RATING OF GRAVITY ROOF VENTILATOR DRAFT

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ABSTRACT

A roof ventilator is often used to increase the draft of a chimney or a ventilator pipe. The goal of this work was to establish a method for testing and rating ventilator draft. It was found that air pressure near the ventilator inlet must be measured as a function of both the external wind speed and the flow rate of air through the ventilator.

Data are given for three ventilator types and for an open pipe. Results show that ventilator draft is (1) increased by separation of the wind stream as it flows over a ventilator and (2) decreased by the pressure losses as air exits the ventilator. Draft can be found by matching a ventilator performance curve to a system curve for a given building. This is similar to finding the operating point of a fan. Dimensional analysis suggests that ventilators can be rated using two dimensionless groups: a pressure coefficient and a loss coefficient.

INTRODUCTION

Both powered and gravity roof ventilators are widely used on commercial and residential buildings. Powered types ventilate with a motor-driven fan. Gravity types ventilate due to the buoyancy of hot exhaust air and the suction produced as wind blows over the ventilator. Figure 1 shows examples of gravity ventilators.

This study considered only gravity ventilators. The goal was to understand the ventilation enhancement or draft produced by these devices. Since powered ventilators were not studied, the term "ventilator" will mean gravity ventilator for the remainder of this paper.

Although the skyline of most cities reveals hundreds of ventilators, there has been little technical study of ventilator draft. We found only a single article (Wendes and Pannkoke 1981), and it only provided qualitative descriptions of various ventilator types. Manufacturers, if they provide data, give the volume flow rate produced by a ventilator for a fixed wind speed. Sometimes a temperature difference between ambient and ventilation air is also given. We found this representation of data to be of little value, and the reasons should be clear from the results reported in this paper.

There are several reasons for studying ventilator drafting. First, there is presently no good way to size a ventilator for a given application. This, in turn, produces backdrafting problems in wood stoves and heating appliances, and the resulting smoke spillage poses a health hazard (Moffat 1986). This problem is especially acute for occupants of tightly sealed houses. Also, the lack of a way to characterize ventilator draft makes comparison between different ventilators impossible. Finally, the lack of a systematic way to test draft makes it difficult to quantify the effects of design improvements. The goals of this work were:

1. To determine what data best characterize the drafting performance of a ventilator.
2. To design, build, and test an experimental apparatus for gathering these data.
3. To gather representative experimental data and interpret these data in the context of ventilator cap performance.
4. To find a way to use these data to predict ventilation provided by a ventilator.

ANALYSIS

Building Model

To find out what data characterize draft, ventilation air flow through a building was modeled. Figure 2 shows a simple building ventilated by a single vent pipe. It was assumed that

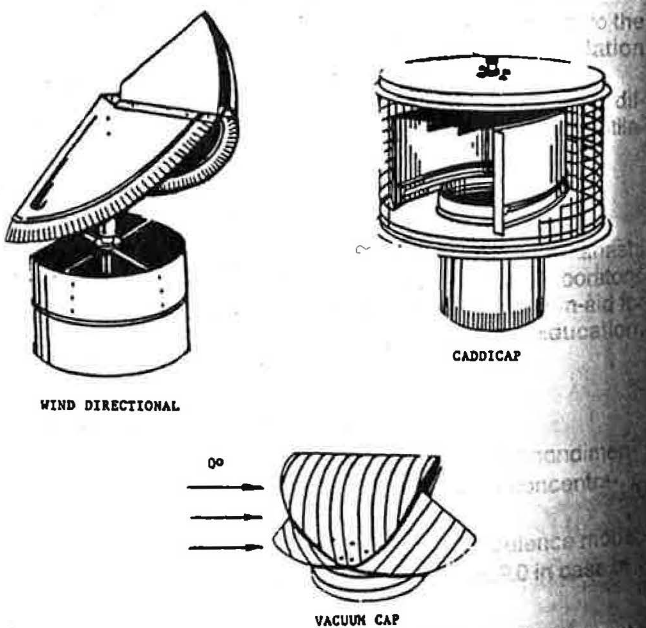

Figure 1 *Pictorial diagrams of the ventilators tested. Both the caddicap and the wind directional cap pivot to align with the wind. The vacuum cap was tested in two orientations: (1) the wind is normal to open end (0° wind angle), which is shown in the figure; and (2) the wind is normal to the flat side (90° wind angle).*

