Technical Paper Reprint From
ASME Winter Annual Meeting, 1990

Effects of Electronic Enclosure Layout on Fan Performance

HILL ENGINEERING
6349 Nancy Ridge Drive, Suite A
San Diego, California 92121
858/552-1140
858/453-0668 FAX
info@hillengineering.com

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47 St., New York, N.Y. 10017

90-WA/EEP-6

Effects of Electronic Enclosure Layout on Fan Performance

THEODORE B. HILL, P.E. and CHARLES C. HILL, P.E.
Hill Engineering
San Diego, CA

ABSTRACT

This paper presents the results of experiments which show the effects of electronic enclosure cooling fan plenum size on fan performance. Plenum depths ranging from one-tenth of a fan diameter to over two times the fan diameter were tested. Typical small, axial flow fans were used. The fans were alternately arranged to both evacuate and pressurize the enclosure. Total volume flow of air was measured for each configuration with an orifice plate flow meter system. Static pressure inside the plenum was measured so the fan performance curve for each geometry could be established. Results are shown as a function of the parameter P/D (plenum depth:fan diameter ratio). Significant reduction in fan performance was noted at P/D ratios below 0.2. The results should aid the designer in selecting fans that provide closer agreement between specified design flow and actual installed fan performance.

NOMENCLATURE

D - Diameter of Fan Tip Shroud, mm
N - Fan Speed, RPM
N_s - Specific Speed
P - Plenum Depth, mm
P_s - Static Pressure, mm of water gage
Q - Volumetric Flow Rate, L/s
t - Fan Axial Thickness, mm

INTRODUCTION

Fan manufacturers, out of necessity, publish fan performance curves as measured in a test apparatus with no appreciable flow restrictions at the fan inlet or outlet. As early as 1932, Marks and Winzenburger had shown that inlet or outlet disturbances can have substantial effects on fan performance. In large centrifugal fans, as used in power plants and heating, ventilating and air conditioning systems, these effects have been quantified and incorporated into standardized industry test codes by the Air Movement and Control Association (1985).

Whitaker (1973) investigated the effects of ducted inlets and outlets on the performance of a large axial flow fan. Roslyng (1984) investigated the loss in performance compared to manufacturer's test data when a large axial flow fan was installed in a real system. Pace (1984) found that fans installed in a military vehicle delivered only 90% of their rated static pressure because of installation effects.

The small, axial fans commonly used to cool contemporary electronic enclosures are affected by inlet and outlet disturbances as larger units are. Quantitative information about inlet or outlet plenum effects on these fans has not been available to guide their application to compact enclosures. The results presented here are a first effort at quantifying some of the effects of electronic enclosure arrangement on fan performance.

Decreased component size has tended to increase power density in electronic systems. The trend toward more compact systems and the desire to shorten product development time makes it essential that early and correct judgments are made in relation to cooling system needs.

We recommend a sequence of design tasks in determining a satisfactory air cooling system for an electronic enclosure that include global, macro and micro analysis and testing to achieve the required performance. We define the global tasks as those which determine and cause the proper mass flow of air to remove the total heat rejection of the system with the specified temperature rise in the cooling air flow. The macro tasks we define as those which determine and cause the proper air flow distribution to the components within the enclosure so sufficient air flows to each component or region to transport the locally rejected heat. The micro tasks are those associated with ensuring that component thermal resistances are compatible with rejecting the locally generated heat to the cooling air without exceeding allowable component temperatures.

Presented at the Winter Annual Meeting
Dallas, Texas—November 25-30, 1990

The results reported here apply to the global task. The cooling system designer, given the total expected heat rejection for the enclosure, will calculate the cooling air mass flow for the desired air temperature rise. Based upon this flow, the designer must then calculate or estimate the total system flow resistance including pressure drops of inlet or outlet grilles and internal pressure drops to arrive at a static head requirement for the fan. Having done this, a cooling fan can be selected, by reference to the manufacturer's data, that has at least the required static head at the necessary flow. Using the information presented here, the designer can estimate the potential reduction in fan performance that might be expected if shallow depth inlet or outlet plenums are present in the design.

EXPERIMENTAL ARRANGEMENT

Fan performance was measured for a range of plenum depths in an enclosure simulating a 133 mm tall rack mount cabinet. The enclosure was made to telescope, allowing the plenum depth to change. Panels of differing percent open area were used to form the wall of the plenum. We investigated the performance of two different axial flow fans. Measurements were made for both flow directions -- pressurizing and evacuating the enclosure.

