Fan Curves and Laws

HOW TO USE THEM IN THERMAL DESIGN

In today's electronics industry, there is a constant and well documented push to higher powered components, tighter grouping of devices, and overall increased system thermal dissipation. The higher dissipation must be managed effectively to ensure long term reliability of the system.

With forced convection being the dominant mode of electronics cooling, more efficient heat sinks are often used for cooling these increased thermal loads. But, they are only half the solution. Due to volumetric constraints, it may not be possible to design an adequate heat sink for a given component. A large amount of air preheating may occur if multiple components lie in the flow path. The increased ambient temperatures resulting from this preheated air often bring the need for a larger heat sink, but the space may not be available.

The solution to higher power levels and decreasing heat sink space is to increase the system's air flow rate. A boost in flow rate has a twofold benefit: first, it lowers the thermal resistance of the heat sink, which reduces the temperature differential from junction to ambient. Secondly, it reduces the overall temperature rise in the chassis. The reduced temperature rise allows downstream components to suffer less preheating and operate at acceptable temperatures.

This direct relationship between air velocity and component temperature indicates the importance of understanding how fans behave in electronics cooling.

System Curve

Prior to selecting any fan it is important to characterize the overall system with respect to air flow and pressure drop. For example, a tightly packed 1U chassis will require a much different fan configuration than a larger desktop one, even if both systems use the same CPU. In the 1U chassis, components are spaced very tightly and exhibit a large resistance to flow. This requires a fan with a high pressure drop. A benefit of the 1U chassis design is less inefficient bypass flow, reducing the need for larger volumetric flow rates. But in an ATX style desktop chassis the requirements are very much the opposite. There is typically much more open space in the ATX chassis, which lowers the chassis pressure drop. The widely spaced components create a less efficient flow path, and thus a larger volumetric flow is needed to ensure adequate cooling of all components.

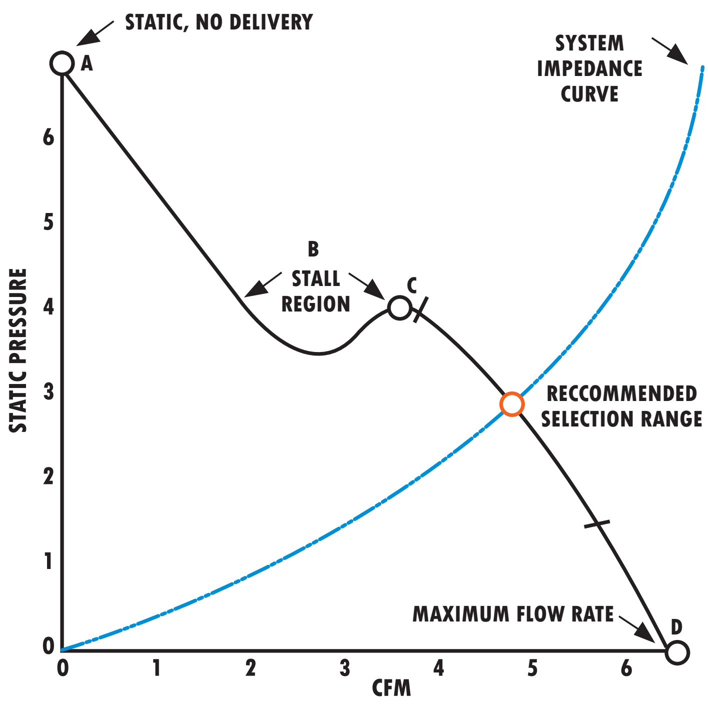


Figure 1. Typical Overlay of a System Curve and Fan Curve.

Fan Curve

A fan curve example is shown in Figure 1. Point A is the "no flow" point of the fan curve, where the fan is producing the highest pressure possible. Next on the curve is the stall region of the fan, Point B, which is an unstable operating region and should be avoided. From point C to point D is the low pressure region of the fan curve. This is a stable area of fan operation and should be the design goal. It is best to select a fan that operates to the higher flow area of this region to improve fan efficiency and compensate for filter clogging.

The system pressure curve can then be compared to a specific fan curve to determine if the fan will be adequate.

AMCA Fan Standard

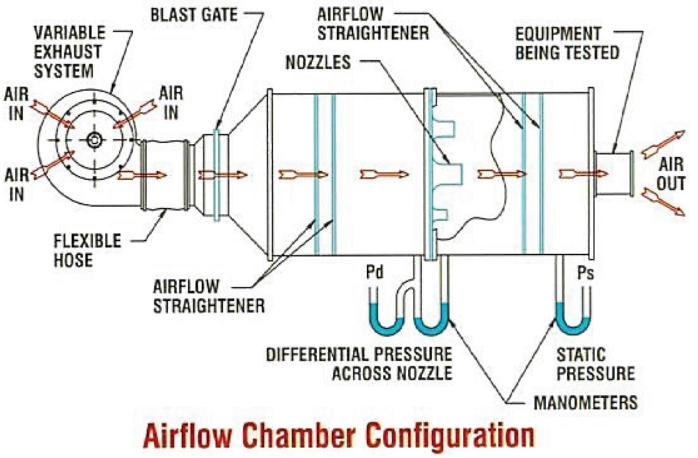


Figure 2a. AMCA Fan Testing Chamber.

