AIR MOVEMENT & VENTILATION CONTROL WITHIN BUILDINGS

12th AIVC Conference, Ottawa, Canada
24-27 September, 1991

PAPER 18

INTERACTION OF MECHANICAL SYSTEMS AND NATURAL INFILTRATION

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SYNOPSIS

Mechanical devices such as exhaust fans and air handlers interact strongly with natural infiltration. In the past, the effects of mechanical systems have either been treated separately from those of natural infiltration or have been combined using simple models.

This paper extends the theory of the interaction of unbalanced mechanical systems with stack-driven infiltration. The effects of leakage distribution and the flow exponent on fan efficiency are discussed.

A simple model for combining the two effects is presented and compared with two previously proposed models. The induced infiltration is one-half of the unbalanced fan flow if the unbalanced flow is less than twice the natural infiltration rate. Otherwise, the induced flow is the difference between the fan flow and the natural infiltration.

The model is supported by detailed flow and pressure measurements made on a home in the Pacific Northwest during winter conditions. The home had a whole-house multiport exhaust ventilation system as well as several other exhaust fans. The measured flow through the multiport exhaust system was 35 L/s. When running, this system induced 17 L/s, in good agreement with the fan-model predictions.

LIST OF SYMBOLS

β	Dimensionless neutral level
β_0	Dimensionless neutral level under conditions of stack only
C	Coefficient from blower door test (L/(s Pa^n))
$\Delta\rho$	Density difference between indoor and outdoor air (kg/m^3)
ε	Fan efficiency (flow added divided by fan flow)
ε_0	Efficiency of an infinitesimally small fan
ε_{crit}	Fan efficiency at the critical point (β=1)
F_s	Stack factor from infiltration model
g	Acceleration due to gravity (9.80665 m/s^2)
h	Height of building (m)
n	Exponent from blower door test
P_s	Stack pressure $P_s = \mid \Delta\rho \mid g\,h$ (Pa)
Q_{add}	Additional flow induced by the fan (L/s)
Q_{crit}	Critical flow (β=1) (L/s)
Q_{fan}	Flow through fan (L/s)
Q_{in}	Flow in through envelope, excluding fan (infiltration) (L/s)
Q_{nat}	Flow under "natural" conditions (fan flow of zero) (L/s)
Q_{out}	Flow out through envelope, excluding fan (L/s)
R	Fraction of leakage in ceiling and floor
X	Leakage difference between ceiling and floor as fraction of total

1. INTRODUCTION AND BACKGROUND

Although it has been traditionally assumed that natural infiltration would provide adequate ventilation in homes, the mandates of energy efficiency combined with new construction materials and techniques have resulted in newer homes being significantly tighter. In many cases, the natural infiltration is no longer adequate. This has led to increasing use of various types of mechanical ventilation systems and growing interest in understanding their performance.

The interaction of mechanical ventilation systems with natural infiltration has been addressed in a number of recent studies. This paper extends the theory of the interaction of unbalanced mechanical systems with stack-driven natural infiltration. The nature of the dependence of the stack-fan interaction on leakage distribution and the power-law exponent are discussed. Based on this discussion, we propose a simple fan interaction model.

The new model is compared with two previously proposed models and is then applied to real-time field infiltration measurements. The chosen home was measured under late winter conditions with stack-dominated natural infiltration and a wide variation in exhaust fan flow rates.

2. THE INTERACTION EFFECT

The interactions of mechanical systems with natural infiltration are quite complex. These interactions include not only the incidental or planned use of exhaust fans or balanced-flow heat recovery ventilation systems, but also the effects which arise from forced air distribution systems used for space heating and cooling. The latter effects can be quite large (Palmiter and Bond 1991; Palmiter et al. 1991) and are largely unintended. In this paper, we will focus on systems which are intended to provide additional ventilation.

Balanced flow systems are those which have two fans, one pumping air into the home and a second pumping air out of the home. These systems are the easiest to understand. If the two flows are equal, there is theoretically no effect on the neutral pressure level and no interaction with natural infiltration. The resultant total infiltration is simply the sum of the natural infiltration and the inward flow through the balanced system.

