, they are protected from an opposite polarity power hook up.

PBT A plastic known as Polybutylene Terephthalate.

Radial/Centrifugal Flow	Air flow along or parallel to the radius of the fan.
RFI (Radio Frequency Interference)	Electromagnetic radiation in the radio frequency spectrum from 15kHz to 100GHz. The best shielding material against RFI is copper and aluminum alloys. The term “EMI” should not be used in place of RFI since shielding materials for the entire electro-magnetic frequency spectrum are not available.
RPM (Revolutions Per Minute)	The number of times an impeller rotates in one minute.
SIL (Speech Interferance Level)	Commonly used rating method for noise. Based upon averages of sound pressure levels on audible frequencies of 500, 1000 and 2000 Hz.*
Static Pressure	The amount of resistance to air flow. This is considered the amount of force the fan can handle.
Terminal Block	Male-type connection on the fan housing used instead of lead wires.
Tube Axial Flow	Air flow along the axis of rotation of axial fan because it “simulates” a venturi tube.
UL (Underwriters Laboratories)	An independent non-profit product safety testing agency.
VDE	A German independent product safety testing agency.
Venturi	Another word for the housing of the fan.
Weibel Analysis Curve	See L-10 Life.

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