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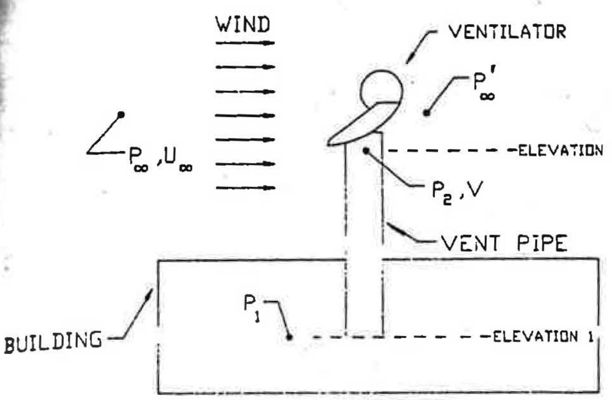


Figure 2 *A building ventilated by a single vent-pipe*

wind over the ventilator is steady in magnitude and direction. The vent pipe is the only path for air to leave the building, so the mass flow of air leaving through the ventilator is balanced by infiltration air flow. The air flow was assumed to be steady. Air infiltration into the building was modeled using the usual relationship (Kiel et al. 1985),

$$Q = C(P_{\infty 1} - P_1)^n \quad (1)$$

where

Q = air leakage rate, cfm (L/s),
C = flow coefficient, cfm/psi^n (L/(s·Pa^n)),
P_1 = average pressure in the building at elevation 1,
$P_{\infty 1}$ = average ambient pressure at elevation 1,
n = flow coefficient, typically between 0.5 and 1.

Using fluid statics between elevations 1 and 2 gives

$$P_{\infty 1} = P_{\infty 2} + \rho_\infty gh \quad (2)$$

where

ρ_∞ = air density evaluated at the outside air temperature,
h = vertical distance between elevations 1 and 2.

Applying Bernoulli's equation between elevation 1 in the building interior and elevation 2 in the vent pipe gives

$$P_1 = P_2 + \rho V^2/2 + \rho gh + (4C_f)(L_e/D)(\rho V|V|/2) \quad (3)$$

where

P_2 = static air pressure in the vent pipe at elevation 2,
ρ = air density evaluated at the vent pipe air temperature,
V = average air velocity in the vent pipe (or vent pipe air velocity) = $4Q/\pi D^2$,
D = vent pipe diameter,
C_f = Fanning friction factor,
L_e = equivalent length of the pipe, including all minor losses,
h = distance between elevations 1 and 2.

Combining Equations 1, 2, and 3 gives the desired result,

$$\rho V^2/2 = \underbrace{\Delta P_v}_{(\mathrm{I})} + \underbrace{\Delta\rho gh}_{(\mathrm{II})} - \underbrace{(4C_f)(L_e/D)(\rho V|V|/2)}_{(\mathrm{III})} - \underbrace{(Q/C)^{1/n}}_{(\mathrm{IV})} \quad (4)$$

where

ΔP_v = ventilation suction pressure = $P_{\infty 2} - P_2$,
$\Delta\rho$ = $\rho_\infty - \rho$.

Thus the vent pipe air velocity (V), which we wish to know, depends on the four terms on the right side of Equation 4. Term I is the ventilator suction pressure, ΔP_v, which is the draft produced by the vent cap. Positive ΔP_v increases ventilation; negative ΔP_v decreases ventilation. Term II is a buoyancy term called the "stack effect." Term III gives the energy loss caused by fluid friction and other head losses in the vent pipe. Term IV gives the energy loss caused by pressure losses as infiltration air enters the building. For a given building, all terms on the right side of Equation 4 are known, except ΔP_v.