The most common type of ventilation system installed in the Pacific Northwest is the exhaust or extract system. Because this system changes the neutral pressure level, the added flow is less than the flow through the fan. In extreme cases, a fan flow of 100 L/s might result in only 10 L/s of added ventilation. It should be noted, however, that the total infiltration is never less than the flow through the fan.

Wind-induced infiltration is poorly understood, as is the interaction of wind and stack effects. The manner in which mechanical systems interact with natural infiltration may depend significantly on the relative mix of wind and stack effects.

Various infiltration studies in the Pacific Northwest have shown that natural infiltration in homes in this region is primarily stack-dominated. Palmiter et. al. (1991) give a summary of these studies. A more detailed discussion of the stack effect and its predominance is given in Palmiter and Brown (1989). We will use the stack effect to illustrate the interaction of mechanical devices with natural infiltration.

After examining the interaction of exhaust systems with stack-driven infiltration, we propose a very simple model in which it is assumed that the total natural infiltration interacts with the exhaust fan in the same manner as the stack effect.

3. THEORY FOR FAN-STACK INTERACTION

For the case where unbalanced fan flow interacts with stack-induced infiltration, explicit equations can be given which, in special cases, can be solved in closed form. The primary benefit of analytic solutions is to provide understanding of how the various parameters influence the total flow.

For simplicity, it is convenient to make several assumptions. The relationship between volumetric flow and pressure difference is assumed to have the form of a power law, in which the same exponent applies for all leakage areas and the overall flow coefficient is derived from a blower door test. Field experience has shown that the exponent is typically 0.65 ± 0.05. It is assumed that the building is a simple rectangular box with leakage in the floor, ceiling and walls, and that the wall leakage density is constant with height.

We follow Sherman (1980) and Wilson (1990) in introducing the dimensionless leakage distribution parameters X and R. The parameter R ($0 \leq R \leq 1$) is the fraction of the total leakage which is in the floor and the ceiling. The parameter X ($-R \leq X \leq R$) is the fraction of leakage in the ceiling minus the fraction of leakage in the floor. With this terminology, the fraction of leakage in the ceiling is $(R+X)/2$, that in the floor is $(R-X)/2$, and that in the walls is $(1-R)$. It is generally assumed that $R=0.5$ and $X=0$ are reasonable default values.

We assume that the pressure due to stack effect is given by the hydrostatic equation, which leads to a linear relationship between stack pressure and height. The height at which the indoor and outdoor pressure lines cross is the neutral pressure height. It is convenient to use the concept of a dimensionless neutral level, β, which is the ratio of the neutral pressure height to the height of the building. Under stack-only conditions, the neutral level is between 0 and 1; the operation of an exhaust fan shifts the indoor pressure line, resulting in an increase in neutral level, perhaps much above the ceiling.

The flows above and below the neutral level can be found by integrating the power law over the linear pressure gradient. A flow balance then results in a nonlinear equation for the neutral level. As a further simplification, we use a volume flow balance. The results are generally rather close to those obtained from a mass flow balance, and the equations are easier to understand.

The interaction of stack effect with unbalanced fan flow depends on the direction of fan flow (i.e., supply or exhaust) and also on whether the indoor temperature is above or below outdoor temperature (i.e., winter or summer conditions). For brevity, we present only the case of an exhaust fan under winter conditions. A rule for translating the results to all other cases is given later in this section.

The fan-stack interaction results in the added flow due to the fan being less than the flow through the fan. It is therefore convenient to define a fan efficiency, ε, as the added flow induced by the fan divided by the fan flow. The interaction can then be expressed in dimensionless form, giving ε as a function of the ratio Q_{nat}/Q_{fan}. The fan efficiency is defined as

$$\varepsilon = \frac{Q_{in} - Q_{nat}}{Q_{fan}} \tag{1}$$

For a balanced flow system, as noted above, the flow added is the flow through the supply fan, but the total fan flow also includes the flow through the exhaust fan. If we consider the efficiency definition to mean added flow divided by total air pumped, the efficiency of the balanced system is always equal to 0.5.

