Title
EFFECTIVENESS OF LOCAL VENTILATION IN REMOVING SIMULATED POLLUTION FROM POINT SOURCES



Author
Revzan, K.L.

Publication Date
1984-06-01

# Lawrence Berkeley Laboratory

UNIVERSITY OF CALIFORNIA



## APPLIED SCIENCE DIVISION

Submitted to Atmospheric Environment

EFFECTIVENESS OF LOCAL VENTILATION IN REMOVING SIMULATED POLLUTION FROM POINT SOURCES

K.L. Revzan

June 1984



APPLIED SCIENCE
DIVISION

Prepared for the U.S. Department of Energy under Contract DE-AC03-76SF00098

Submitted to
*Atmospheric Environment*

LBL-16701
EEB-Vent 84-18

# EFFECTIVENESS OF LOCAL VENTILATION IN REMOVING SIMULATED POLLUTION FROM POINT SOURCES

K. L. Revzan

Building Ventilation and Indoor Air Quality Program
Lawrence Berkeley Laboratory
University of California
Berkeley, CA 94720

June, 1984

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Buildings Energy Research and Development, Buildings Systems Division of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098, and by the U.S. Environmental Protection Agency, Office of Research and Development.

Although the research described in this article has been funded wholly or in part by the United States Environmental Protection Agency through Interagency Agreement AD-89-F-2A-062 to the U.S. Department of Energy, it has not been subjected to EPA review and therefore does not necessarily reflect the views of EPA and no official endorsement should be inferred.

*Abstract*

The effectiveness of range hoods and window fans in removing indoor pollutants is considered. Tests were conducted in a two-room test space designed to represent modern residential building practices. Pollutants were simulated using sulfur hexafluoride as a tracer gas. Range hood tests were carried out with heated and unheated tracer gas. In the former case, ventilation efficiency was roughly linear over a range of flow rates from 10.3 to 60.0 l/sec; the highest measured efficiency was 0.77. With unheated tracer gas, effectiveness was highly dependent on ambient environmental conditions. Window fan tests were conducted with the source of tracer gas in each of the two rooms, the fan itself remaining fixed. With the source in the room without the fan, fairly good agreement with a mass-balance model was obtained, with mixing factors ranging from ~1.0 to ~17.5, depending on fan flow rate and on ambient conditions. With the source in the same room as the fan, agreement with the model was poor. In neither case did the average concentration in the room without the source differ from that in the room with the source by more than 50%. A useful indicator of local efficiency was the ratio between the steady-state concentrations in the outlet duct and in the fan room, which reached ~4 at a flow rate of 45.2 l/sec, with the source and fan in the same room.

## INTRODUCTION

A number of common indoor pollutants derive from sources which may be considered, for the purpose of analysis, to be points; among these are the products of combustion arising from gas cooking or smoking, e.g., oxides of nitrogen, carbon monoxide, carbon dioxide, and particles. A common and inexpensive method for removing these pollutants is local ventilation, i.e., exhausting air from a point sufficiently close to the source to diminish transport of the pollutant to the larger part of the space.

Where cooking is the source, the usual form of local ventilation is the range hood, which consists of a fan and ductwork that expands at the inlet to the area of the cooking surface, and which is placed at the smallest convenient distance above that surface, usually ~1 m. It has been observed (Traynor, 1982), however, that measurable amounts of pollutants generated by cooking may appear in rooms distant from the kitchen even when a range hood is in use throughout the cooking period. The probable transport mechanism is natural convection arising from the buoyancy of the heated combustion products and from temperature differences among the interior walls. While hood configuration is important (ASHRAE, 1980), the determination of the dependence of pollutant concentrations on the fan flow rate for a given configuration is a useful first step toward a more complete understanding of the process of pollutant removal.

To determine the relationship between fan flow rate and pollutant levels, experiments were conducted in a two-room test space using a tracer gas introduced to simulate cooking, with the fan flow rate varying between 10.3 and 60.0 l $s^{-1}$. The influence of buoyancy on transport was studied by conducting tests with the tracer gas both heated and unheated. Measurements of tracer gas concentration were made at seven points within the test space and one in the fan outlet duct, providing an indication of the spatial distribution of the pollutant.

To study a more general case of local ventilation, tests were also conducted with a fan exhausting at rates varying from 10.3 to 45.2 l $s^{-1}$ through a window of one of the test rooms. Tracer gas was released at the approximate center of each of the two test rooms, and the concentration measured, as before, at seven points in the rooms and in the outlet duct. The results of the tests were compared with a simple two-room mass balance model (Sandberg, 1981) in an effort to determine the extent of mixing between the rooms during the course of the experiment. The magnitude of the mixing factor and the relationship of the tracer gas concentrations in the test space and in the exhaust provide indices of the effectiveness of the fan in local pollutant removal.

Because our tests were conducted at relatively low flow rates, it is not suggested that they reflect strictly realistic configurations, particularly in the case of window fans. The results are intended to be taken as an indication of the kinds of mixing patterns that result from the combination of forced and free convection and of the changes in local ventilation efficiency that may be expected as flow rates and ambient conditions change.

## APPARATUS

The test space comprised two rooms of a three-room experimental area (Figure 1), designed to reflect modern building practice with respect to air leakage and thermal characteristics. The natural infiltration rate of the entire space has been determined, using tracer gas decay, to be ~0.05 $hr^{-1}$. The volumes of rooms 1 and 2 were, respectively, 36,600 and 32,100 l; the ceiling height was 2.4 m. Two distinct configurations of sampling points are shown in Figure 1: those labelled A1 through A8 were used during some preliminary testing and in one range hood test, while those labelled B1 through B8 were used during the remainder of the testing; the sampling points within the test space were located midway between floor and ceiling. The positions of the small instrument-cooling fans used for creating well-mixed air are also shown. A reciprocating fan was placed on the floor in the doorway between rooms 1 and 2 for additional mixing.

The range hood (Kenmore 76000) measured 76 cm wide by 30 cm deep, which was approximately the size of the cooking surface (Kenmore 33491). The hood opening was 60 cm above the surface. Tracer gas was introduced, in the manner described below, at a point 10 cm above the right front burner of the cooking surface. An aluminum plate was placed on the burner to reduce the temperature of the tracer gas to ~80° C for those tests during which the burner was on.

The outlet duct of the window fan was placed 60 cm above the floor of room 2 in the position shown in Figure 1. The ductwork, of 10 cm internal diameter, is shown extending from the range hood; appropriate changes, not shown, were made to allow window fan operation. The pressure drop across an orifice plate, measured by a water manometer with an electronic sensor, was used to determine the flow rate. This technique has been found (Fisk, 1984) to be accurate to within ±5% or better, when compared with the results of a traverse of the duct with a Pitot tube. The blower, which was used as the window fan, was also employed as an auxiliary range hood fan, since the resistance of the orifice plates precluded the attainment of adequate flow rates with the hood fan alone. The flow rate through the duct was controlled by a mechanical damper.

Measurement of the tracer gas concentration in the exhaust (point B4) was made by sampling just downstream of the orifice plate, in order to take advantage of the turbulence characteristically found at this point. To determine the accuracy of this measurement technique, a test was conducted in which pure $SF_6$ was injected into the test space with the air well mixed and the range hood in operation. A difference of <5% was found among the measured concentrations at the several points within the space and at the point in the duct.

To provide a tracer gas, pure $SF_6$ was introduced at a controlled rate (Brooks 5811 mass flow controller) of 26.6 cc/min to the inlet of a peristaltic pump, which diluted the gas by an approximate factor of 20 with outside air. The molecular weight of the resulting combination was ~ 25% greater than that of air. The placement of the injection line for range hood experiments has been described above; for the window fan experiments, the line was placed at mid-height in the center of the appropriate room. In all experiments, a fritted glass diffuser at the end of the injection line was used to avoid a highly directional outflow.

Measurement of $SF_6$ concentration was made with two infrared analyzers (Wilks Miran 101), operating between 0 and 25 ppm. The first analyzer sampled points 1-4 and the second points 5-8. The concentration is obtained from the expression

$$C = -A_1 \log (A_2 - A_3 V), \tag{1}$$

where V is the measured voltage and the $A_n$ are calibration coefficients. Multipoint calibrations of both analyzers have been carried out, with the result that (1) is accurate to within ±5% for a range of 5 to 25 ppm, provided that recalibration is carried out sufficiently often to compensate for analyzer drift. Below 5 ppm, equation (1) is accurate to ~0.2 ppm.

Instrument control and data acquisition were achieved through the use of a microprocessor (Intel 8020) and a BASIC program (Nazaroff, 1981). The program allows the user to enter a sequence of events in a queue, each to be performed at a specified time in a specified manner. The system then executes each event in the queue in turn as its time occurs. Certain of these events cause the system to itself enter an event in the queue, to allow automatic performance of activities requiring regular service. In the experiments described herein, the system controlled the range hood or window fan, the room mixing fans, the cooking burner, and the injection of tracer gas, in each case turning the device on or off at the appropriate time, collected environmental data, sequentially sampled $SF_6$ concentration at 8 points, and periodically recalibrated the $SF_6$ analyzers. Data were collected on magnetic tape and subsequently transferred to a larger system for analysis.

## EXPERIMENTAL PROTOCOL

### *Range Hood Tests*

Range hood tests were carried out at flow rates of 10.3, 20.8, 32.2, 44.4, and 60.0 l $s^{-1}$, corresponding to air exchange rates of 0.54, 1.09, 1.68, 2.32, and 3.14 $hr^{-1}$, respectively. At each flow rate, tests were made with and without the burner in operation. $SF_6$ measurements were taken at the points labelled "B" in Figure 1 at all flow rates except 60.0 l $s^{-1}$, for which the points labelled "A" were used.

Environmental data (the temperature in rooms 1, 2, and 3, and outdoors, and the wind speed and direction) were collected at 30 min intervals. Sampling of $SF_6$ concentration was performed sequentially at approximately two minute intervals; a complete sequence was therefore performed every 8 minutes. The $SF_6$ analyzers were recalibrated at 2 hour intervals, using secondary standards of approximately 0, 10, and 25 ppm $SF_6$, themselves calibrated against primary standards.

The tests were conducted as follows: At the beginning of each test the range hood was turned on, the flow rate of the fan measured, the injection of $SF_6$ begun, and the cooking burner turned on if required. After one hour, the hood was turned off and the mixing fans turned on for one hour to provide a measure of the average concentration in the entire space. Subsequently, the mixing fans were turned off and the $SF_6$ concentration allowed to diminish to a level (<1 ppm) suitable for beginning the next test. The one hour mixing period proved sufficient to reduce differences among sampling points to ~5%. The concentration measured at the end of the mixing period was adjusted to the beginning of the period through the use of the natural infiltration rate of the test space, which was determined by analysis of the data obtained during the same period; the infiltration rate was sufficiently small (~0.05 $hr^{-1}$) that the error resulting from the calculation is insignificant.

*Window Fan Tests*

With the tracer gas source in room 1, window fan tests were made at flow rates of 10.3 and 20.3 l $s^{-1}$, corresponding to air exchange rates of 0.54 and 1.06 $hr^{-1}$, respectively. Sampling was done at points "B" of Figure 1. Three tests were made at each flow rate. With the source in room 2, window fan tests were made at flow rates of 10.3, 19.4, 31.7, 39.2, and 45.2 l $s^{-1}$, corresponding to air exchange rates of 0.54, 1.01, 1.63, 2.03, and 2.36 $hr^{-1}$, respectively. Sampling was again at points "B" of Figure 1. Two tests were made at each of the four lower flow rates and one at 45.2 l $s^{-1}$.