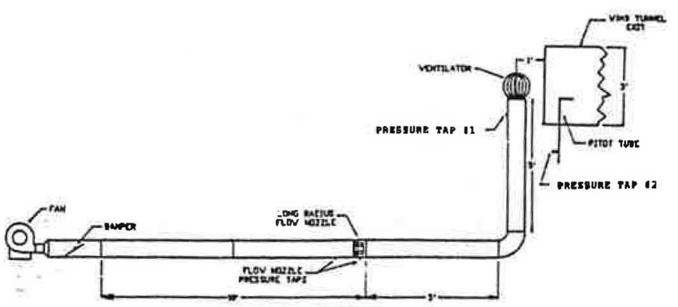


Figure 3 *The experimental apparatus*

Dimensional Analysis

For constant unidirectional wind over a ventilator, ΔP_v depends on seven variables:

$$\Delta P_v = \Delta P_v\{U_\infty, V, D, \mu_\infty, \rho, \rho_\infty, \text{ventilator geometry}\} \quad (5)$$

where

μ_∞ = viscosity of air at the ambient temperature,
D = a characteristic ventilator dimension; we used vent pipe diameter.

Performing dimensional analysis on Equation 5 gives

$$C_p = C_p\{V/U_\infty, \mathrm{Re}, \rho/\rho_\infty, \text{ventilator geometry}\} \quad (6)$$

where

C_p = $-2\Delta P_v/(\rho_\infty U_\infty^2)$ = pressure coefficient,
Re = $U_\infty D\rho_\infty/\mu_\infty$ = Reynolds number.

In summary, Equation 4 shows that ΔP_v characterizes draft. Equation 6 shows that, for a given cap, ΔP_v depends on three independent dimensionless variables.

In the experimental work, constant temperature air ($\rho/\rho_\infty = 1$) was used. For a given ventilator geometry, this implies that two independent variables need to be controlled during testing. Wind velocity (U_∞) and vent-pipe air velocity (V) were selected as these two variables.

DESCRIPTION OF THE EXPERIMENTAL APPARATUS

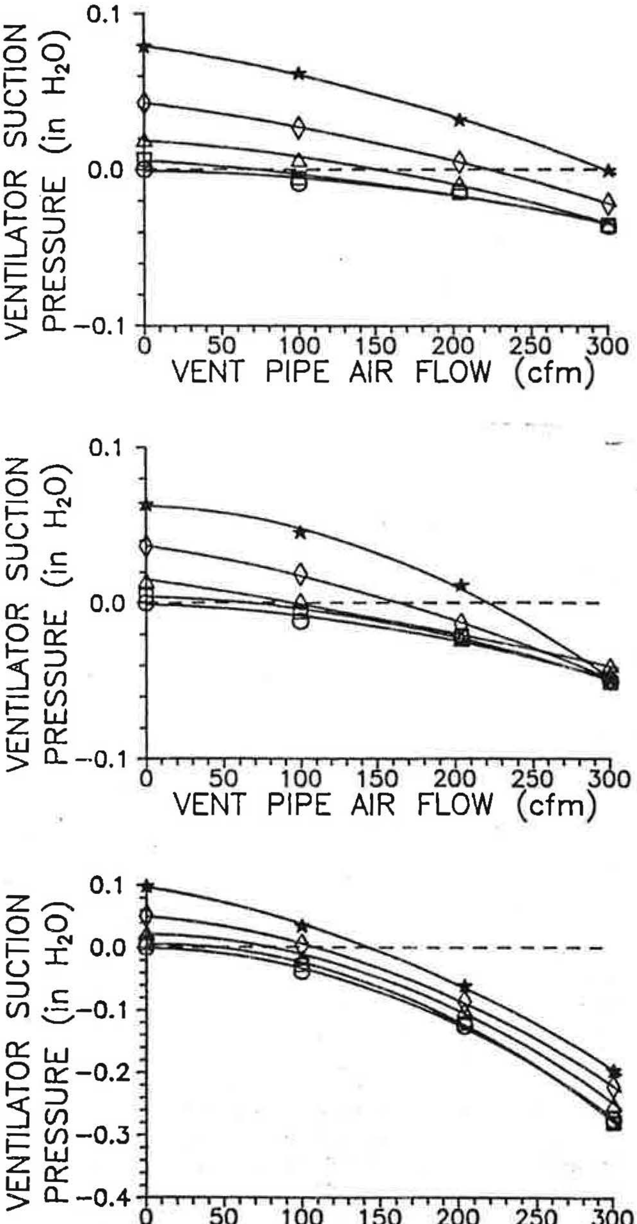
The apparatus is shown in Figure 3. ΔP_v was measured as a function of wind speed and vent pipe air velocity. A wind jet provided a steady crosswind over the vent cap, and a fan provided the vent pipe airflow. The advantage of this setup was that the wind speed and vent pipe air velocity could be controlled independently.