The volumetric flow due to stack effect can be written

$$Q_{nat} = CF_s P_s^n \quad \text{where} \quad 1/2 \le n \le 1 \tag{2}$$

where the stack factor, F_s, accounts for the effects of leakage distribution.

If there is sufficient flow through the fan, the neutral level can be raised to the ceiling (β=1). After this point, the flow through the ceiling reverses. All flow comes in through the walls, floor, and ceiling, and all flow exits through the fan; therefore, the total flow is known. We refer to this point as the "critical" point and the fan flow at this point as the "critical" flow.

When the neutral level is below or at the ceiling ($\beta \le 1$), flow in and out are given by

$$Q_{in} = CP_s^n \left(\frac{1-R}{n+1} \beta^{n+1} + \frac{R-X}{2} \beta^n \right) \tag{3}$$

$$Q_{out} = CP_s^n \left(\frac{1-R}{n+1} (1-\beta)^{n+1} + \frac{R+X}{2} (1-\beta)^n \right) \tag{4}$$

For an exhaust fan, a flow balance yields the basic equation

$$Q_{fan} = Q_{in} - Q_{out} \tag{5}$$

By setting β=1 in (3), we obtain the critical flow

$$Q_{crit} = Q_{in} = CP_s^n \left(\frac{1-R}{n+1} + \frac{R-X}{2} \right) \tag{6}$$

The efficiency at the critical point is

$$\varepsilon_{crit} = 1 - \frac{Q_{nat}}{Q_{crit}} = 1 - \frac{F_s}{(1-R)/(n+1)+(R-X)/2} \tag{7}$$

For $Q_{fan} > Q_{crit}$, the total flow in is equal to the flow through the fan, and the efficiency can be expressed as

$$\varepsilon = 1 - \frac{Q_{nat}}{Q_{fan}} \tag{8}$$

The dimensionless neutral level for a fan flow smaller than Q_{crit} can be obtained by solving (5) for β. The flows Q_{in} and Q_{out} can then be calculated from (3) and (4). In particular, the stack neutral level, β_0, is found by solving (5) with Q_{fan}=0 (that is, setting Q_{in} equal to Q_{out}). The resulting flow Q_{nat} is obtained by substituting β_0 into (3) so that, in this case, the term in brackets is the stack factor F_s.

Because (5) is nonlinear in β, its solution generally requires an iterative technique. However, in the context of doing simulations or modeling, it should be noted that β_0 is a constant and the nonlinear equation for β_0, which yields the stack factor, need be solved only once.

Closed form solutions can be found for the above equations in certain special cases. For a flow exponent of n=1 or n=0.5, there is a full solution for all values of fan flow. In the case of an exponent of n=0.5, the solution for β_0 involves the root of a cubic equation; it is quite long and complex and has therefore been omitted. Solutions for the stack-only and critical-point values for general exponents are given below for certain special cases.

Sherman (1990) has derived a partial solution for the efficiency, ε_0, of an infinitesimally small fan. By a limiting argument, it is possible to derive an explicit expression for ε_0 as a function of β_0. This extends Sherman's result to all values of n, R, and X. The equation is

$$\varepsilon_0 = \frac{\frac{1-R}{n}\beta_0^n + \frac{R-X}{2}\beta_0^{n-1}}{\frac{1-R}{n}\beta_0^n + \frac{R-X}{2}\beta_0^{n-1} + \frac{1-R}{n}(1-\beta_0)^n + \frac{R+X}{2}(1-\beta_0)^{n-1}} \tag{9}$$

This is a good approximation to the actual efficiency when $Q_{nat} > 4\,Q_{fan}$. When X=R=1, we define ε_0=1.