Each window fan test consisted of a three-hour injection followed by a decay to <1 ppm $SF_6$ concentration throughout the test space. Measurements of flow rate, concentration, and environmental parameters were carried out in the same manner as for the range hood tests.

## RANGE HOOD EXPERIMENTS

### *Theory*

If perfect mixing prevailed throughout the test space, the range hood would have no local effectiveness and mass balance would require that the concentration be

$$C_0(t) = \frac{f}{\lambda V}(1 - \exp(-\lambda t)) \tag{2}$$

where $C_0$ is the concentration, f the injection rate, $\lambda$ the air exchange rate, V the volume, and t the elapsed time from start of injection. If the actual measured concentration is C, we may define a ventilation efficiency, $\eta$, by

$$\eta = \frac{C_0 - C}{C_0} \tag{3}$$

This measure of efficiency is independent of time only if steady-state has been reached. In other circumstances, it is valid only if the measured concentration follows an exponential curve similar to (2). We will demonstrate that (3) is a satisfactory, if inexact, definition of efficiency under the conditions prevailing in our experiments.

The measure of efficiency provided by (3) affords no indication of the variation in concentration from point to point within the test space. The simple models that are available assume that perfect mixing obtains in each room, and are therfore inapplicable to circumstances in which the tracer gas concentration near the outlet is greater than that found elsewhere. No single number can represent the degree of mixing within a room or between rooms; examination of the data in each case is necessary.

*Results and Discussion*

We consider first the tests conducted with the burner on during the tracer gas injection period. It was found during preliminary work (see Appendix A) that results were highly repeatable, so that only a single test at each flow rate was used for analysis. The results, shown in Table 1 and Figure 2, demonstrate that ventilation efficiency, here defined by (3), increases roughly linearly with flow rate (Figure 2). If a straight line is fit to the data, using the simple least squares method, we find the best fit to be

$$\eta_c = 0.0611 + 0.0118\, F, \tag{4}$$

where $\eta_c$ is the calculated efficiency and F the flow rate in $l\ s^{-1}$; the correlation coefficient is 0.991. By extrapolation, we find that an efficiency of .90 would be reached at a flow rate of 71.1 $l\ s^{-1}$, although it is possible that some levelling off would occur before reaching this point.

Figure 3 is illustrative of the results obtained. Three runs at a flow rate of 32.2 $l\ s^{-1}$ are shown: the first two, carried out with the burner off, are discussed below; the third run, with the burner on, shows the exponential buildup of the tracer gas during the injection period. Calculations made on the basis of this and the other experiments show that the rate of the buildup is sufficiently close to the air exchange rate to permit neglecting the time-dependence of equation 3.

Two additional indicators of local ventilation efficiency are the ratios between the post-injection concentration at the center of room 1 and those at the center of room 2 and in the outlet duct, respectively. We see (Table 2) that the concentration in the outlet duct relative to that in the rooms increases with flow rate, which is indicative of the increasing local efficiency. The concentration in room 2 is comparable to that in room 1 until we reach a flow rate of 60.0 l $s^{-1}$; below this rate, the hood does not prevent the transport of pollutants from the source room to the remainder of the space, i.e., any pollutant that is not removed by the hood is likely to be distributed widely over any adjacent open space.

When the burner was not employed, results were highly variable. The first two runs shown in Figure 3 are representative: in the first run, we see an extremely high local efficiency, the post-injection concentration in the test space being <1 ppm; in the second, the concentration is actually higher than for the third run, during which the burner was on. It is inferred that free convection plays a much more important role in transport when the tracer gas lacks buoyancy. The results are discussed further in Appendix A.

## WINDOW FAN TESTS

### *Theory*

The tests conducted under conditions simulating a window fan offer an opportunity for comparison of experimental results with a simple two-room mass balance model (Sandberg, 1981). We assume that air is perfectly mixed within each of the rooms, of volumes $V_1$ and $V_2$, respectively, and that air is removed from room 2 at a flow rate F (Figure 4). The infiltration rates of the rooms are assumed to be equal, so that the flow rate into each is F/2; without considerably more information on the test space, this assumption is the best that can be made. The flow rate from room 2 to room 1 is then taken to be $\beta F/2$, where $\beta$ is the mixing factor; the mass-balance the requires that the flow rate from room 1 to room 2 be $(1+\beta)F/2$. The mixing factor runs from zero, representing no mixing between rooms, to infinity, representing perfect mixing. Tracer gas is assumed to enter rooms 1 and 2 at rates $Q_1$ and $Q_2$, respectively. To simplify the analysis, we further assume that $V_1/V_2=1$ (the ratio in the test space is actually 1.14). The advantage gained in clarity outweighs the small error resulting from this assumption.

Assuming that $C_1(0) = C_2(0) = 0$, the concentrations in rooms 1 and 2, with the source in room 1, are

$$C_1(t) = (1 - a(t))\,\frac{2+\beta}{1+\beta}\,\frac{Q}{F} + b(t)\,\frac{Q}{V} \tag{5}$$

$$C_2(t) = (1 - a(t))\,\frac{Q}{F}, \tag{6}$$

where a(t) and b(t) are defined below (eqns 9 and 10). The steady-state concentrations are, for $\beta << 1$, $C_1 = 2C_2 = 2Q/F$; for $\beta >> 1$, $C_1 = C_2 = Q/F$.

When the source is in room 2, we have

$$C_1 = (1 - a(t))\,\frac{\beta}{1+\beta}\,\frac{Q}{F} \tag{7}$$

$$C_2 = (1 - a(t))\,\frac{Q}{F} + b(t)\,\frac{Q}{V} \tag{8}$$

The steady-state concentrations are, for $\beta << 1$, $C_1 = 0$ and $C_2 = Q/F$; for $\beta >> 1$, $C_1 = C_2 = Q/F$. In each case,

$$a(t) = \frac{\lambda_1 \exp \lambda_2 t - \lambda_2 \exp \lambda_1 t}{\lambda_1 - \lambda_2} \tag{9}$$

$$b(t) = \frac{\exp \lambda_1 t - \exp \lambda_2 t}{\lambda_1 - \lambda_2}, \tag{10}$$

where

$$\lambda_1 = -\frac{F}{2V}; \qquad \lambda_2 = -\frac{F}{V}(1 + \beta) \tag{11}$$

and F is the fan flow rate, V the volume of each room, Q the source strength, t the time, and $\beta$ the dimensionless mixing factor.

For experimental situations in which there is good intra-room mixing, the mixing factor provides a useful measure of ventilation efficiency. When the intra-room mixing is poor, the use of average concentrations over each room to determine $\beta$ can indicate the extent of inter-room transport. It is important to note, however, that the relationship between $\beta$ and efficiency depends on what one is trying to accomplish. If it is desired to remove pollutants from rooms other than that in which the fan is located, a ***high*** mixing factor is indicative of high efficiency; if, on the other hand, it is desired to prevent pollutants originating in the fan room from moving to other rooms, a *low* mixing factor is indicative of high efficiency.

When the fan and pollutant source are in the same room, it is possible for the concentration in the outlet duct to be greater than any measured concentration in the test space. In these circumstances, knowledge of the inter-room mixing factor is useful as an indicator of the degree to which pollutants are transported out of the source room, but it can provide only a partial measure of local efficiency. It may be necessary to use a model in which one "room" is a relatively small area embracing the source and the outlet; it is then possible to relate local ventilation efficiency to the mixing factor(s) between this space and the remainder of the area under study. The theory for this model is developed in Appendix B.

### *Source in Room 1: Results and Discussion*

Results of window fan tests with the source of tracer gas in room 1 are summarized in Table 3. The spatial average concentrations in each of the two rooms at the end of the injection period have been tabulated as $C_1$ and $C_2$; the concentration in the outlet duct and the ratio of $C_1$ to $C_2$ are also shown. A visual fit of the data to the predictions of the mass-balance model has been made, and a rough value of the mixing factor has been obtained. The average temperatures over the injection period in room 1, room 2, room 3 (the room adjacent to the test space, as shown in Figure 1), and outdoors are tabulated as $T_1$, $T_2$, $T_3$, and

$T_{out}$.

The results suggest a dependence of the mixing factor on ambient conditions, although they do not support any conclusion as to the nature of that dependence. Since the variation in inter- and intra-room transport over the course of the experiments must come from the indoor-outdoor and inter-room temperature differences, it might be expected that a correlation would be found between these differences and the mixing factor. The tests at 10.3 l $s^{-1}$ support this hypothesis, the relatively high mixing factor for the second run corresponding to a relatively high indoor-outdoor temperature difference and the mixing factors for the first and third runs being roughly equal. The tests at 20.3 l $s^{-1}$, however, display a weak correlation at best.

A detailed picture of the results at 20.3 l $s^{-1}$ is given in Figure 5, in which three runs, each consisting of a three-hour injection and subsequent decay, are shown. The first frame shows the concentrations at the three sampling points in room 1 (B1-B3 of Figure 1) and that in the outlet duct (B4), while the second shows the concentration in the inter-room doorway (B5) and those in room 2 (B6-B8). Superimposed on both frames are the indoor and outdoor temperatures.

The figure illustrates the difficulty of applying the mass-balance model to situations in which ambient conditions are changing. In the first run, we see a slow initial increase in the room 2 concentration due to the absence of significant temperature differences, followed by a rapid increase as the outdoor temperature declines. In the second, we see an exponential increase in the room 2 concentration, presumably as a result of the relatively steady temperatures. The relatively poor mixing seen within room 1 during this period may be accounted for by the increased influence of the characteristic air flow patterns of free convection over those of the fan. In the third run, there is a striking change in the nature of the concentrations at sunrise (~7:30 A.M), when the outdoor temperature begins to increase rapidly. The concentrations at points B2 and B3 and in the outlet duct (B4) decline, while that at B1 continues to rise sharply; the concentrations in room 2 become steady.

These results are sufficient to support the idea that ventilation efficiency is strongly influenced by air movement produced by ambient temparature differences or by local heating due to incident sunlight; they are insufficient to allow more than a qualitative description of the nature of that influence.

*Source in Room 2: Results and Discussion*

The results of experiments with the tracer gas injection in room 2 are summarized in Table 4, in which the spatial average concentrations for the two test rooms and the outlet concentration, all taken at the end of tracer gas injection, are shown; the time-average temperatures over the course of the injection are also tabulated. The distinctive result is the absence of any significant difference in the concentrations in the two rooms until a flow rate of 45.2 l $s^{-1}$ is reached. The increased ventilation efficiency at higher flow rates is reflected in the ratio between the measured concentration in the outlet duct and that in the test space, which is seen to increase sharply at a flow rate of 39.2 l $s^{-1}$. Because of the changes in this ratio over time, its use, or that of a more sophisticated indicator, such as the integrated concentration over time, as an measure of efficiency cannot be supported.

Figure 6 is illustrative of the time-dependent behavior. The several concentrations for two runs at 39.2 l $s^{-1}$ are shown; the temperatures have been superimposed in the same manner as in Figure 5. The relatively small changes in the indoor-outdoor temperature difference do not appear adequate to account for the differences between the first and second runs, although it is possible that the effect of very small changes in the pattern of air movement, especially near the source, may be much greater than expected. In any case, it is clear that the similarity of the concentrations at the end of the injection period for the two runs gives a misleading picture of the actual situation. Integration of the central room concentrations over the time of the injection would be equally misleading, particularly in view of the large fluctuations seen in run 1.