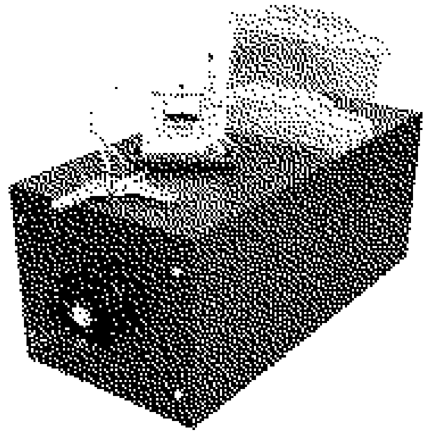


Figure 2b. FCM-100 Fan Characterization Module from Advanced Thermal Solutions.

To compare fan curves from different manufacturers, it is important to follow a testing standard. For electronics applications, the relevant standard is the AMCA 210-99/ASHRAE 51-1999 test guidelines.

The AMCA fan testing chamber, shown in Figure 2a, consists of a supply fan, a variable blast gate, two test chambers, flow nozzles and an opening to place the test fan. A commercialized testing module from Advanced Thermal Solutions, Inc. is shown in 2b.

During a typical fan test, a dozen or more operating points are plotted for pressure and flow rate, and from this data a fan curve is constructed. To obtain the highest pressure rating of the fan, the blast gate shown in Figure 2a, is closed to ensure zero flow while the fan is running. The chamber pressure is then read from the static pressure manometer to obtain the maximum pressure rating of the fan. The blast gate is then slightly opened in successive steps to obtain additional operating points. Finally, the maximum flow capability of the fan is found by opening the blast gate completely and running the supply fan. The supply fan ensures the secondary chamber is operating at atmospheric pressure, which removes the flow losses in the system.

Figure 3. Various AMCA Nozzles (CTS, Inc.)

The operating pressure of the fan curve is found by taking measurements from a static manometer. The volumetric flow rate, Q, is found by measuring the pressure drop across an AMCA nozzle (Figure 3) using a differential manometer. The flow rate through an AMCA nozzle is a function of its size and differential pressure as shown in the following equation.

In contrast, the FCM-100 is void of any nozzles and works based on volumetric flow rate measurement using the ATVS technology flow sensing system. It is compact, portable and capable of characterizing single fans or fan trays.

Air Flow

$$G = 60 \times C \times \frac{\pi}{4} \times D^2 \times \sqrt{2g/(r \times 0.10197 \times P_n)} \quad \text{m}^3/\text{min}$$

Where:

C = Coefficient of nozzle air flow

D = Diameter of nozzle (m)

r = Air density $(1.293 \times \frac{273}{273+t} \times \frac{P}{1013.25})$ kg/m^3

t = Temperature (°C)

P = Air pressure (hPa)

P_n = Differential pressure of air flow (Pa)

g = 9.8 m/s^3

Fan Laws

Fan laws are a set of equations applied to geometrically indentical fans for scaling and performance calculations.

Volumetric Flow rate: $G = KqND^3$

Mass Flow Rate: $\dot{m} = K_m ND^3$

Pressure: $P = K_p N^2D^3$

Power: $HP = K_{Hp} N^3D^3$

Sound: $L_{w2} = L_{w1} + 55_{\log 10}(\frac{N_2}{N_1})^1 + 55_{\log 10}(\frac{D_2}{D_1})^1 + 55_{\log 10}(\frac{P_2}{P_1})$

Where:

K = constant for geometrically and dynamically similar operation

G = volumetric flow rate

$\dot{m}$ = mass flow rate

N = fan speed in RPM

D = fan diameter

HP = power output

ρ = air density

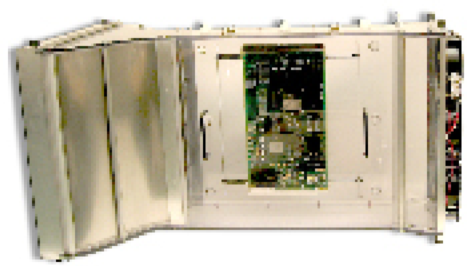
L_W = sound level, dB

ATS' laboratory grade bench top wind tunnels are designed with polynomial shapes to provide highly uniform flow for accurate characterization of components, circuit boards and cooling devices such as heat sinks, heat exchangers and cold plates. Each features ports to accommodate a variety of probes including thermocouples, Pitot tubes and temperature and velocity sensors. Each bench top wind tunnel is designed to be lightweight and compact and feature Plexiglas® text sections for clear views of the specimens and flow visualization.