We now summarize the solutions for stack-only and critical-point flows and fan efficiencies. The efficiencies are expressed both in terms of n, R, and X and in terms of β_0.

3.1 The case n=1

If the flow exponent n is equal to 1, the equations for natural and critical flow reduce to

$$\beta_0 = (1+X)/2 \qquad \varepsilon_0 = \frac{1-RX}{2} = \frac{1+R}{2} - R\beta_0 \tag{10),(11}$$

$$Q_{nat} = CP_s\frac{(1+R)(1-X^2)}{8} \qquad Q_{crit} = CP_s\frac{1-X}{2} \tag{12),(13}$$

$$\frac{Q_{nat}}{Q_{crit}} = \frac{(1+R)(1+X)}{4} = \frac{1+R}{2}\beta_0 \qquad \varepsilon_{crit} = 1 - \frac{(1+R)(1+X)}{4} = 1 - \frac{1+R}{2}\beta_0 \tag{14),(15}$$

3.2 The case R=1

If all of the leakage is in the floor and the ceiling, the equations reduce to

$$\beta_0 = \frac{(1+X)^{1/n}}{(1+X)^{1/n}+(1-X)^{1/n}} \qquad \varepsilon_0 = \frac{(1-X)^{1/n}}{(1-X)^{1/n}+(1+X)^{1/n}} = 1-\beta_0 \tag{16),(17}$$

$$Q_{nat} = CP_s\frac{1-X}{2}\beta_0^n \qquad Q_{crit} = CP_s\frac{1-X}{2} \tag{18),(19}$$

$$\frac{Q_{nat}}{Q_{crit}} = \beta_0^n \qquad \varepsilon_{crit} = 1-\beta_0^n \tag{20),(21}$$

3.3 The case X=0

If the leakage in the floor and ceiling are equal, the equations are

$$\beta_0 = 1/2 \qquad \varepsilon_0 = 1/2 \tag{22),(23}$$

$$Q_{nat} = CP_s\left(\frac{1}{2}\right)^{n+1}\frac{1+nR}{n+1} \qquad Q_{crit} = CP_s\left(\frac{2-R+nR}{2(n+1)}\right) \tag{24),(25}$$

$$\frac{Q_{nat}}{Q_{crit}} = \left(\frac{1}{2}\right)^n\frac{1+nR}{2+R(n-1)} \qquad \varepsilon_{crit} = 1-\left(\frac{1}{2}\right)^n\frac{1+nR}{2+R(n-1)} \tag{26),(27}$$

3.4 Extension to summer conditions and supply fans

As mentioned above, the fan efficiency depends on the direction of fan flow (i.e., supply or exhaust) as well as on whether the indoor temperature is above or below outdoor temperature (i.e., winter or summer conditions).

Equations for either an exhaust fan under summer conditions or a supply fan under winter conditions may be obtained by replacing the parameter X in the equations with $-X$. For a supply fan under summer conditions, there are two sign changes in X, so the resulting equations are identical to those given above for an exhaust fan under winter conditions.

3.5 Simple fan interaction models

Chapter 23 of the *ASHRAE Handbook of Fundamentals* (1989) proposes use of addition in quadrature for predicting the effect of unbalanced flow systems. In this method, the total flow is given by

$$Q_{tot} = \sqrt{Q_{nat}^2 + Q_{fan}^2} \tag{28}$$

Wilson and Walker (1987) proposed another model in which half the fan flow is combined using an exponent rule and the other half is added linearly.

$$Q_{tot} = Q_{fan}/2 + ((Q_{fan}/2)^{1/n} + Q_{nat}^{1/n})^n \tag{29}$$

The authors have proposed a very simple model (Palmiter and Bond 1991) which we will refer to as the 0.5 rule. However, no justification or derivation of the model was given. In this model, the added flow is one-half the fan flow up to a critical point of twice the natural infiltration rate.

$$Q_{tot} = 0.5\,Q_{fan} + Q_{nat} \quad \text{for} \quad Q_{fan} \le 2Q_{nat} \tag{30}$$

$$Q_{tot} = Q_{fan} \quad \text{for} \quad Q_{fan} \ge 2Q_{nat} \tag{31}$$

This is equivalent to asserting that the fan efficiency ε is 0.5 up to the critical point, at which $Q_{fan}= 2\,Q_{nat}$. Above that point, the efficiency is given by (8). This model was also extended to include the effects of balanced and unbalanced duct leakage in a forced air distribution system.