The mass-balance model does not agree well with the results shown in Table 4. We might expect the concentrations in room 1 to be lower than those in room

2, with the ratios of the former to the latter diminishing as the flow rate increases, due to the effect of the fan in overcoming the naturally occurring transport of air from room 2 to room 1. Instead we find that, up to 31.7 l $s^{-1}$, the concentrations are roughly comparable and that, in some cases, there is actually a greater concentration in room 1 than in either room 2 or the outlet. This situation might be accounted for by assuming that the infiltration rate of room 2 is much greater than that of room 1, in which case the concentrations predicted by the model become roughly equal, although there is no evidence to support such a assumption.

At the two highest flow rates, it is useful to compare the results with a model in which a small area near the outlet duct is treated as one of the two rooms and the remainder of the test space as the other. The principal assumption of this model is that perfect mixing prevails throughout much of the space; this assumption appears to be supported by the data taken at the two highest flow rates. The data at 39.2 l $s^{-1}$ proves to be consistent with a mixing factor of ~0.75 and that at 45.2 l $s^{-1}$ with a factor of ~0.35. In particular, the second run at the former flow rate shows good agreement with the time-dependent predictions of the model (Appendix C). The fit of the remaining two runs, one at the former and one at the latter rate, is far less satisfactory; different ambient conditions may account for the discrepancy between theory and experiment.

The results of the wall fan experiments with both source placements are discussed further in Appendix C.

## SUMMARY

We have conducted a series of experiments designed to investigate the local ventilation efficiency of range hoods and window fans, and to determine the usefulness of a simple mass balance model in predicting the efficiency of the latter devices. The tests show that, when a heated tracer gas is used, the local efficiency of the range hood increases roughly linearly with flow rate. An efficiency of 77% was attained at a flow rate of 60.0 l $s^{-1}$. It may be inferred that flow rates of 75-100 l $s^{-1}$ will produce efficiencies approaching 100%, although further testing must be done if the inference is to be proven. With the tracer gas unheated, the effect of the range hood depended significantly on ambient conditions as well as on flow rate, so that no definition of efficiency could be supported.

Tracer gas concentrations obtained with the window fan in operation and the source in the room adjacent to the fan agree with the two-chamber mass-balance model so long as ambient conditions do not vary widely over the course of the experiment. Calculated values of $\beta$, the mixing factor, ranged from 1.0 to 17.5, depending on fan flow rate and ambient conditions, assuming the infiltration rates of the two rooms to be equal. If the infiltration rate of the fan room were higher than that of the adjacent room, as the data from the experiments with the source in the fan room appear to indicate, the calculated mixing factors would be higher.

When the fan and source are in the same room, the results cannot be wholly accounted for by the mass-balance model. The measured concentrations in the outlet duct agree quite well with the predicted room 2 concentration, which does not depend on the mixing factor. However, the measured room 2 concentrations at the two highest flow rates proved to be considerably lower than predicted, and were roughly comparable to the room 1 concentrations. A two-room model in which one "room" consists only of a relatively small area near the outlet can account for these results; the calculated values of the mixing factors are 0.35 and 0.75. The use of integrated concentrations as a measure of efficiency, which would naturally suggest itself, is precluded by the possible dependence of the results on ambient conditions.

**ACKNOWLEDGEMENTS**

I should like to thank my colleague W. Nazaroff, in particular, for his support in the undertaking of the work herein described, and for his careful review of this report. I should also like to thank P. Cleary, W. Fisk, and D. Grimsrud for their assistance.

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Buildings Energy Research and Development, Buildings Systems Division of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098, and by the U.S. Environmental Protection Agency, Office of Research and Development.

Although the research described in this article has been funded wholly or in part by the United States Environmental Protection Agency through Interagency Agreement AD-89-F-2A-062 to the U.S. Department of Energy, it has not been subjected to EPA review and therefore does not necessarily reflect the views of EPA and no official endorsement should be inferred.

## REFERENCES

1. Traynor G.W., Apte M.G., Dillworth J.F., Hollowell C.D., and Sterling E.M. (1982) The Effects of Ventilation on Residential Air Pollution Due to Emissions from a Gas-Fired Range. *Environment International*, *8*, 447-452.
2. ASHRAE (1980) *ASHRAE Handbook and Product Directory, 1980 Systems*, ch. 22, American Society of Heating, Refrigeration, and Air-Conditioning Engineers.
3. Sandberg M. (1981) What is Ventilation Efficiency? *Building and Environment*, *16*, *2*, 123-135.
4. Fisk W. (1984) Personal communication.
5. Nazaroff W.W., Revzan K.L., and Robb A.W. (1981) Instrumentation for a Radon Research House. University of California Lawrence Berkeley Laboratory Report LBL-12564, Berkeley, CA.

Table 1. Ventilation efficiency of range hood with heated tracer gas (see text for explanation).

| f ($l\ s^{-1}$) | ach | $C_0$ (ppm) | C (ppm) | $\eta$ | $\eta_c$ |
|---|---|---|---|---|---|
| 10.3 | 0.544 | 17.9 | 15.1 | .16 | .18 |
| 20.8 | 1.088 | 14.2 | 9.4 | .34 | .31 |
| 32.2 | 1.681 | 11.3 | 6.3 | .44 | .44 |
| 44.4 | 2.324 | 9.0 | 3.9 | .57 | .58 |
| 60.0 | 3.140 | 7.1 | 1.6 | .77 | .77 |

Table 2. Concentrations in the centers of rooms 1 and 2 and in the outlet duct, measured at the end of tracer gas injection, for range hood tests with heated tracer gas.

| Flow rate ($l\ s^{-1}$) | $C_1$ (ppm) | $C_2$ (ppm) | $C_{duct}$ (ppm) | $C_2/C_1$ | $C_{duct}/C_1$ |
|---|---|---|---|---|---|
| 10.3 | 19.96 | 17.42 | 20.86 | 0.87 | 1.05 |
| 20.8 | 11.67 | 10.17 | 14.74 | 0.87 | 1.26 |
| 32.2 | 8.61 | 7.38 | 11.34 | 0.86 | 1.32 |
| 44.4 | 5.59 | 4.70 | 8.66 | 0.84 | 1.55 |
| 60.0 | 2.63 | 0.71 | N/A | 0.27 | N/A |

Table 3. Fan experiments with tracer gas source in room 1: all concentrations are taken at the end of tracer gas injection; the room 1 and 2 concentrations ($C_1$ and $C_2$, respectively) are spatial averages; $C_{duct}$ is the concentration in the outlet duct. The temperatures are time averages over the injection period; the subscripts indicate rooms 1, 2, and 3, and outside.

| Flow rate ($l\ s^{-1}$) | $C_1$ (ppm) | $C_2$ (ppm) | $C_{duct}$ (ppm) | $C_1/C_2$ | $\beta$ | $T_1$ (°C) | $T_2$ (°C) | $T_3$ (°C) | $T_{out}$ (°C) |
|---|---|---|---|---|---|---|---|---|---|
| 10.3 | 43.31 | 32.52 | 28.88 | 1.33 | 3.5 | 22.4 | 22.4 | 19.6 | 21.1 |
| | 35.77 | 33.81 | 33.26 | 1.06 | 17.5 | 20.6 | 20.2 | 20.3 | 13.4 |
| | 40.03 | 31.61 | 31.63 | 1.27 | 5.0 | 22.7 | 22.8 | 18.3 | 21.7 |
| 20.3 | 32.53 | 28.65 | 27.67 | 1.14 | 1.0 | 22.8 | 22.8 | 21.8 | 19.8 |
| | 29.41 | 25.56 | 25.09 | 1.15 | 1.5 | 21.2 | 20.8 | 19.8 | 13.9 |
| | 32.14 | 21.48 | 21.26 | 1.50 | 1.0 | 20.2 | 19.9 | 18.0 | 13.2 |

Table 4. Fan experiments with tracer gas source in room 2: all concentrations are taken at the end of tracer gas injection; the room 1 and 2 concentrations ($C_1$ and $C_2$, respectively) are spatial averages; $C_{duct}$ is the concentration in the outlet duct. The temperatures are time averages over the injection period; the subscripts indicate rooms 1, 2, and 3, and outside.

| Flow rate ($l\ s^{-1}$) | $C_1$ (ppm) | $C_2$ (ppm) | $C_{duct}$ (ppm) | $T_1$ (° C) | $T_2$ (° C) | $T_3$ (° C) | $T_{out}$ |
|---|---|---|---|---|---|---|---|
| 10.3 | 26.28 | 33.04 | 32.44 | 21.4 | 21.9 | 18.4 | 18.4 |
| | 33.18 | 32.53 | 30.80 | 20.0 | 19.7 | 19.2 | 14.6 |
| 19.4 | 19.42 | 18.99 | 16.32 | 21.6 | 21.4 | 21.0 | 17.5 |
| | 23.26 | 22.32 | 21.87 | 19.2 | 18.8 | 18.4 | 13.0 |
| 31.7 | 12.15 | 12.07 | 13.23 | 19.4 | 19.2 | 17.9 | 17.8 |
| | 11.87 | 11.80 | 12.77 | 18.2 | 17.8 | 17.4 | 13.4 |
| 39.2 | 4.64 | 4.69 | 9.11 | 19.9 | 19.7 | 18.8 | 15.9 |
| | 4.62 | 4.99 | 10.71 | 18.0 | 17.6 | 17.1 | 12.8 |
| 45.2 | 1.56 | 2.64 | 10.53 | 19.8 | 20.2 | 16.6 | 19.1 |