Air flow was measured with an orifice plate flow meter system (Figure 1). The flow meter complies with the requirements of the ASME Standard MFC-3M-1989, *Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi* (1990). The flow meter is bi-directional and has a variable speed blower to overcome the pressure drop through the orifice run. Figure 2 presents the calculated uncertainty of the flow measurement. Uncertainty in the flow measurement comes from two sources. The flow and expansion coefficients defined in the Standard are one source of uncertainty. Measurement uncertainty of orifice and pipe dimensions, orifice differential pressure and fluid density also contribute to the flow uncertainty. The flow meter system is instrumented with a manometer for orifice plate differential pressure measurement and a J type thermocouple for measurement of the flowing air temperature. Accuracy of these instruments is incorporated in the total uncertainty for the flow meter system.

The simulated electronic enclosure is shown in Figure 3. The geometry is based on a rack mount cabinet with front panel mounted fans. The enclosure has space for four circuit boards stacked horizontally. Initial testing was done with four circuit cards installed to establish a baseline system flow resistance. Panels with various percent open area replaced the circuit boards during actual tests to define the extent of the plenum space and to offer a resistance to air flow similar to that of the circuit boards. The fan performance measurements presented here were all made with removable panels of varying open areas that presented flow resistances similar to the circuit cards.

Two commercially available small, axial flow fans were tested. Different fan diameters were selected to ascertain if the plenum depth:fan diameter ratio (P/D) would be a useful parameter for relating plenum size and fan performance. Table 1 summarizes the characteristics of each fan. These particular fans were selected because of their similar diameter:thickness ratios (D/t) and specific speeds. Specific speed is defined as

$$N_s = NQ^{0.5}P_s^{-0.75} \qquad (1)$$

An electronic micromanometer (Dwyer Instruments, Inc. Microtector) was used to measure the static pressure inside the enclosure plenum relative to atmospheric pressure. The micromanometer has a resolution of 0.013 mm of water gage. Repeatability is within 0.013 mm wg. We have estimated the uncertainty of the static pressure measurement as not exceeding 0.025 mm wg. An array of six static pressure taps was mounted in the enclosure wall. The static taps were connected in parallel to give an average static pressure measurement. However, probing of the plenum space revealed that the static pressure did not vary significantly throughout the plenum region. The tap diameter was 1.32 mm.

The experimental procedure for determining the fan capacity curves is as follows:

1. The enclosure was set up with the particular fan, flow direction, plenum depth and flow resistance panel under consideration.

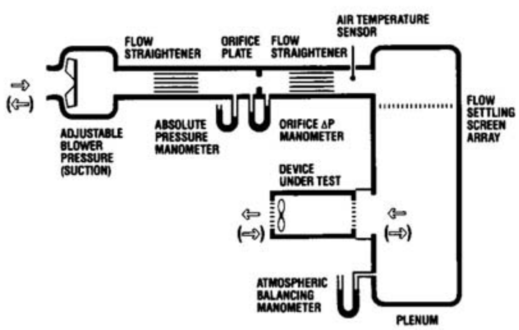

Figure 1. Orifice Plate Flow Meter System

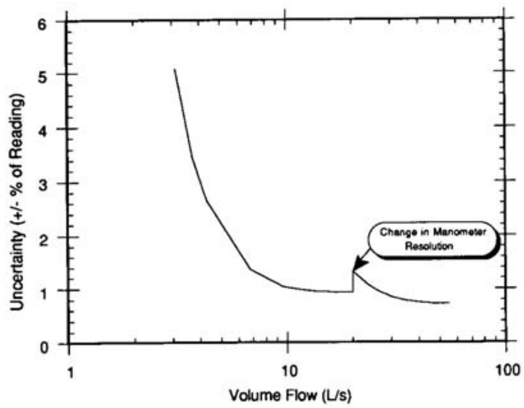

Figure 2. Flow Meter Measurement Uncertainty

Table 1 Axial Flow Fan Characteristics

	Fan A	Fan B
Tip Shroud Diameter, D [mm]	77.1	112.9
Hub Diameter [mm]	36.1	52.3
Thickness, t [mm]	25.0	38.9
D/t	3.08	2.90
Number of Blades	7	3
Speed, N [RPM]	3100	3100
Specific Speed, N_s (at 75% of free flow)	9960	8990
Electrical Input Power [W]	2.2	5.6
Rotation Direction (viewed from inlet)	CCW	CCW
Motor Support Strut Location	Outlet	Inlet
Free Inlet/Outlet Performance:		
Max. Static Pressure, P_s [mm wg.]	4.09	7.11
Free Flow, [L/s]	17.7	47.5

2. Flow through the flow meter was increased until the free delivery point (P_s=0) was detected.

3. About eight equally spaced flow increments between free delivery and shut off are determined.

4. Data (Q, P_s) is taken at each of these points as well as the shut off static pressure.

The free inlet/free outlet performance of each fan was measured in a similar fashion. The fans were mounted on the wall of the flow meter system plenum box (Figure 1). The plenum box has an internal volume of 220 liters, effectively an "infinite" plenum space for fans of this size.