4. DISCUSSION

Many of the general features of the fan-stack interaction are easily predicted using the equations given above. In order to explore the interaction in more detail, we solved the nonlinear equations for a number of cases. Some typical results are illustrated in Figure 1. Each panel shows fan efficiency plotted versus the ratio of natural infiltration to flow through the exhaust fan. In all cases, we used the typical exponent of 0.65.

In each panel, the straight line sloping down from the top left corner has a slope of unity and represents the cases where all flow out is through the fan. The curved portions intersect with the straight line at the critical point. Toward the right side, the curves asymptotically approach ε_0. It is clear that convergence is nearly complete at a natural infiltration to fan flow ratio of about 4-to-1.

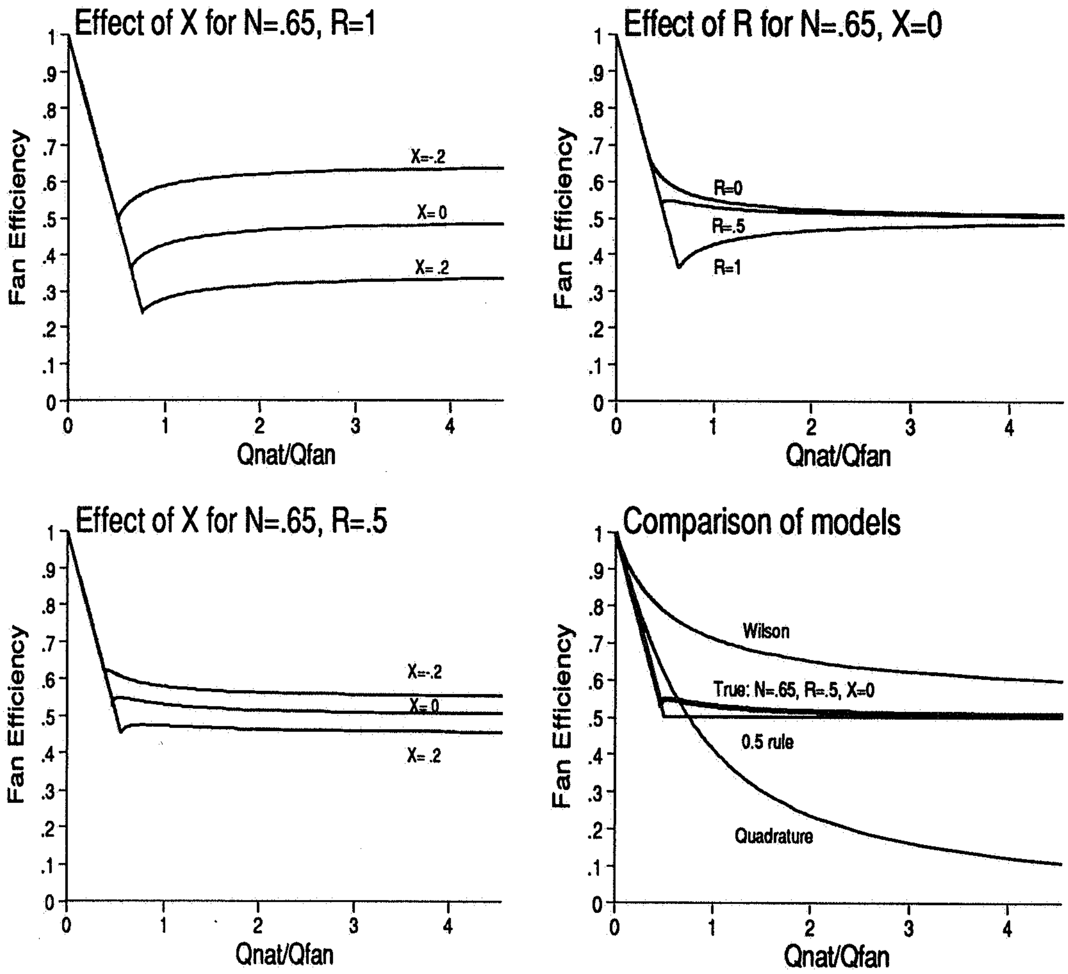

Figure 1. Fan efficiencies. Each panel shows fan efficiency versus the ratio of natural flow to fan flow for n=0.65. The left panels show the effect of varying X; R=1 for the upper left panel and R=0.5 for the lower left panel. The upper right shows the effect of varying R with X=0. The lower right panel compares three simple fan models; the 0.5 rule is the model proposed in this paper.

The efficiency is most strongly influenced by the parameter X when R=1 (i.e., all leakage is in the floor and ceiling). This is shown in the upper left panel. The upper and lower curves are for X of −0.2 and 0.2, respectively. These represent a 60-40 distribution of leakage between the floor and the ceiling. Note that efficiency is lowest at the critical point. These cusps are due to the combination of a power law flow model with a concentration of leakage at a single height. It is clear that more extreme values of X will result in considerable enhancement or reduction of fan efficiency, indicating the undesirability of placing inlet vents high on the walls.

The effect of X when R has the more typical value of 0.5 (i.e., half the leakage is in the walls) is shown in the lower left panel. Notice that the effect of varying X is greatly reduced. The curves are all reasonably close together.

The upper right panel shows the effect of varying R when X=0. Notice that there is relatively little difference between R=0.5 and R=0. A decrease in R tends to increase the efficiency near the critical point; for R=0.5 the cusp effect due to the exponent is almost exactly balanced.