A conventional three-foot-square blowing wind tunnel produced the wind jet. Using a wind jet, instead of the confined flow in a wind tunnel test section, solved the problem of the ventilator restricting the flow area. In turn, this allowed testing of larger ventilators with the same size tunnel. To show that the wind jet simulated a uniform wind flow, we measured the radial velocity field. The data showed that the velocity outward from the ventilator reached the freestream wind velocity.

The ventilator testing apparatus consisted of an 11-in. centrifugal fan, a 1-hp motor, conventional HVAC ducting, and a damper. Figure 1 shows the ventilators that were tested. All ventilators fit on 7-in. ducting.

Vent pipe airflow rate was measured using a long-radius flow nozzle. ASME (1971) equations and discharge coefficients were used to calculate flow rate. Three sizes of flow nozzle were used, depending on the flow rate to be measured. This was done for two reasons (1) to produce a flow nozzle Reynolds number in the range of the tabulated discharge coefficients and (2) to reduce data uncertainty (i.e., better impedance match).

Pressure across each flow nozzle was measured with a manometer with a sensitivity of 0.02 or 0.1 in. H_2O, depending



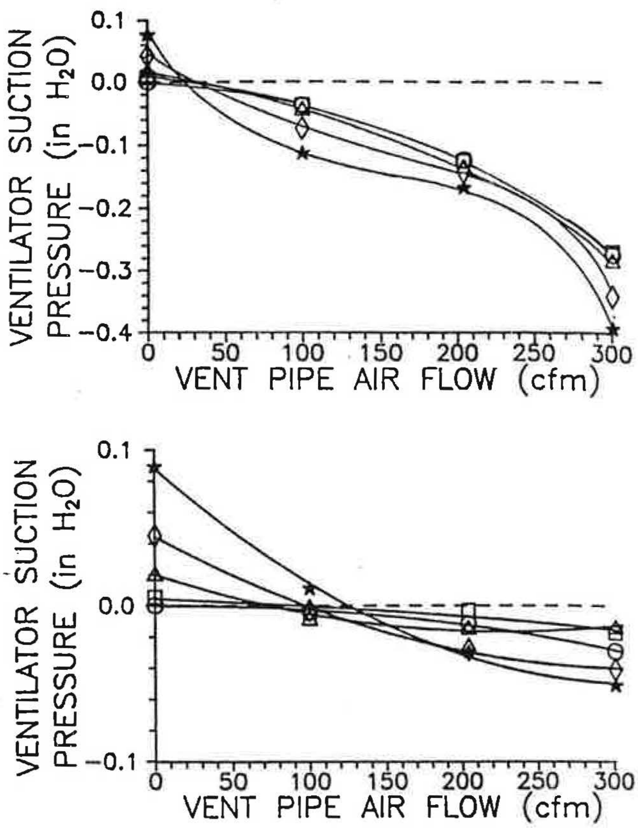


Figure 4 Ventilator performance curves. Windspeed U_∞: ★ 20 mph; ◇ 15 mph; △ 10 mph; □ 5 mph; O 0 mph. Ventilators: (a) wind directional; (b) caddicap; (c) vacuum cap; 90° wind angle; (d) vacuum cap, 0° wind angle; (e) open pipe. Typical error bars are shown on (a). The solid lines are used only to connect the data points.