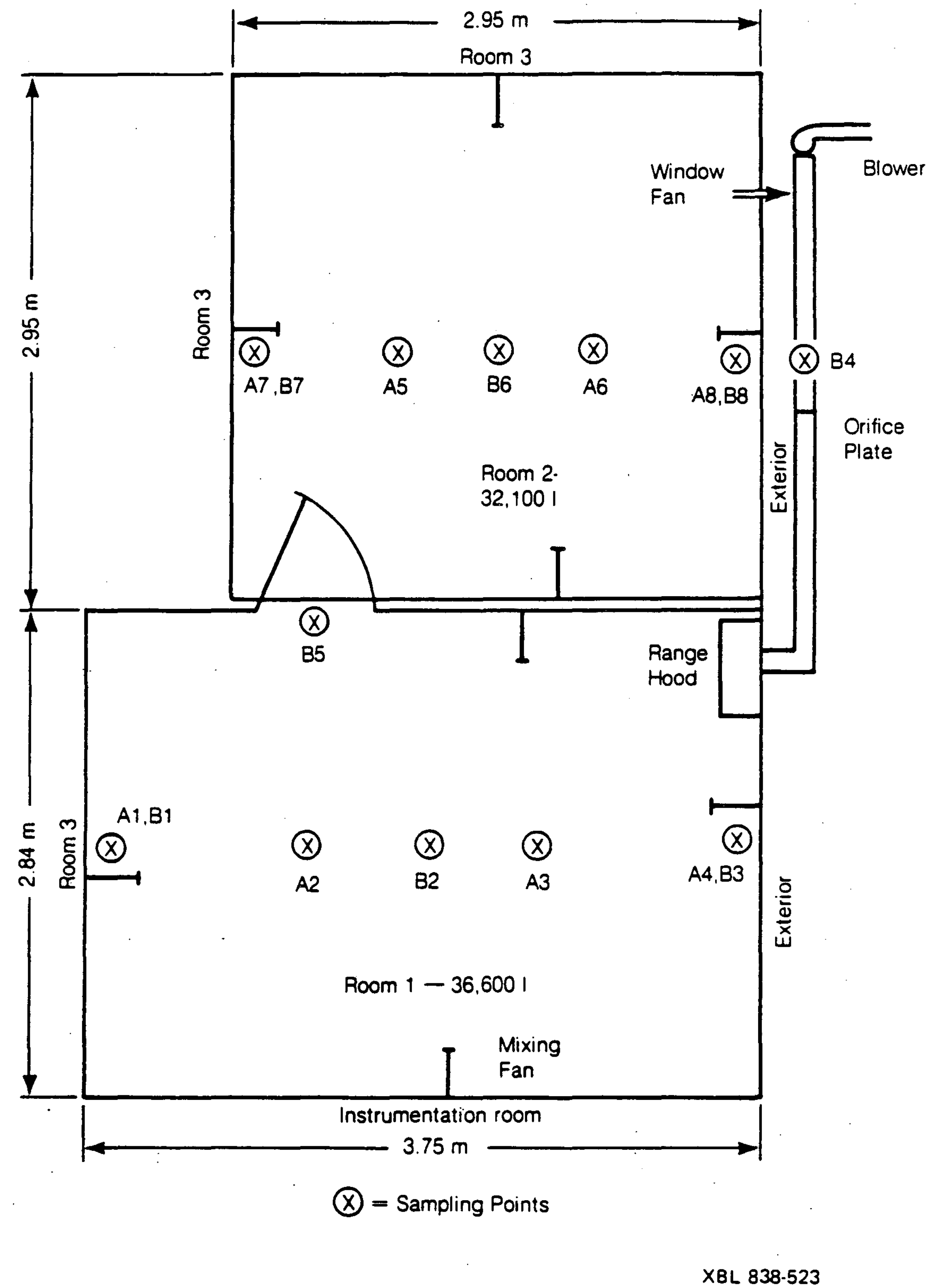



Figure 1. Test space configuration.

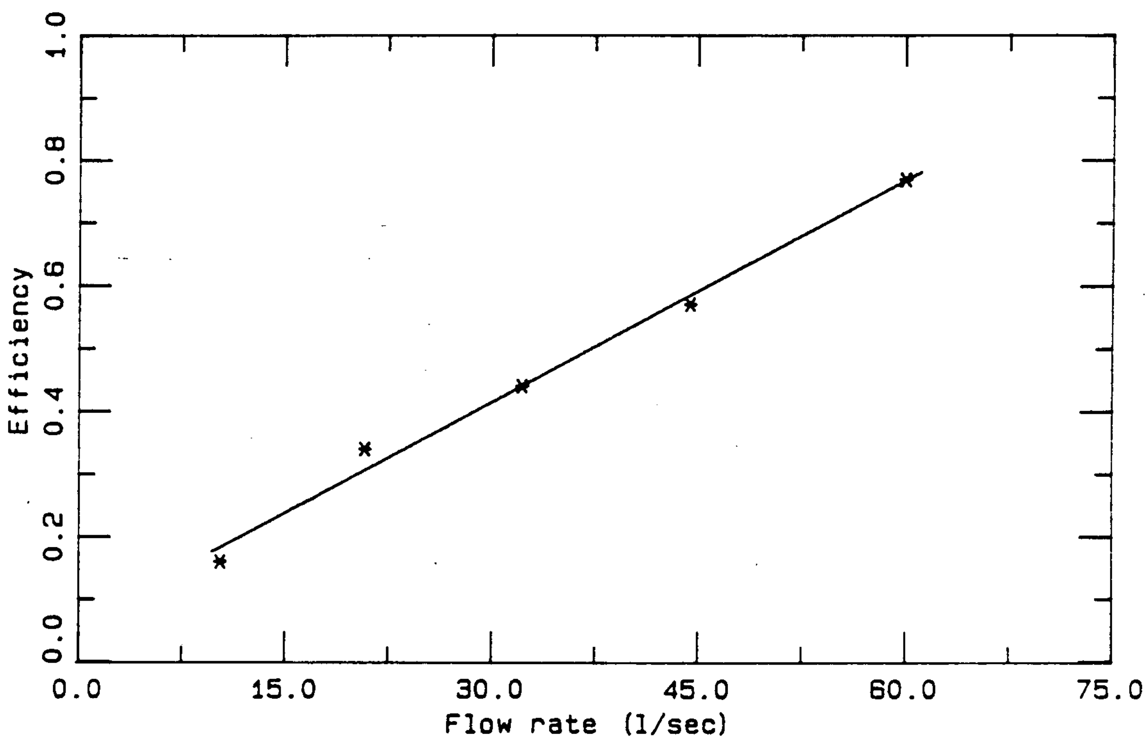
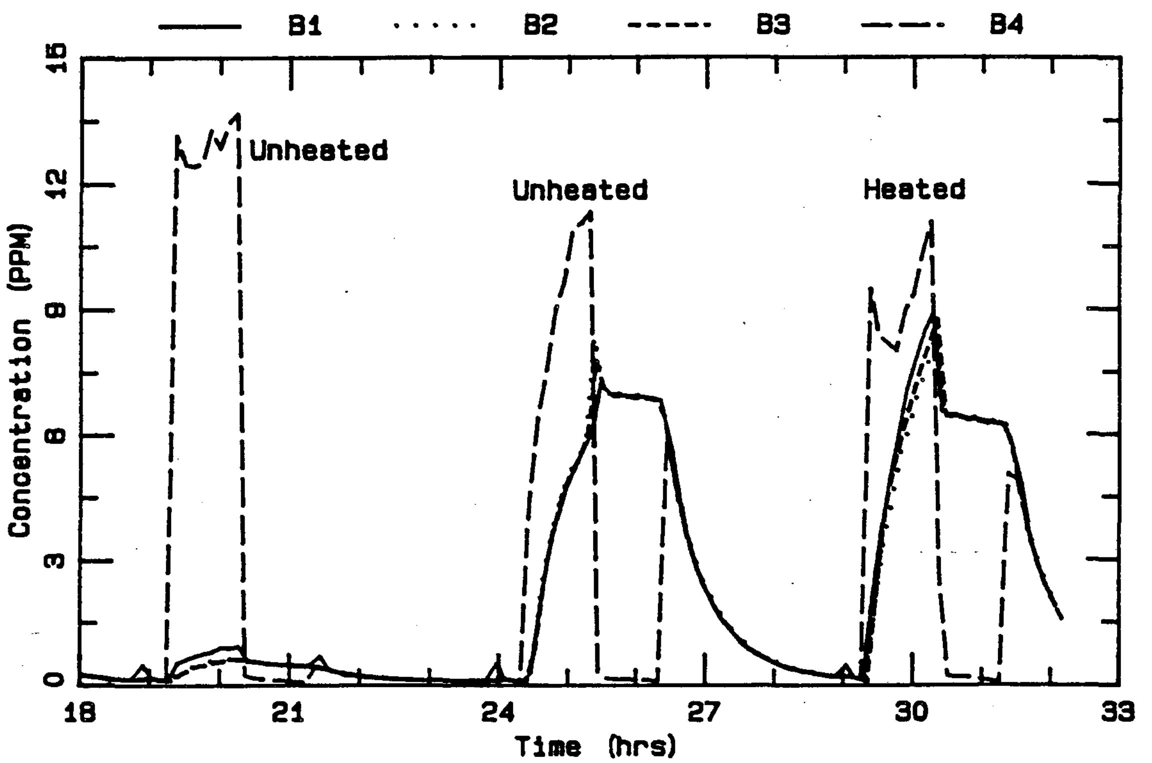



Figure 2. Range hood ventilation efficiency.



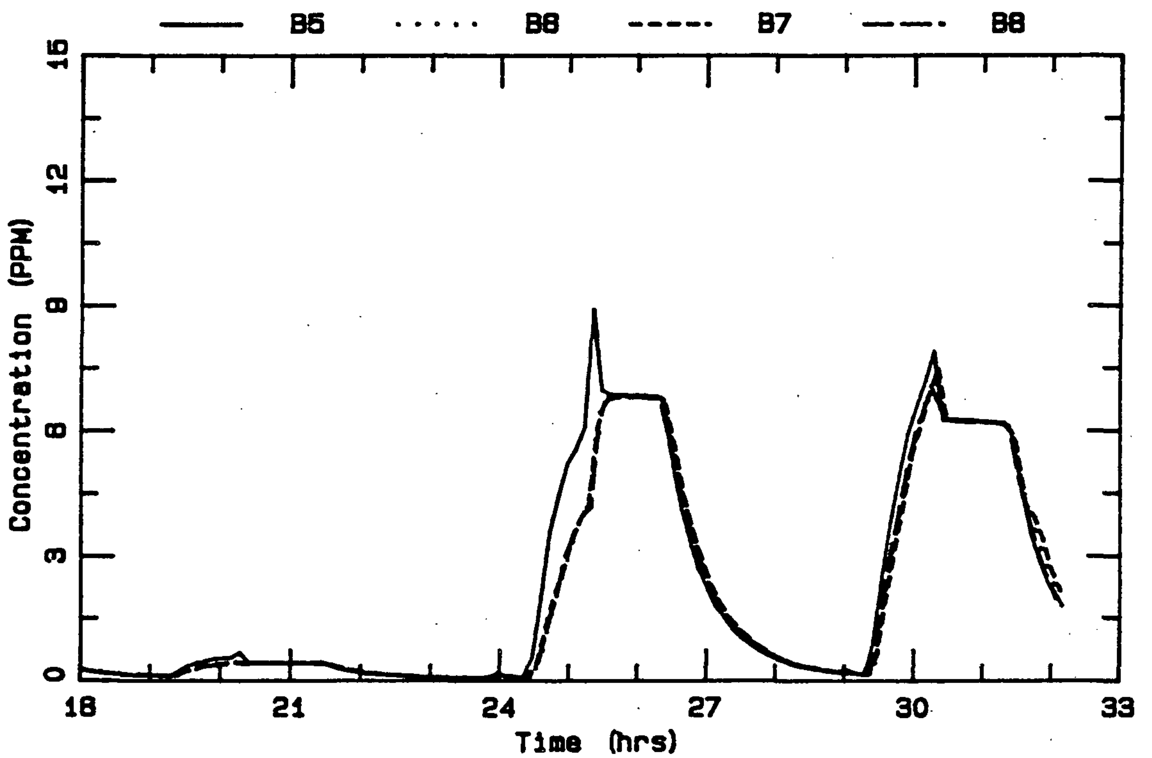




Figure 3. Range hood tests at 32.2 l/sec.