The lower right panel compares the simple models for fan-stack interaction given in the theory section. For comparison, we also show the standard case of X=0, R=0.5 and n=0.65. The quadrature rule tends to underpredict the added fan flow except near the critical point, where there is significant overprediction. The Wilson model overpredicts in all cases. The 0.5 rule is very close to the true value. A comparison with the other panels shows that it gives reasonable predictions for fan efficiency and total flow whenever X does not assume extreme values.

Although it would be easy to model the fan-stack interaction more accurately for more extreme values of X and R, the model would be sensitive to supply versus exhaust flow and summer versus winter conditions. Since the model has to be used for the wind component as well, the improved accuracy would be largely illusory until more is known about the wind component. For the present, we propose to simply apply the 0.5 rule and its extensions to the total natural infiltration.

5. COMPARISON WITH FIELD DATA

In the winter of 1990, detailed real-time measurements were made on a recently constructed, two-story home in Snohomish County, Washington which had been built under a local energy-efficiency program (Palmiter and Bond 1991). Based on a survey of 49 homes located in Snohomish County (Palmiter et. al. 1990), this home was typical in its size, ventilation system, tightness, and measured infiltration rate. The home was all-electric; the heating system consisted of electric resistance wall units with fans. Characteristics of the home are listed in Table 1.

Table 1. Home and environmental characteristics of test home.

Floor area	144 m^2
Volume	350 m^3
Full height	4.95 m
Average stack height	4.18 m
Blower door leakage function	Q (L/s) = 60.1 $\Delta P^{0.629}$ (ΔP in Pa)
Effective leakage area @ 4 Pa	560 cm^2
ACH at 50 Pa	7.24
Specific leakage area (SLA)	3.88
Average ΔT	11.6 C
Average wind speed	0.53 m/s
Average infiltration	37 L/s (0.383 ACH)
Average natural infiltration	28 L/s (0.289 ACH)
Induced by multiport exhaust	17 L/s (0.177 ACH)

The data taken included tracer gas measurements made with a multitracer measurement system (Sherman and Dickerhoff 1989), pressure measurements across each face and the ceiling and floor of the home, and indoor and outdoor temperature measurement. One-time measurements of fan flow were also made using tracer gases and a flow hood.