on which scale was being used. Ventilator suction pressure (pressure tap #2 minus pressure tap #1) was measured with a micro-manometer with a sensitivity of 0.0008 in. H_2O. All pressure taps were built to ASME (1971) standards.

A conventional root-mean-squared calculation provided uncertainty estimates. On the average, wind speed measurements had an uncertainty of $\pm 5.8\%$, and flow rate measurements had an uncertainty of $\pm 4.4\%$.

EXPERIMENTAL DATA AND VENT CAP PERFORMANCE

Figure 4 shows ΔP_v data for three ventilators and for an open pipe. We call the downward sloping lines "ventilator performance curves." Each line gives ΔP_v as a function of vent pipe airflow rate for a constant wind speed. Positive ΔP_v means the ventilator is inducing draft; negative ΔP_v means that the ventilator is impeding draft. Clearly, ΔP_v depends strongly on both wind speed and vent-pipe air velocity.

Figures 4d and 4e look like bad data. Why? We speculate that (1) the air leaving the vent pipe is influencing the separation of the crosswind over the vent pipe and (2) the crosswind is inhibiting the air from leaving the vent pipe. This coupling effect of the two airflows produced the unusual results shown. Comparison of Figure 4c with Figure 4d shows that the coupling effect severely reduces draft.

Figure 5 shows the data of Figure 4 plotted using dimensionless groups: four curves (wind speeds of 5, 10, 15, and 20 mph) reduced to a single curve, showing that ventilator draft depends only on V/U_∞ and not on Reynolds number over the range of Reynolds numbers used. The horizontal line in each plot of Figure 5 gives the pressure coefficient behind a flat circular disk in crossflow at a Reynolds number of 10^5 (Rouse 1946). Thus, the pressure coefficient of a ventilator at $V/U_\infty = 0$ is very close to the pressure coefficient of a disk in crossflow.