LABORATORY GRADE BENCH TOP WIND TUNNELS

BWT-104

- Produces flow velocities from 0 to 6 m/s (1200 ft/min)
- Test Section Dimensions: 50.8 cm x 17.25 cm x 10 cm (20 x 17.25 x 4")

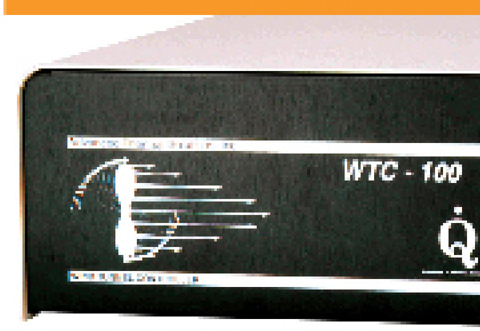


BWT-100

- Produces flow velocities from 0 to 2 m/s (400 ft/min)
- Test Section Dimension: 21.6 cm x 25.4 cm x 2.5 cm (8½ x 10 x 1")

WTC-100

- Measures temperatures from -10°C to 150°C (±1°C)
- Capable of controlling velocities from up to 50 m/s (10,000 ft/min) depending on the fan tray
- Features a user friendly, labVIEW based, application software



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Published fan laws apply to applications where a fan's air flow rate and pressure are independent of the Reynolds number. In some applications, however, fan performance is not independent and thus the change in Reynolds number should be incorporated into the equation. To determine if the Reynolds number needs to be considered, it must first be calculated.

$$\mathrm{Re} = \frac{\pi \rho N D^2}{C_R \mu}$$

ρ = Density (kg/m^3)

N = Speed (Rev/Sec)

D = Fan Diameter (m)

C_R = Correction (1)

μ = Absolute Viscosity (N-s/m^2)

According to AMCA specifications, an axial fan's minimum Reynolds number is $2.0\mathrm{x}10^6$ When the calculated Reynolds number is above this value, its effects can be ignored.

Fan Law Application

During a product's life cycle a redesign may be carried out which replaces older components with new, higher powered ones. Due to the resulting higher heat flux, increased cooling is often needed to maintain adequate junction temperatures and reduce temperature rise within the system.

Consider for example a telecomm chassis using a single 120 mm fan for cooling. The maximum acceptable temperature rise in the box is 15°C. The chassis dissipates 800 W, but a board redesign will increase the power to 1200 W. The current 120 mm fan produces a 3 m³/min flow rate at 3000 RPM using 8 W of power. How do we calculate the requirements of a substitute fan for the higher powered system?

First, calculate the required G (m^3/min):

$$G = \frac{0.05(P)}{\Delta T}$$

G = ~4

Where:

G = Required volumetric flow rate (m³/min)

P = Power dissipated (Watts)

ΔT = Temperature rise from inlet to exit (°C)

Next, calculate the change in RPM needed:

$$RPM_2 = (\frac{G_2}{G_1}) * RPM_1$$

$$RPM_2 = (\frac{4}{3}) * 3000$$

$$RPM_2 = 4{,}000 \text{ rpm}$$

And finally, calculate the change in fan power:

$$HP_2 = HP_1(\frac{RPM_2}{RPM_1})^3$$

$$HP_2 = 8W(\frac{4000}{3000})^3$$

$$HP_2 = 18.9 \text{ W}$$

Thus, to meet this example's cooling requirement for 1200 W, a fan is needed with a 4 m³/min flow rate, 4,000 RPM speed and 18.9 W of power. Note that the system power, flow rate and fan RPM all increased in a linear fashion from those in the original system. However, the fan power increased by nearly a factor of three.

Summary

Bulk testing of electronics chassis provides the relationship between air flow and pressure drop and determines the fan performance needed to cool a given power load. The fan rating is often a misunderstood issue and published ratings can be somewhat misleading. Knowledge of fan performance curves, and how they are obtained, allows for a more informed decision when selecting a fan. Continued and ever shortening product design cycles demand a "get it right the first time" approach. The upfront use of system curves, fan curves and fan laws can help meet this goal.

References:

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Electronics Cooling, October 1995.

2. Daly, W., Practical Guide to Fan Engineering, Woods of Colchester, Ltd, 1992.

3. Turner, M., All You Need to Know About Fans, Electronics Cooling, May 1996.

4. Certified Ratings Program - Product Rating Manual for Fan Air Performance, AMCA 211-05 (Rev. 9/07).