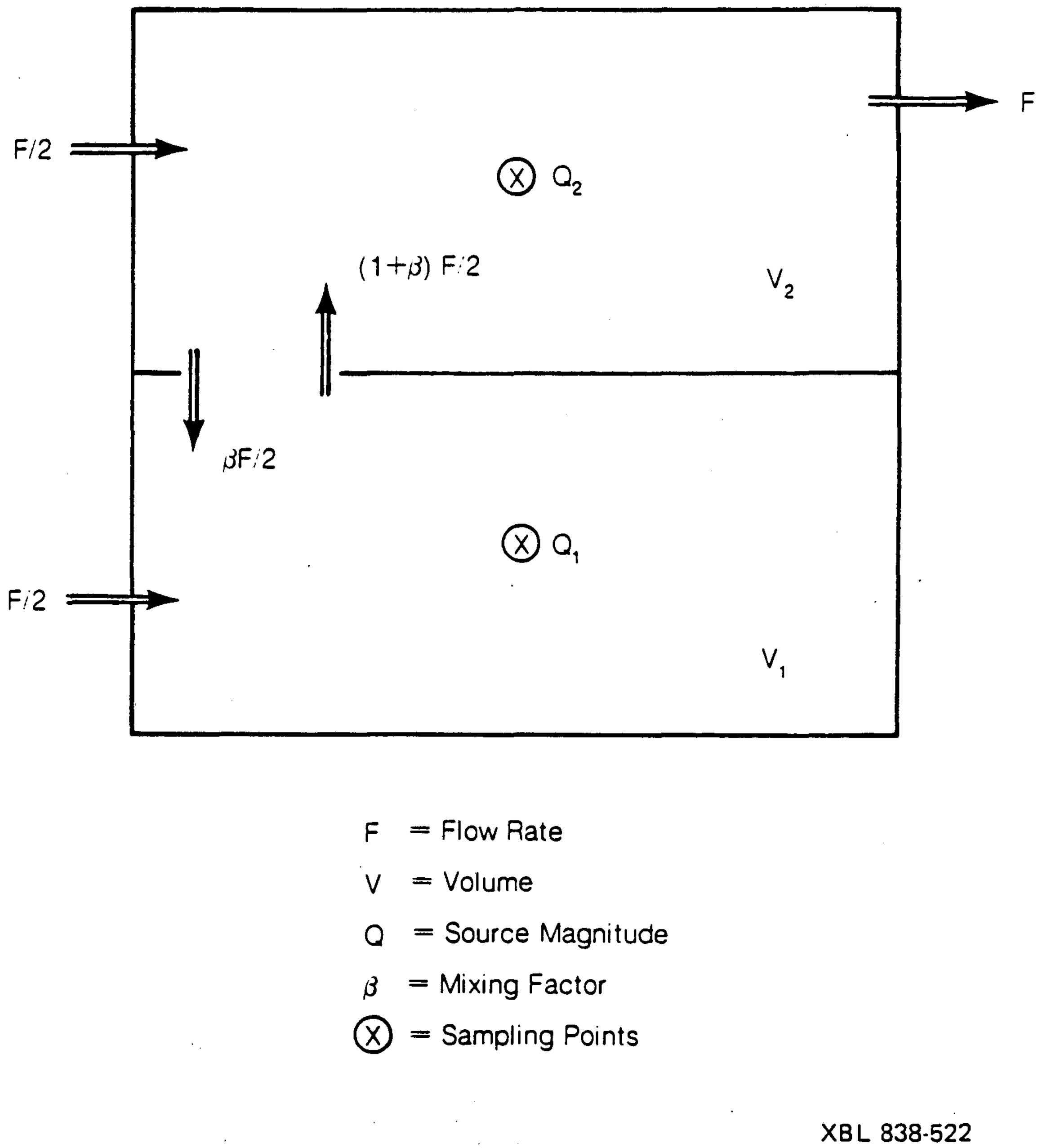



Figure 4. The mass-balance model.

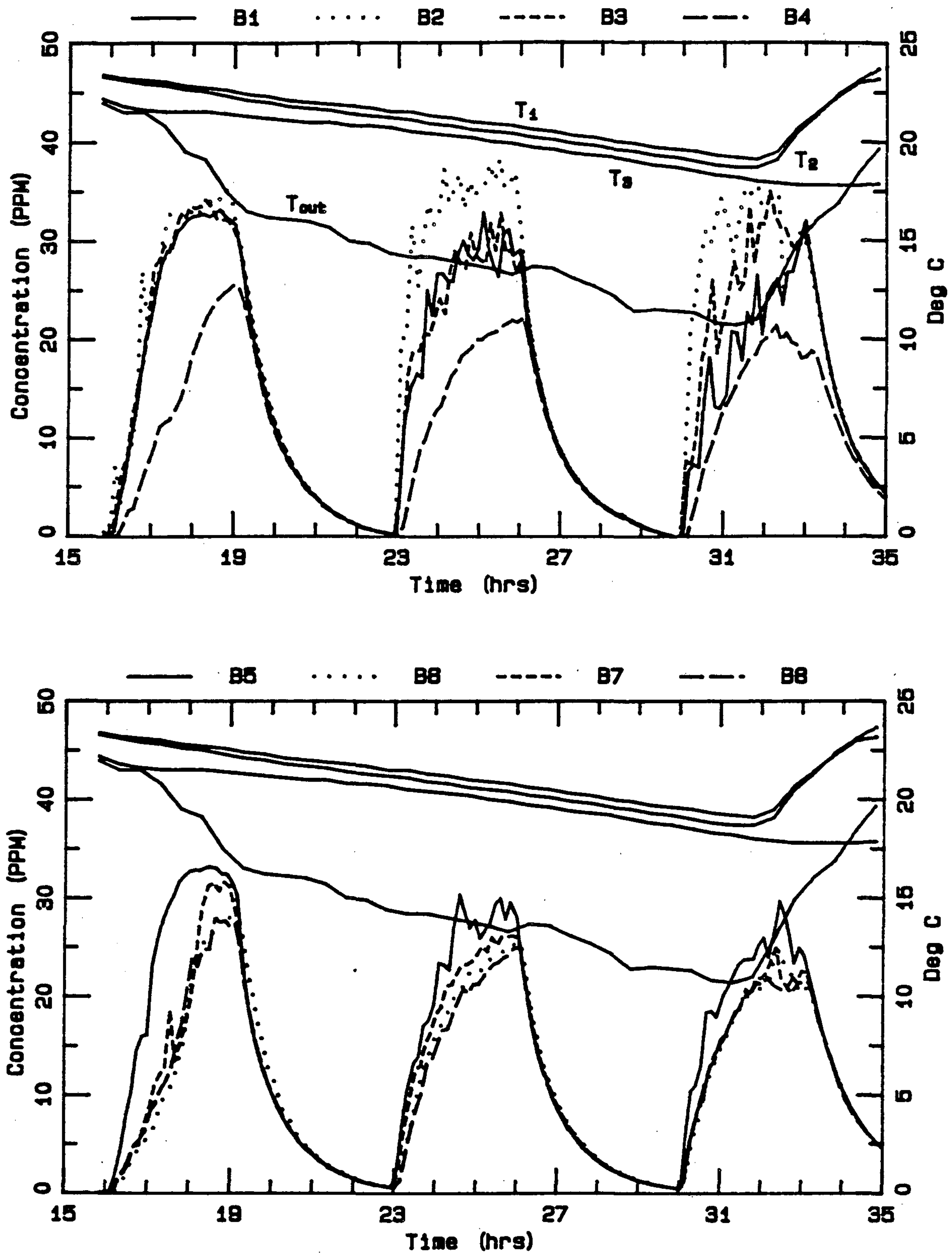



Figure 5. Fan with source in room 1 at 20.3 l/sec.

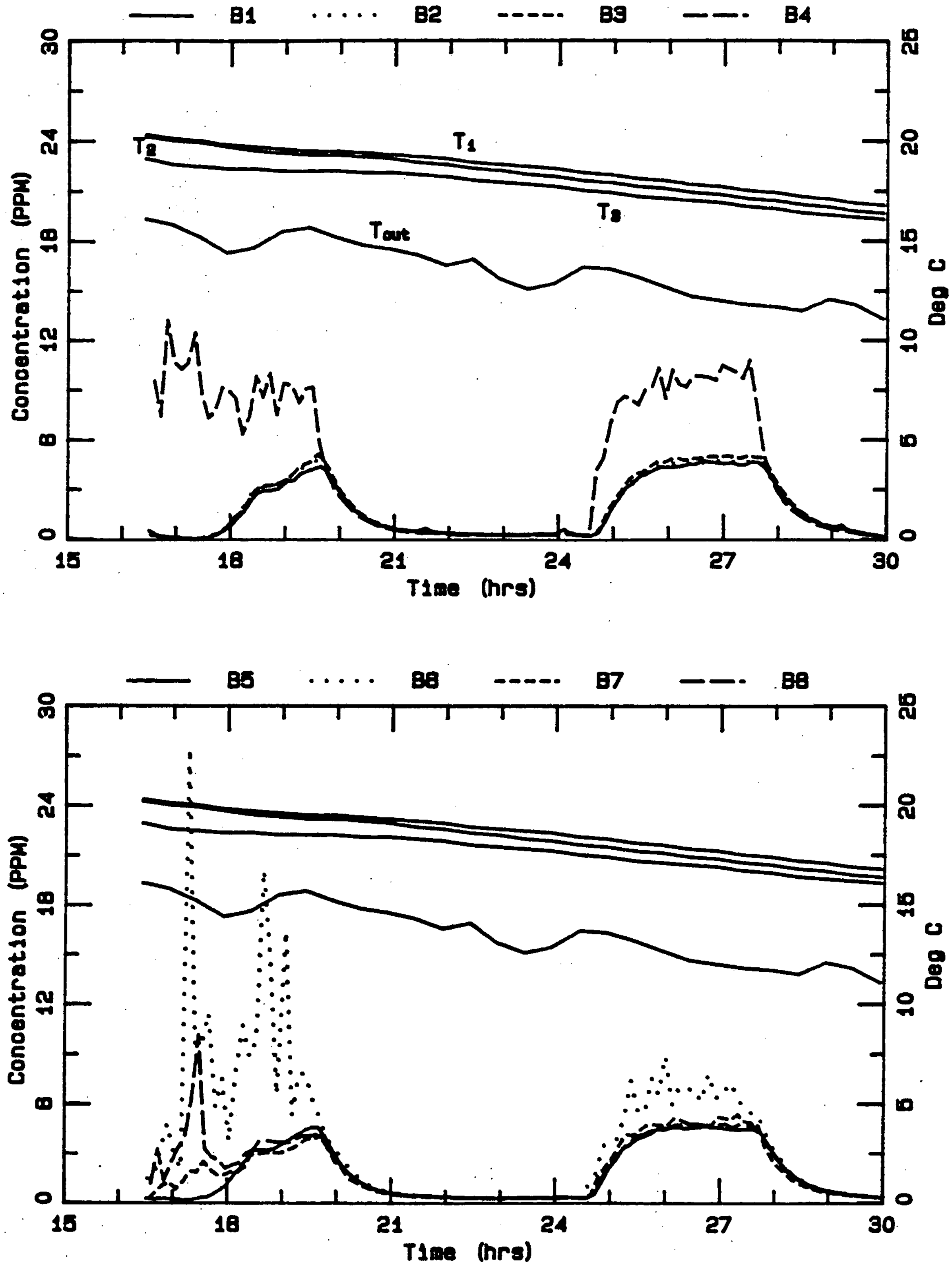



Figure 8. Fan with source in room 2 at 38.2 l/sec.

## APPENDIX A:

## FURTHER DISCUSSION OF RANGE HOOD EXPERIMENTS

To support the conclusion that the influence of ambient conditions on range hood efficiency is negligible when the tracer gas is heated, we present, in fig. A1, the results of five runs at 24.1 l/sec (1.26 $hr^{-1}$ exchange rate) with sampling at the points labelled "A" in Fig. 1 and no post-injection mixing. In the first two runs, with the burner off, results are strikingly variable, while the final three runs, with the burner on, show almost identical patterns for all eight sampling points. It was concluded that the buoyancy of the heated tracer gas dominates all other transport mechanisms.