DISCUSSION OF RESULTS

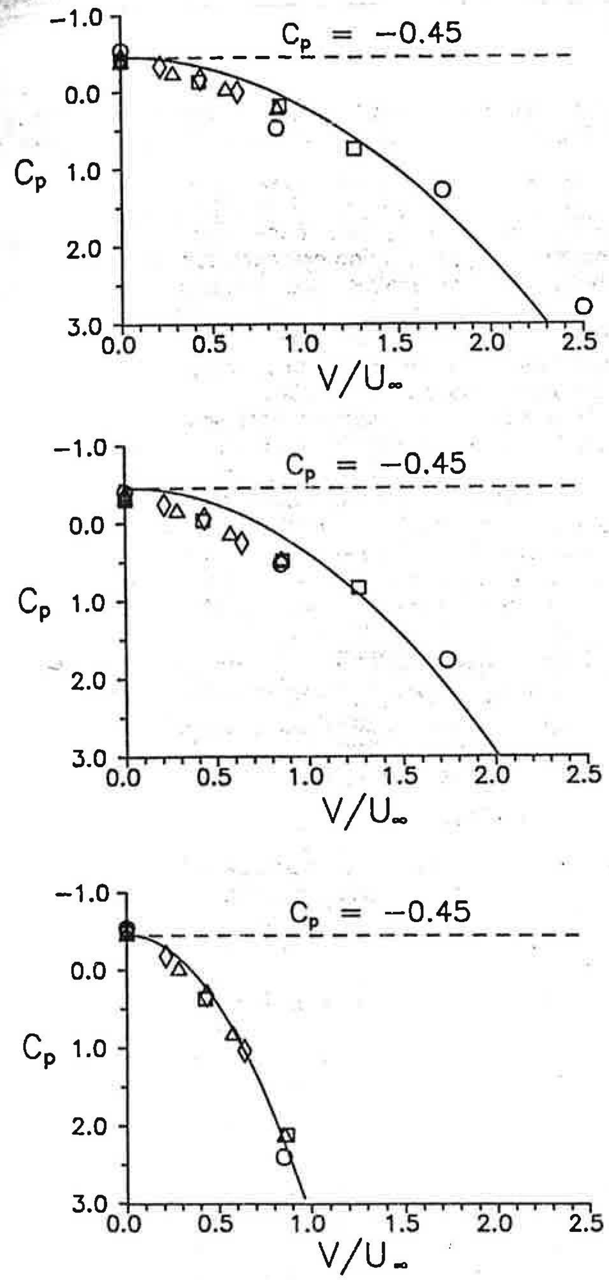
Modeling Ventilator Performance

The data in Figures 4 and 5 suggest that the ventilator draft is determined by two competing effects: (1) negative pressure caused by separation of the wind stream increases draft, and (2) a pressure loss as air exits the ventilator reduces the ventilator draft. This can be written by using Bernoulli's equation from elevation 2 in the vent pipe to the point marked P'_∞ (see Figure 2). After equating kinetic energy terms and neglecting the small potential energy term, we have

$$\Delta P_v = \Delta P_\infty - K\rho V|V|/2 \qquad (7)$$

where

ΔP_v = ventilator suction pressure,
$\Delta P_\infty = P_{\infty 2} - P'_\infty$ = separation suction pressure,
K = loss coefficient for air exiting the ventilator,
$K\rho V|V|/2$ = discharge pressure loss.



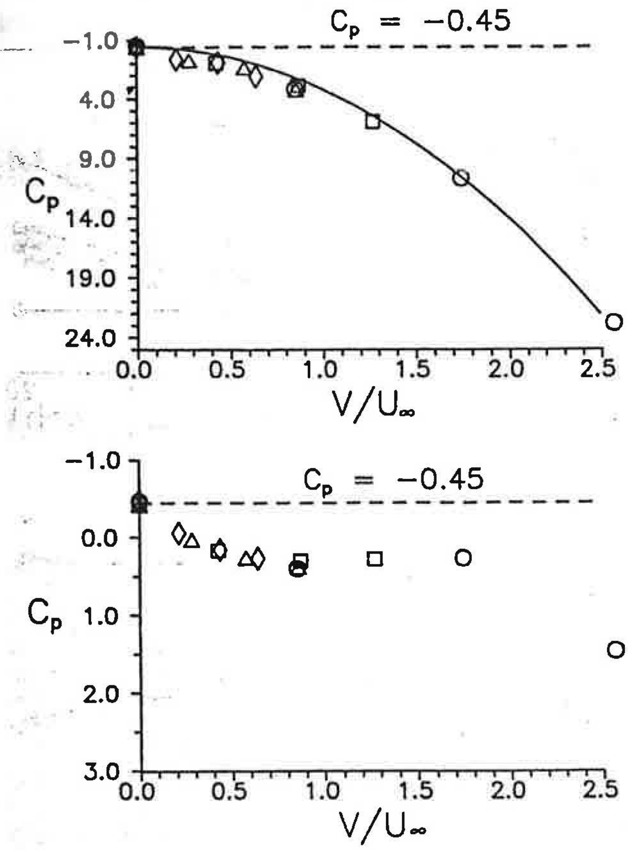


Figure 5 *Ventilator performance curves plotted using dimensionless variables. Reynolds number: 9.9 x 10^4; △ 7.4 x 10^4; □ 5 x 10^4; O 2.5 x 10^4. The solid line is Equation 8 with a best fit for the loss coefficient, K. Ventilators (a) wind directional; (b) caddicap; (c) vacuum cap, 90° wind angle; (d) vacuum cap, 0° wind angle; (e) open pipe.*

Nondimensionalizing Equation 7 and using the data plotted in Figure 5 shows that ΔP_v correlates as

$$C_p|_v = C_p|_{disk} + K(\rho/\rho_\infty)(V/U_\infty)^2 \qquad (8)$$

where

$C_p|_v$ = $-2\Delta P_v/\rho_\infty U_\infty^2$ = ventilator pressure coefficient,

$C_p|_{disk}$ = pressure coefficient behind a circular flat disk when flow is normal to the disk = -0.45.