RESULTS

The ratio of plenum depth, P, to fan tip shroud diameter, D, was selected as the parameter for comparing the effects on fan performance. Initial investigation of P/D ratios up to 2.0 revealed that ratios greater than 1.0 did not yield significantly better fan performance than a P/D of unity. Test results are presented for P/D ratios of 0.1, 0.2 and 1.0 as well as baseline fan performance curves obtained with no obstructions on either inlet or outlet.

The fan performance results shown graphically in Figures 4 through 8 have been made nondimensional by dividing by the free inlet/free outlet maximum static pressure and free flow shown in Table 1.

Figure 4 shows how the fan performance curve is affected by the percent open area of the plenum wall. In this series of tests, the enclosure geometry and fan size and flow direction were constant. Three plenum wall panels of varying open area were alternately tested. Two of the panels were constructed of wire mesh mounted on a thin frame. The wire mesh panels had open area percentages of 30.3% and 46.2%. A third panel, this one with slots, had an open area of 35%. As expected, fan performance improved as open area increased. The slotted panel was selected for further testing since it most closely resembled a typical card cage structure.

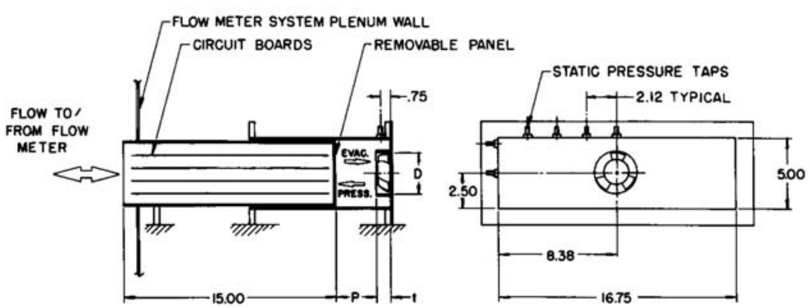

Figure 3. Experimental Arrangement of Simulated Electronic Enclosure

Figures 5 through 8 show the fan performance curves obtained for each fan and flow direction combination for the three plenum depth ratios. The baseline fan performance curves are also reproduced on each graph for reference. Some trends in the data are apparent.

1. The evacuating flow direction is more susceptible to degraded fan performance at low P/D ratios than the pressurizing flow direction.

2. The plenums acted to improve the head capacity of the fans near the "knee" in the fan curve and near the zero flow points when the flow direction was pressurizing the enclosure.

3. Fan B (the larger fan) was not affected by the lower P/D ratios as much as Fan A for either flow direction.

4. In all cases, a P/D ratio of 1.0 had a minimal effect on the fan performance compared to the baseline performance measured with free inlet and outlet conditions.

DISCUSSION

At this point, we cannot offer a functional relationship between fan and plenum geometry and the extent of the fan performance loss. Further work extending to more fan sizes and plenum configurations needs to be done. The chaotic nature of the air flow in and near an axial flow fan may make it impossible ever to develop a generalized relationship. Additional areas for study also include velocity distribution effects and noise generation.

The work performed here should prove useful to designers faced with similar enclosure geometries. We caution that our results should not be generalized beyond the range of fan sizes tested.

To avoid reduced fan performance from plenum geometry effects, we recommend that a P/D ratio of at least 0.25 and preferably 0.5 be maintained. If the enclosure design requires a shallower plenum depth, a fan with sufficient head capacity in this reduced performance region must be selected.

REFERENCES

Air Movement and Control Association, 1985, *Fans and Systems*, Publication 201, , Arlington Heights, Illinois.

American Society of Mechanical Engineers, 1990, *Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi*, ASME MFC-3M-1989, New York.

Marks, L.S. and Winzenburger, E.A., 1932, "Influence of Inlet Boxes on the Performance of Induced-Draft Fans," *Transactions of the ASME*, Vol. 54, No. 21, pp. 213-220.

Pace, I.S., 1984, "Cooling System performance in Fighting Vehicles," *Installation Effects in Ducted Fan Systems*, Institution of Mechanical Engineers Conference Publications 1984-4, C112/84, pp. 47-54.