In Figs. A2-A6, we present the results at each of five flow rates. At the rate of 60.0 l/sec, sampling was done at the points labelled "A" in Fig. 1; at the other rates, sampling was done at those labelled "B". At all rates, the previously discussed post-injection mixing period was employed. The use of the burner for a given test is indicated on the figures.

A distinctive feature of all these tests is the appearance of the tracer gas at sampling point 1, on the opposite side of room 1 from the range hood, before its appearance at the sampling point nearest the hood (point #3 in Figures A2-A5, point #4 in Figure A6), which suggests very strongly the prominent role of convection patterns in transport. It is inferred that the heated gas travels first to the ceiling, then across to the wall opposite the hood, down that wall, across the floor, and up the wall nearest the hood, appearing last in the center of the room. The pattern is most apparent at the highest flow rate, becoming successively less significant at each lower rate. The concentration in the outlet duct, in comparison with that within the test space, increases with each increase in flow rate, which is consistent with the rising efficiency.

At 60.0 l/sec, the tracer gas reaches room 2 first at the wall opposite the range hood (point 7), and only second at the point in the doorway between the rooms (point 5), which is consistent with the hypothesis of movement along the walls. At the lower flow rates, however, the first appearance, by a few minutes, is in the doorway. In all cases, the appearance at points 6 and 8, at the center and the wall most distant from the doorway, is effectively simultaneous. There is a sharp difference in the relative room 2 concentration between 60.0 l/sec and the lower rates. In the former case, the concentration in the center of room 2 is hardly significant (< 1 ppm) throughout the injection period, while in the latter it is on the order of 80-90% of the room 1 concentration. A considerable decrease in the inter-room mixing factor appears to have occurred between 44.4 and 60.0 l/sec. The question of inter-room mixing is discussed further in connection with the window fan tests.

When the injected tracer gas was unheated, results were highly variable. Not only were results at any given flow rate not repeatable, but the relationship between measured concentrations from one flow rate to another was not what would be intuitively expected. Figs. A2-A6 display the data. At 10.3 l/sec (Fig.

A2), the first run gives results in accordance with expectations, i.e., the concentration measured in the outlet duct rises immediately to a level double that of any point within the test space; the ultimate test space concentration is roughly 60% of that produced by a heated injection. The second and third runs, however, show a pattern almost identical to that produced by a heated injection, in fact producing concentrations which are somewhat higher (16.8 and 17.0 ppm as opposed to 15.1 ppm). Environmental data taken during the experiment do not provide an explanation for the difference between run 1 and runs 2 and 3, nor is there an explanation for the striking similarity between the latter two.

Results at 20.8 l/sec (Fig. A3) show two nearly identical runs, while an earlier test at 24.2 l/sec (Fig. A7), using sampling configuration "A" and without post-injection mixing, shows three dissimilar patterns, each with a lower concentration than that of any run of the former experiment. Two runs at 32.2 l/sec show wide differences (Fig. A4); one taken at 44.4 l/sec (Fig. A5) shows a concentration intermediate between the former two at a lower flow rate. At 60.0 l/sec, however, the concentrations in all runs with unheated tracer gas were consistently below measurable levels. It may well be that the air movement produced by a flow rate of this magnitude is sufficient to overcome all effects of differing flow patterns produced by free convection.

Fig. A7 presents a basis for speculation on the influence of ambient temperatures on the concentration patterns. During the three runs, the temperature difference between the test area and the adjacent (unused) room diminished steadily, suggesting that free convection was the dominant factor in transport. The spatial variations in concentration are consistent with this hypothesis, since a relatively cold wall opposite the range hood (point A1) would drive air in a circular pattern toward the wall nearest the hood (point A4), accounting for the relatively high concentration seen at that point in run 1. It would appear that, by the time of run 3, the direction of air movement had reversed itself and that the driving force had substantially diminished, since the tracer gas appears first at point A1 and the concentrations at all sampling points are considerably reduced. If a deeper understanding of the nature of air movement indoors is to be gained, more extensive experimentation under highly controlled conditions is necessary.

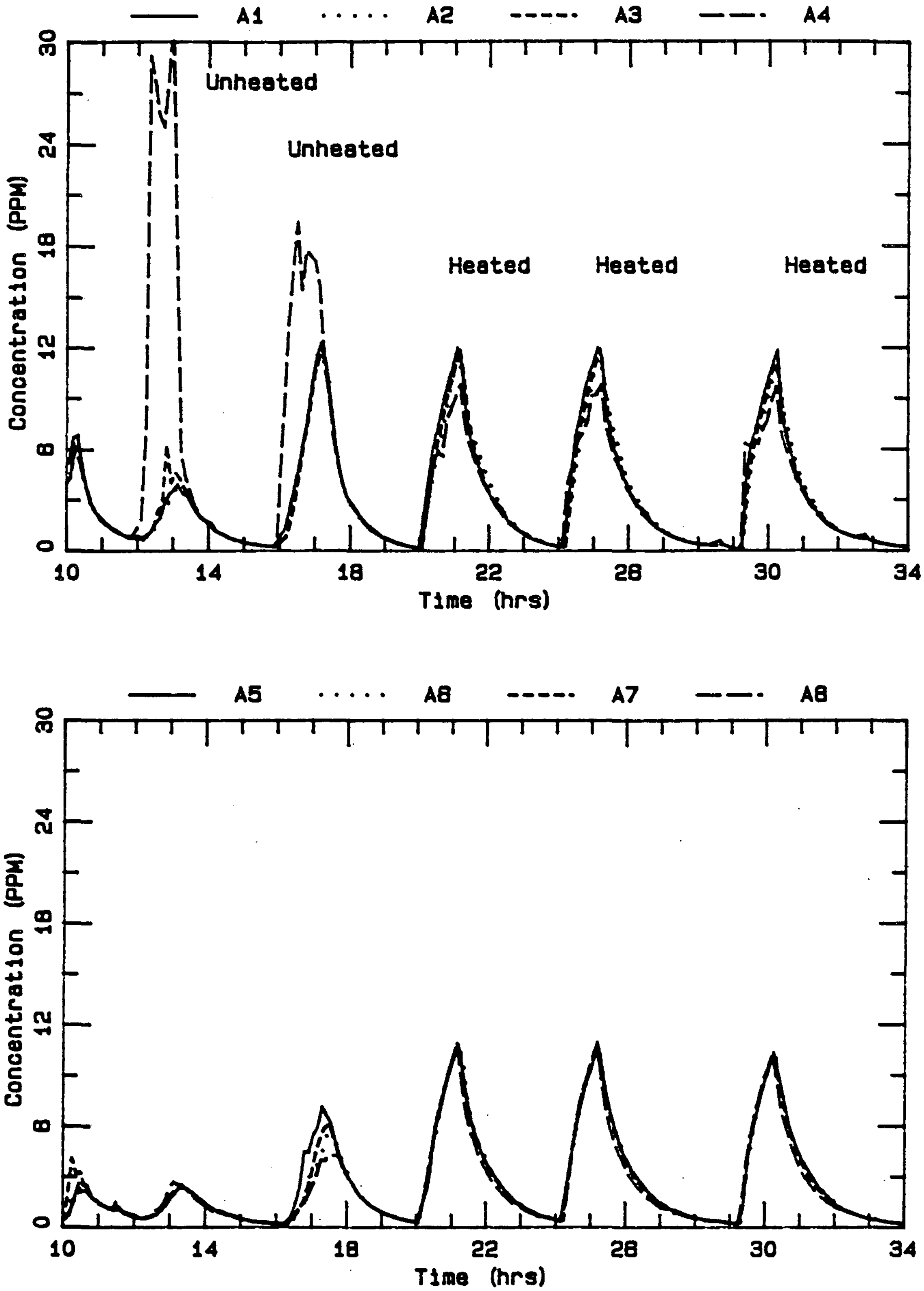
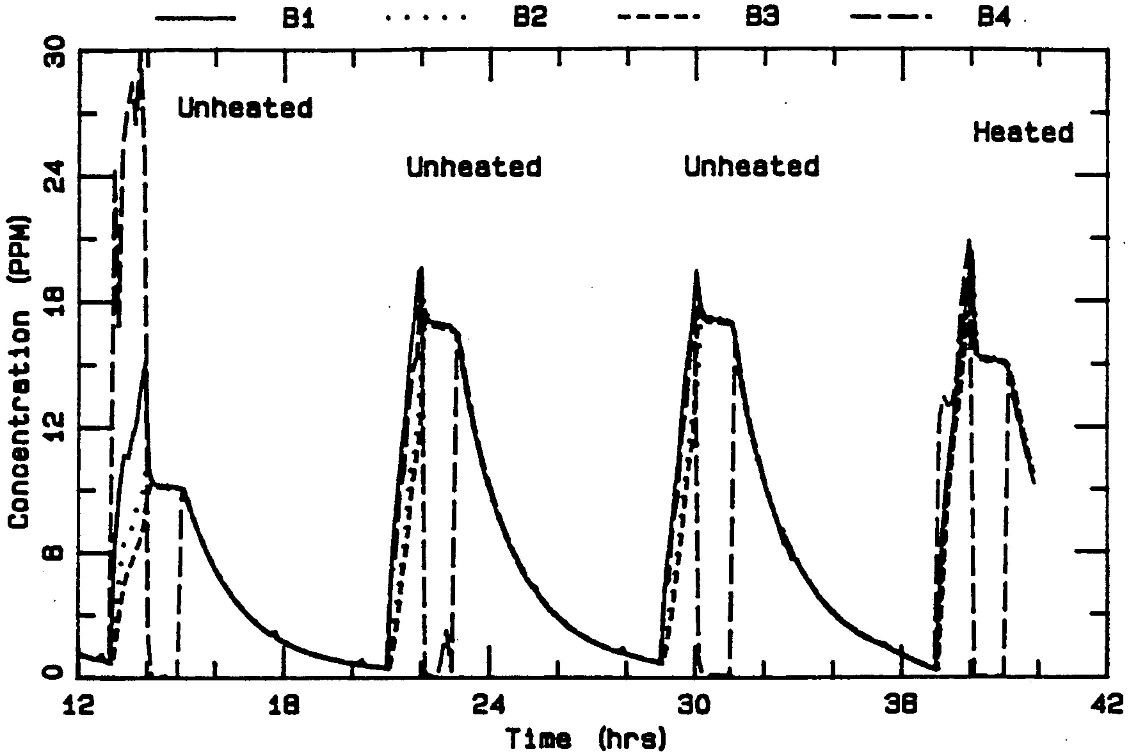



Figure A1. Range hood tests at 24.2 l/sec.