Equation 8 suggests that ventilator draft can be characterized using two parameters: a pressure coefficient measured with a velocity ratio of zero (V/U_∞ = 0) and a loss coefficient for air exiting the ventilator.

Establishing the Ventilation Provided by a Ventilator

The volume flow of air through a vent pipe can be calculated from the building model (Equation 4) using experimental ΔP_v data. Since this calculation is implicit, the easiest solution is graphical. First, plot a building model curve (ΔP_v calculated from Equation 4). Next, plot ΔP_v data. Points of intersection give flow rates. This procedure is analogous to finding the operating point of a fan or pump. The building model provides the system curve, and the ventilator performance curve is analogous to the fan curve.

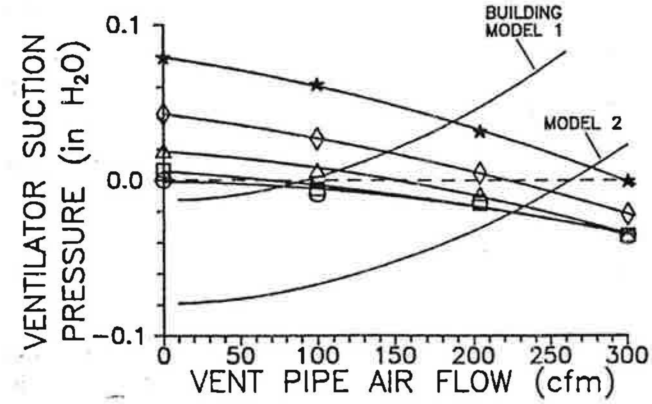


Figure 6 *Ventilator performance curves for a wind-directional ventilator and building model curves. Windspeed U_∞: ★ 20 mph; 15 mph; △ 10 mph; □ 5 mph; O 0 mph.*

Before a building model curve could be drawn, specific building characteristics had to be defined. Appendix A defines characteristics for two generic buildings. Ventilation in building model 1 is driven by a vent cap and by a stack effect due to a 50°F (28°C) temperature difference. Ventilation in building model 2 is driven by a vent cap and by a stack effect due to a 200°F (111°C) temperature difference.