The whole-house ventilation system consisted of a multiport exhaust system, which drew air from five ports on the second floor, and a single through-wall inlet port with a pressure-regulated damper on the first floor. During the tracer test, the exhaust system was operated by a timer, set to run four hours on and eight hours off.

In addition to the multiport exhaust system, the home had six exhaust fans. The fans and their measured flows, including the whole-house ventilation system, are listed in Table 2. Some of the measured flows are quite low due to poor installation of the fans. The occupants logged their use of the exhaust fans.

The home was well sheltered, so wind-induced infiltration was small and the stack-only infiltration model should be a reasonable approximation to the natural infiltration. We predicted the stack infiltration using the LBL infiltration model (Sherman 1980) and the AIM-2 infiltration model (Walker and Wilson 1990) with the indoor and outdoor temperatures measured at the site. During periods of no fan operation and low wind speed, the LBL model was about 10% too high and the AIM-2 model about 10% too low.

We adjusted the results of the AIM-2 model by 10% so that it generally agreed with the measured natural infiltration. The total fan flow at any given time was the sum of the measured flows through all fans operating. The adjusted natural infiltration and total fan flow were used with the fan combination rule to predict a total flow.

Pressure differences across the floor and ceiling, measured as indoor minus outdoor pressure, are shown in the top graph of Figure 2. The periods of exhaust fan operation, resulting in depressurization of the home and an increase in neutral level, are clearly visible. The pressure across the ceiling is normally positive due to stack effect. When it drops below zero, the flow across the ceiling has been reversed. The pressure datalogger failed on the evening of day 79 and there are no pressure data after that time.

The stack pressure is obtained by subtracting the measured pressures across the floor and ceiling. The effects of fans do not appear in the result. Assuming that the pressure gradient is linear with height, the dimensionless neutral level β is obtained by dividing the pressure across the floor by the measured stack pressure. This neutral level is shown in the second graph of Figure 2. The periods where $\beta>1$ correspond to the periods where the ceiling pressure has been reversed.

The measured and predicted infiltration are shown in the lower two graphs of Figure 2; the top graph shows stack-only predictions and the lower graph shows fan model predictions. The lower graph also includes the signal of the multiport ventilation system.

On the whole, the predictions of the fan model are fairly good. The two spikes in the measured data on day 77 are due to door openings, which show clearly in the temperature and pressure data. We attribute the other discrepancies to errors in the occupant log and uncertainties about flows through some of the fans. All in all, the 0.5 rule gives very reasonable predictions. We note that additional comparisons, not given here, yielded comparable results (Palmiter and Bond 1991).

Table 2. Exhaust fans at test home.

Fan Location	Measured Flow (L/s)	Fan Location	Measured Flow (L/s)
Multiport exhaust	35	Master bathroom	6
Dryer	31	First floor bathroom	14
Laundry	15	Second floor bathroom	29
Kitchen (range hood)	47		