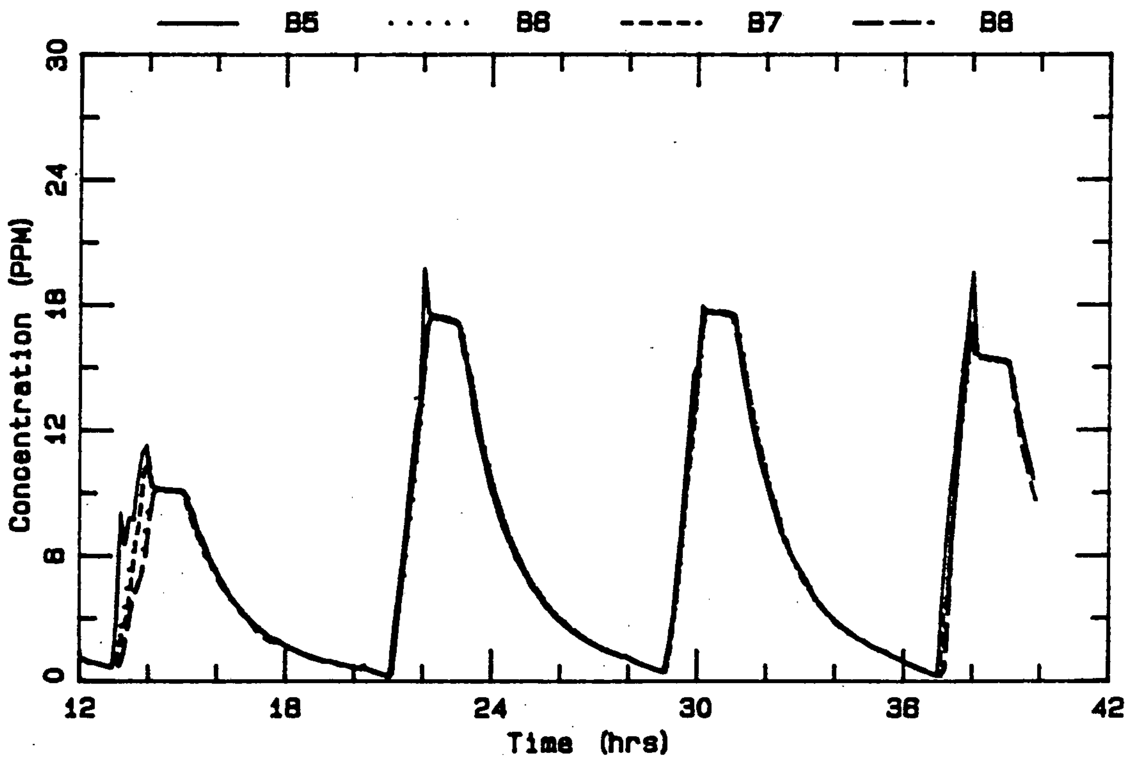
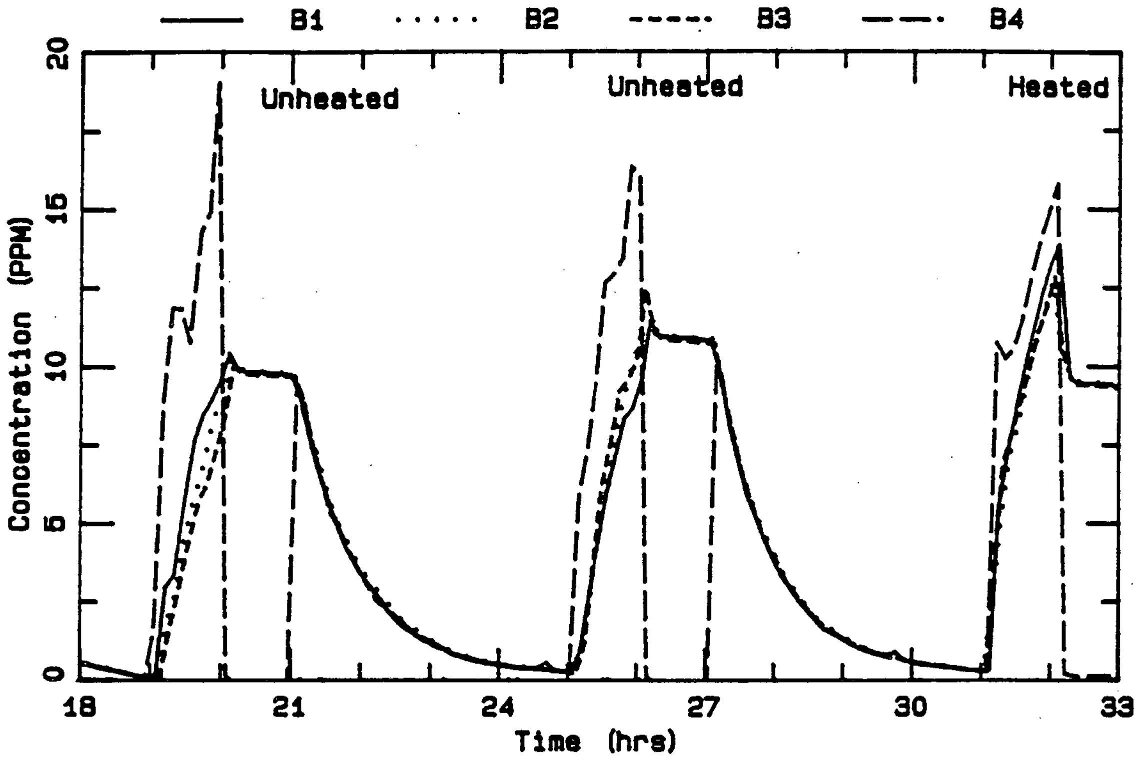




Figure A2. Range hood tests at 10.3 l/sec.



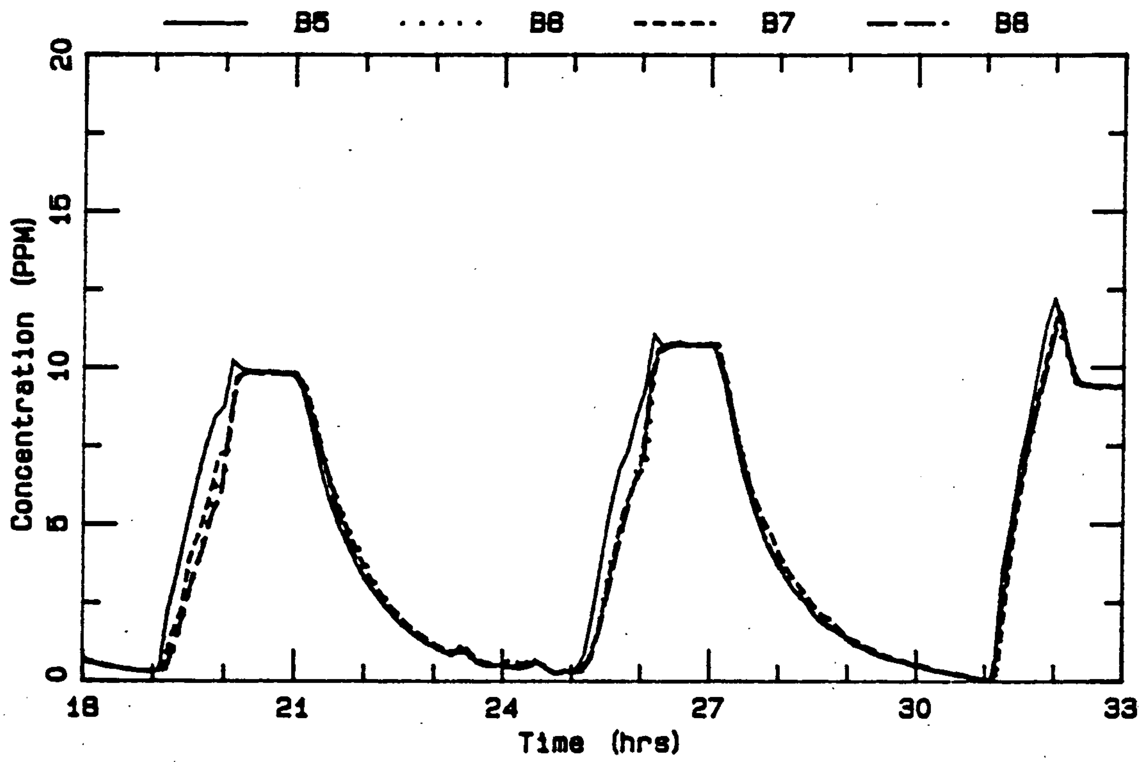
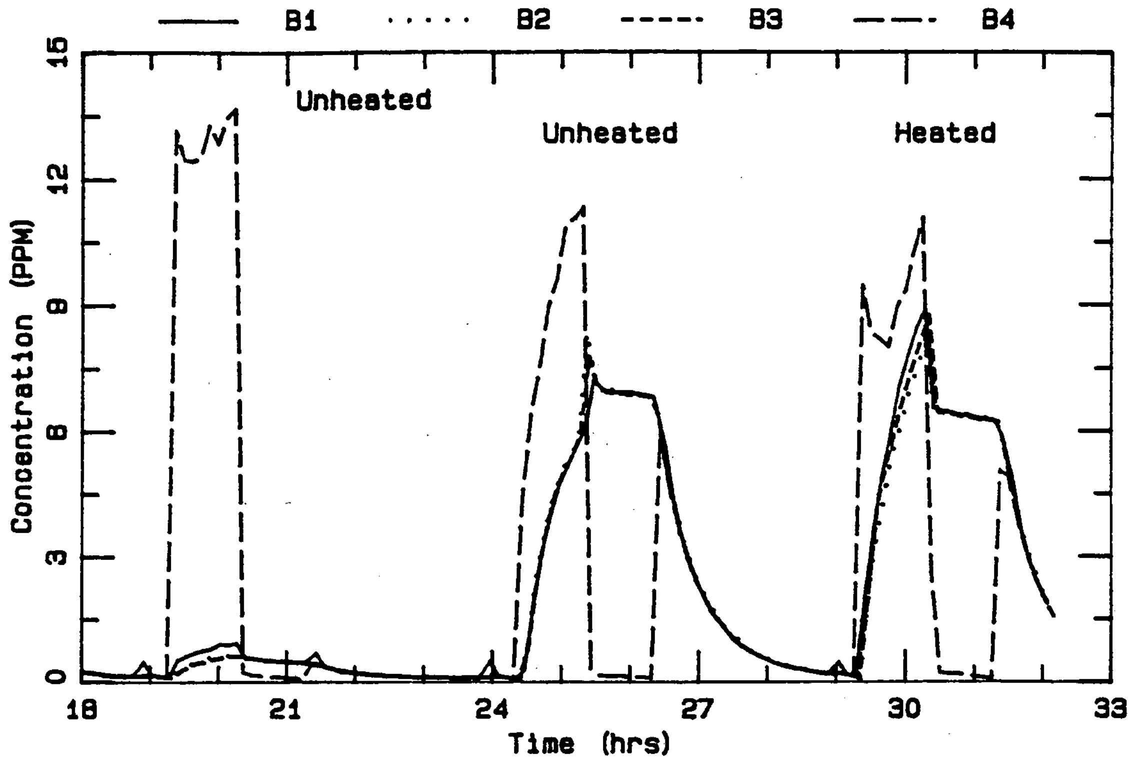




Figure A3. Range hood tests at 20.8 l/sec.