Roslyng, O., 1984, "Installation Effect on Axial Flow Fan Caused by Swirl and Non-Uniform Velocity Distribution," *Installation Effects in Ducted Fan Systems*, Institution of Mechanical Engineers Conference Publications 1984-4, C114/84, pp. 21-28.

Whitaker, J., 1973, "Fan Performance Testing Using Inlet Measuring Methods," *Proceedings of the Institution of Mechanical Engineers*, Vol. 187, No. 33/73, pp. 405-412.

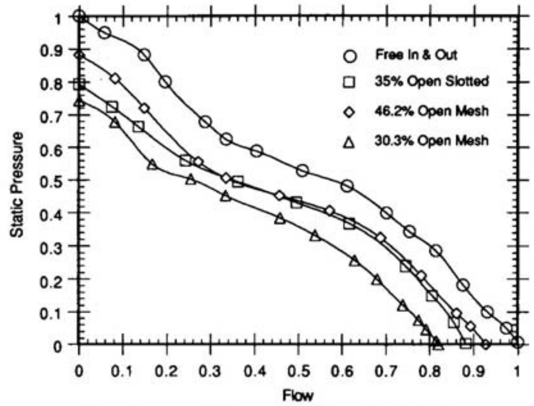

Figure 4. Plenum Wall Open Area Effects
(Fan B Evacuating - P/D = 0.1)

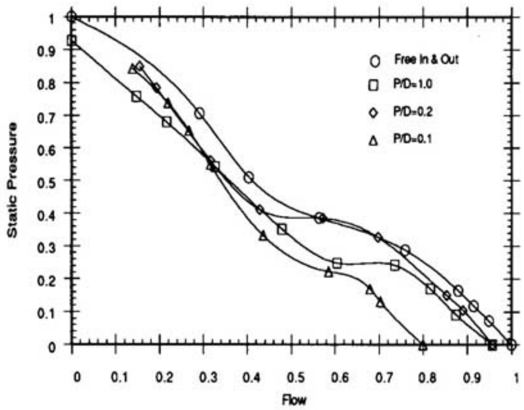

Figure 5. Fan A Pressurizing
35% Open Area Slotted Panel

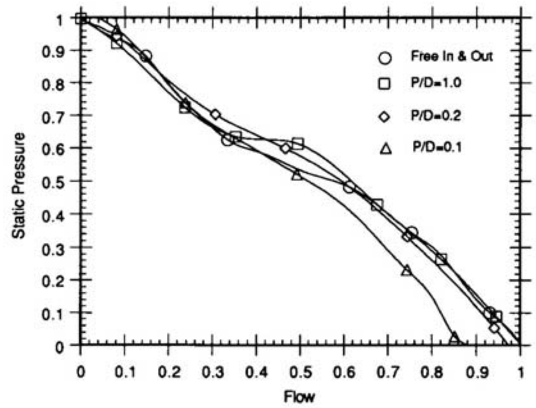

Figure 7. Fan B Pressurizing
35% Open Area Slotted Panel

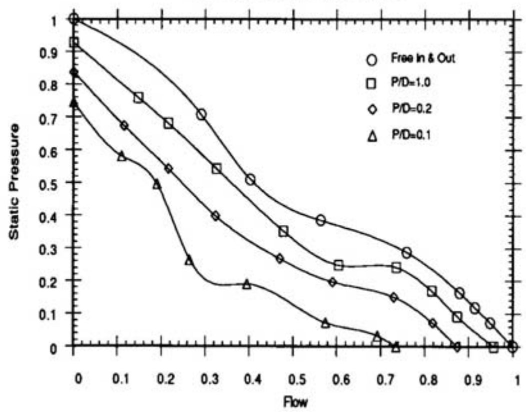

Figure 6. Fan A Evacuating
35% Open Area Slotted Panel

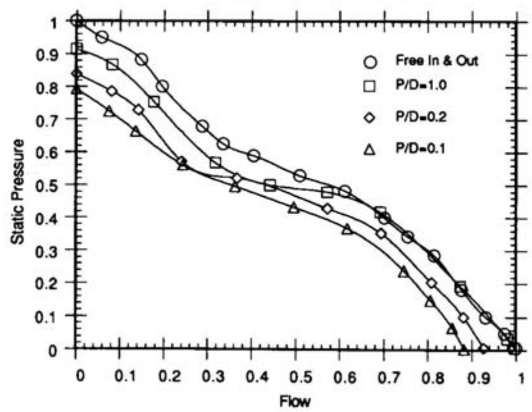

Figure 8. Fan B Evacuating
35% Open Area Slotted Panel