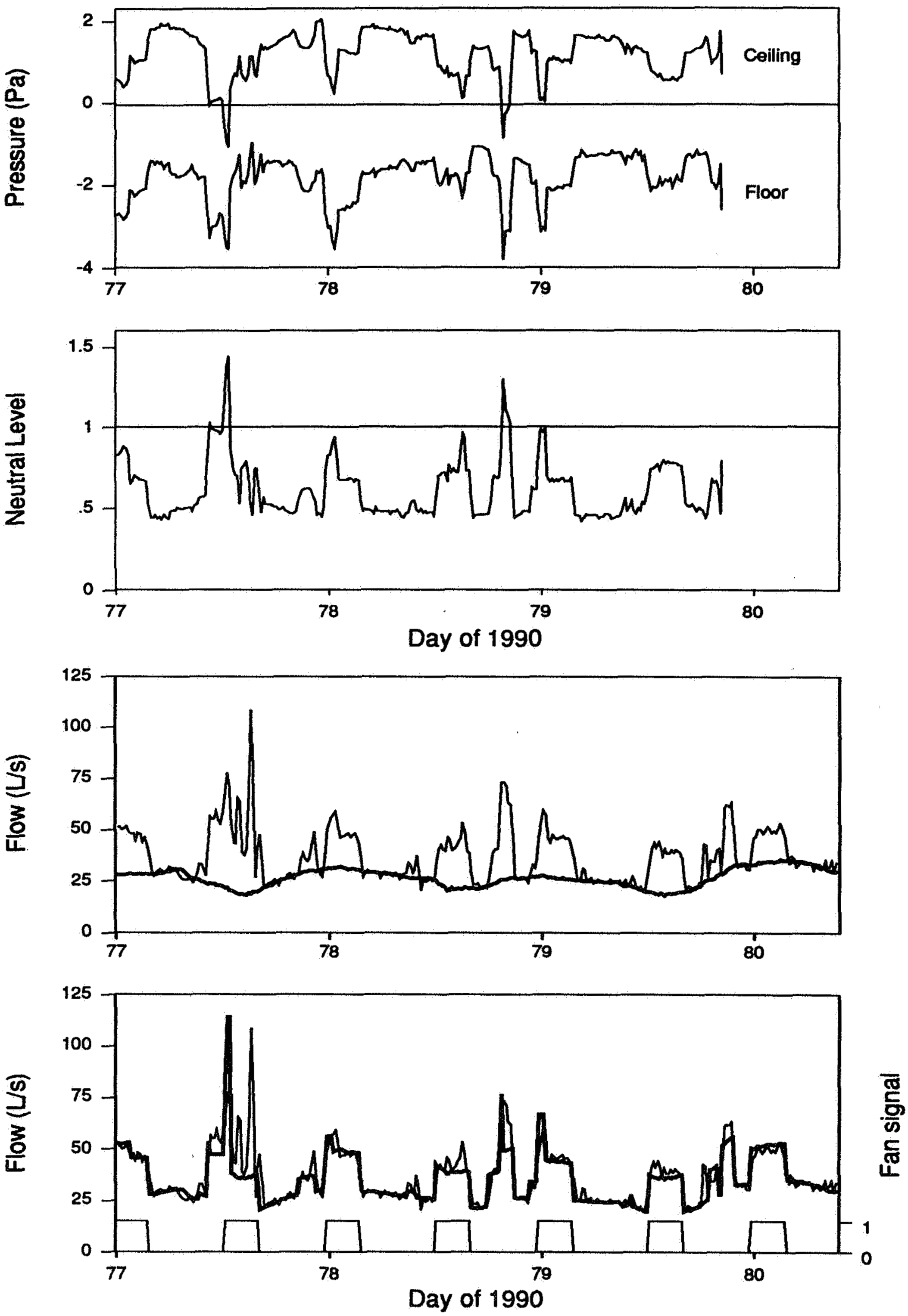

Figure 2. Pressures and infiltration at test home. The top graph shows measured pressures across the ceiling and floor; the second graph shows the dimensionless neutral level. The third graph compares the adjusted AIM-2 stack prediction (bold) with measured infiltration and the fourth shows the fan model predictions (bold), again compared with measured infiltration.

6. CONCLUSIONS

The theoretical development shows that the 0.5 rule for combining unbalanced mechanical flow with natural infiltration works well for stack-dominated infiltration, except in the case where R=1 (all leakage in the floor and ceiling). In particular, it is very good for the typical case of n=0.65, R=0.5, X=0. It is a considerable improvement over the quadrature and Wilson models, both of which have the wrong functional form.

A comparison with detailed field data shows the model to be reasonably good. Although the fan-stack interaction model could be easily improved, the necessity of using it with wind effect and combined wind and stack effects tends to negate the apparent increase in accuracy. Further investigation of the effect of wind is required.

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