Figure 6 shows a set of ventilator performance curves and

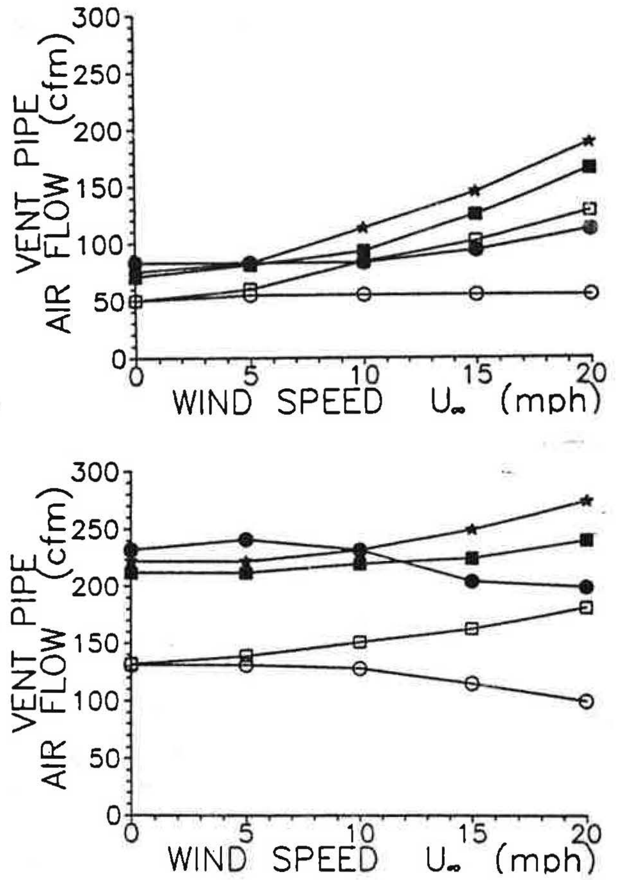


Figure 7 *Comparison of flow rates for a building ventilated by a 7-in. pipe capped with different ventilators: (a) building model 1, (b) building model 2. Ventilators are ★ wind directional; ■ caddicap; □ vacuum cap, 90° wind angle; O vacuum cap, 0° wind angle; ● open pipe.*

the building model curves. Intersections of the curves define ventilator airflow rates as a function of wind speed. For example, when the wind velocity is 20 mph (8.9 m/s), building 2 would operate with a ventilator air flow of about 275 cfm (130 L/s).

Curves of ventilation airflow rates were found for each ventilator tested. Figure 7 shows the results. The differences between Figures 7a and 7b show that ventilation flow rates depend strongly on the building models used. Thus, while these results are useful for comparing ventilator performances, they apply only to buildings identical to the models used.

CONCLUSIONS

Ventilator suction pressure data (ΔP_v) characterize gravity ventilator draft. ΔP_v is the static pressure in the wind stream minus the static pressure in the vent pipe just before the ventilator. It is critical to record these data as a function of both the wind speed and the speed of the air in the vent pipe.

Representative ΔP_v data from three different 7-in. ventilators and an open pipe were gathered. The data suggest that two competing effects establish the draft of a ventilator: (1) as wind moves over a ventilator, separation causes a low-pressure wake that promotes draft, and (2) as air exits a ventilator, the resulting pressure drop reduces draft.

Dimensional analysis showed that ΔP_v depends on three groups: Reynolds number, a ratio of vent pipe air velocity to wind speed, and a ratio of vent pipe air density to ambient air density. For a density ratio of one, data correlation showed that the ventilator suction pressure depended only on the velocity ratio. For ventilators without a coupling effect, ventilator suction pressure correlated with a simple equation (Equation 8). This suggests that ventilators should be rated by two parameters: (1) a pressure coefficient at a flow ratio (V/U_∞) of zero and (2) a loss coefficient (K).

How much does ventilation increase when a ventilator is used? A building model (Equation 4) shows that this can only be answered for a specific building. In particular, ΔP_v, a buoyancy term, and a head loss term match to define the flow rate of a ventilator. The easiest way to find this flow rate is graphically. The flow rate is defined as the intersection of a ΔP_v curve and a building model curve. This is the same method used to find the operating point of a fan or a pump.

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APPENDIX A
DETAILS OF THE BUILDING MODELS

Rewriting Equation 4 from the main text gives a building model curve:

$$\Delta P_v = \rho V^2/2 - gh\Delta\rho + 4C_f(L_e/D)\rho|V|V/2 + (Q/C)^{1/n}. \quad \text{(A1)}$$

To select representative values of C and n, we curve-fit Figure 12 of chapter 22 in *ASHRAE Fundamentals* (ASHRAE 1985). The results were C = 22,360 cfm/psi$^{1/2}$ (127 L/s·Pa$^{1/2}$) and n = 0.5. Other variables selected were,

Variable	Building Model 1	Building Model 2
$T_{outside}$	25°F (−4°C)	20°F (−7°C)
T_{inside}	70°F (21°C)	230°F (110°C)
E	0.0005 ft (152 μm)	0.0003 ft (91 μm)
L_e	16.4 ft (5 m)	16.4 ft (5 m)
h	9.8 ft (3 m)	16.4 ft (5 m)