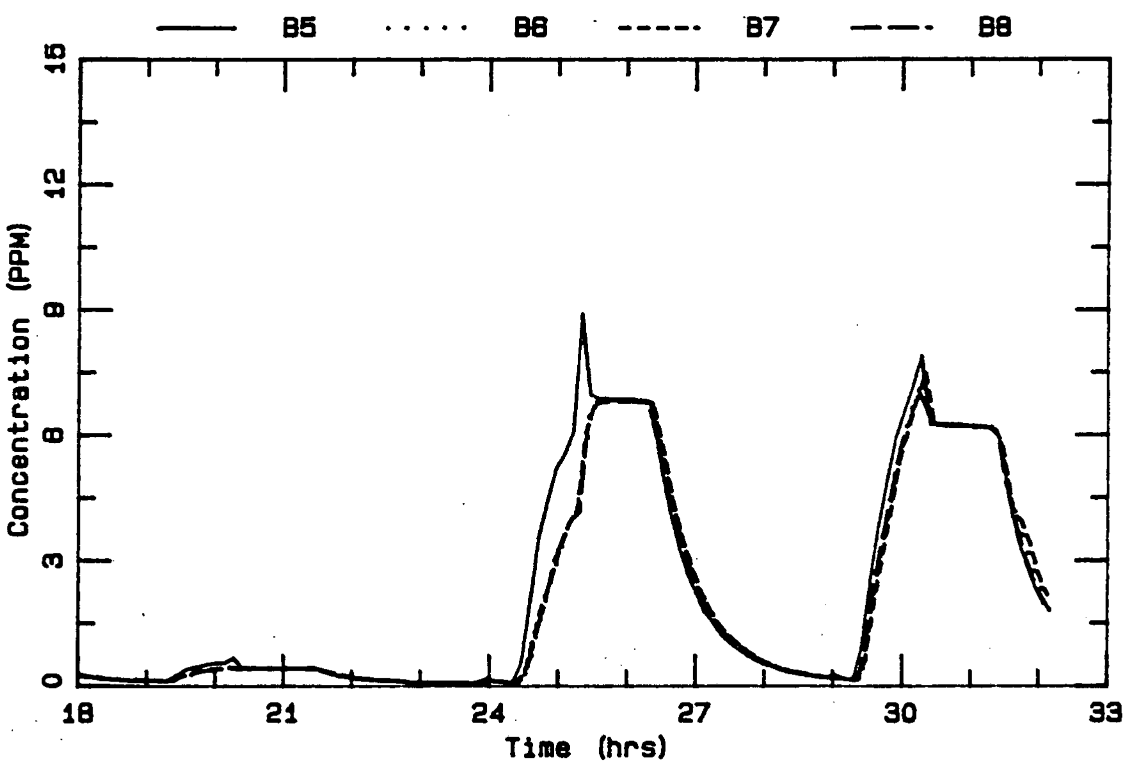




Figure A4. Range hood tests at 32.2 l/sec.

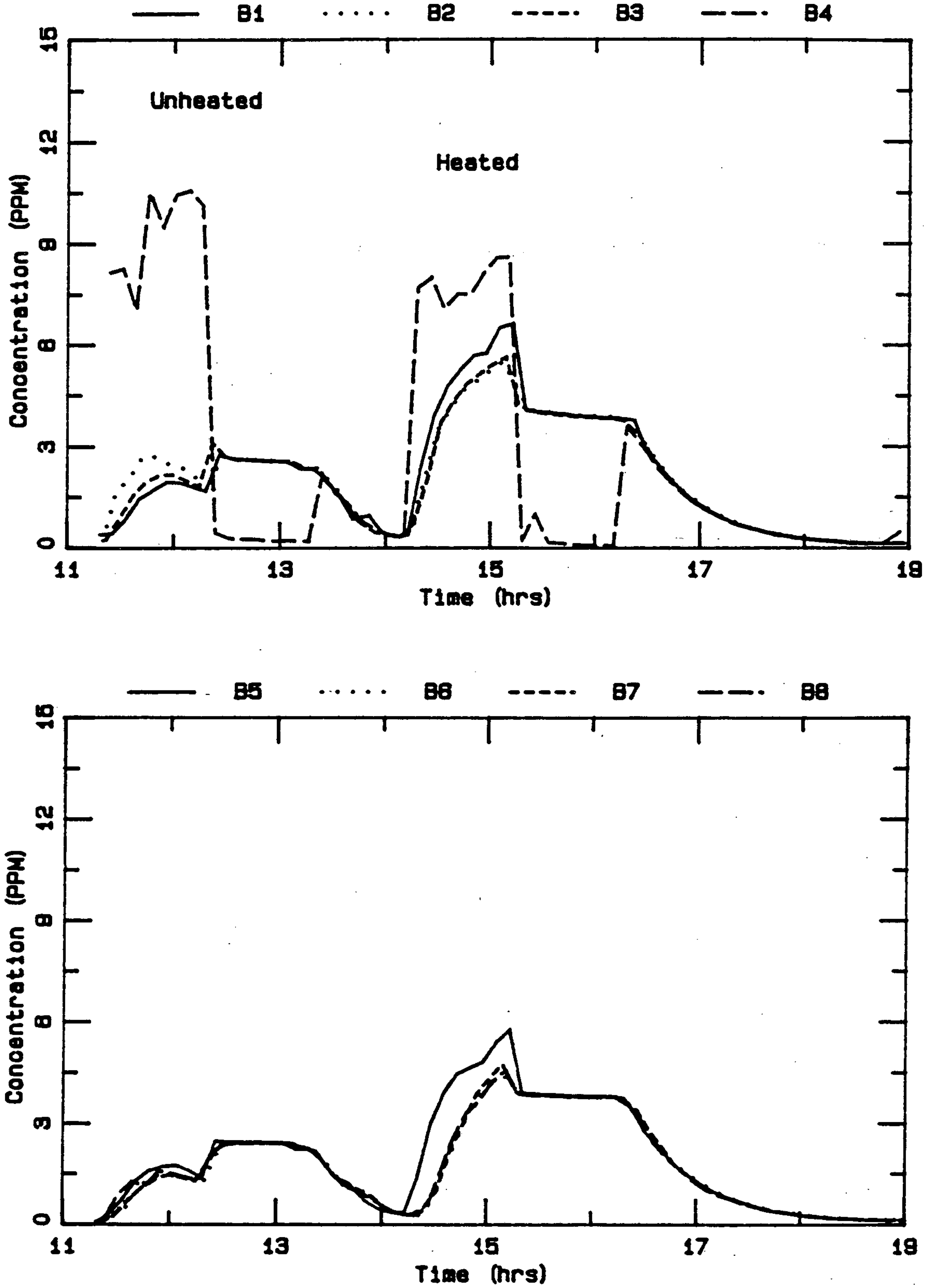
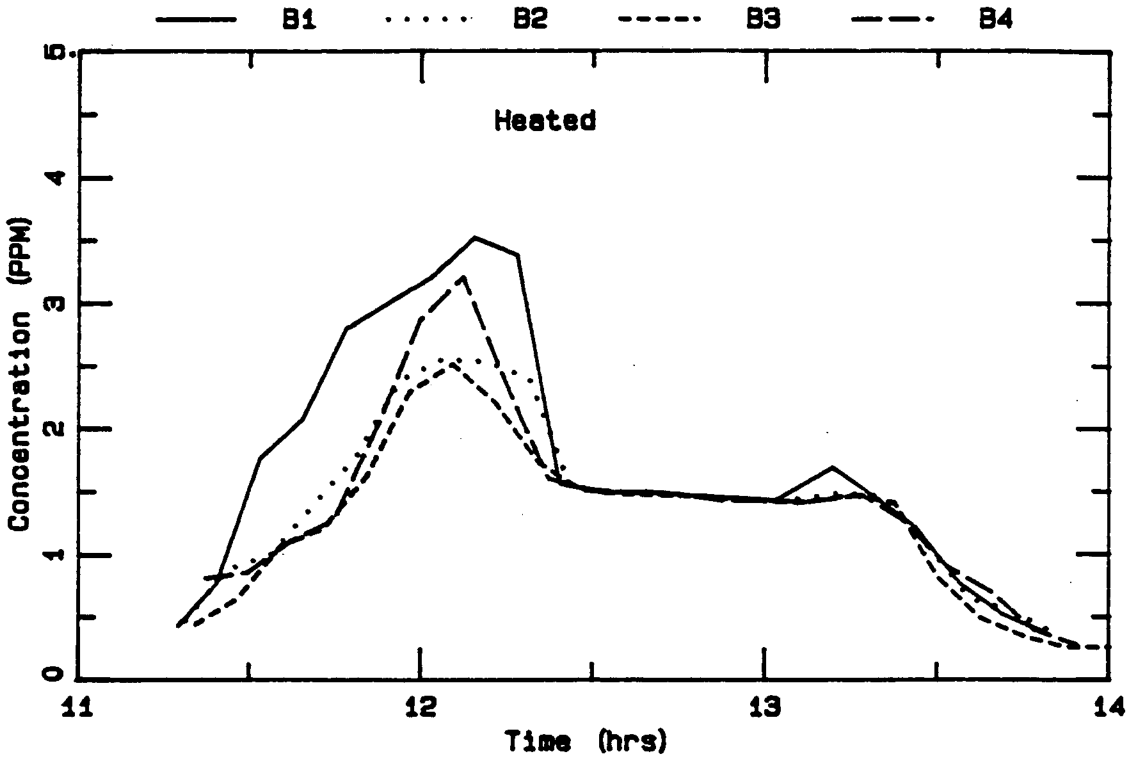



Figure A5. Range hood tests at 44.4 l/sec.



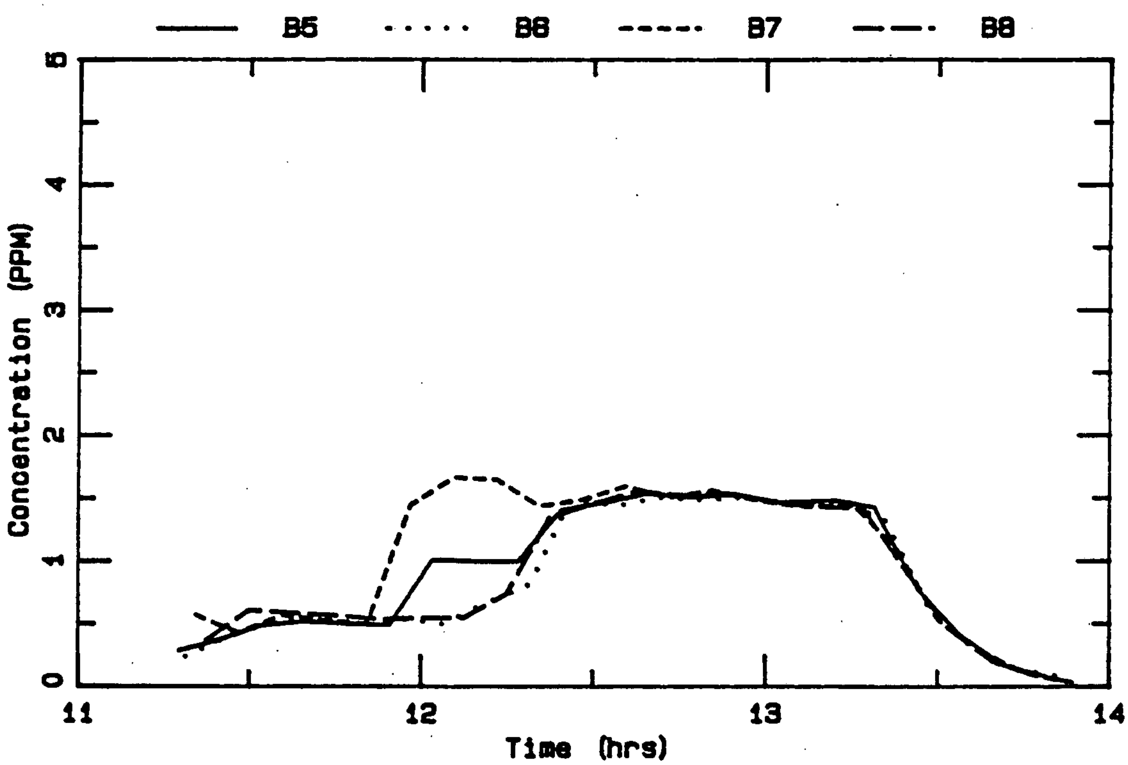
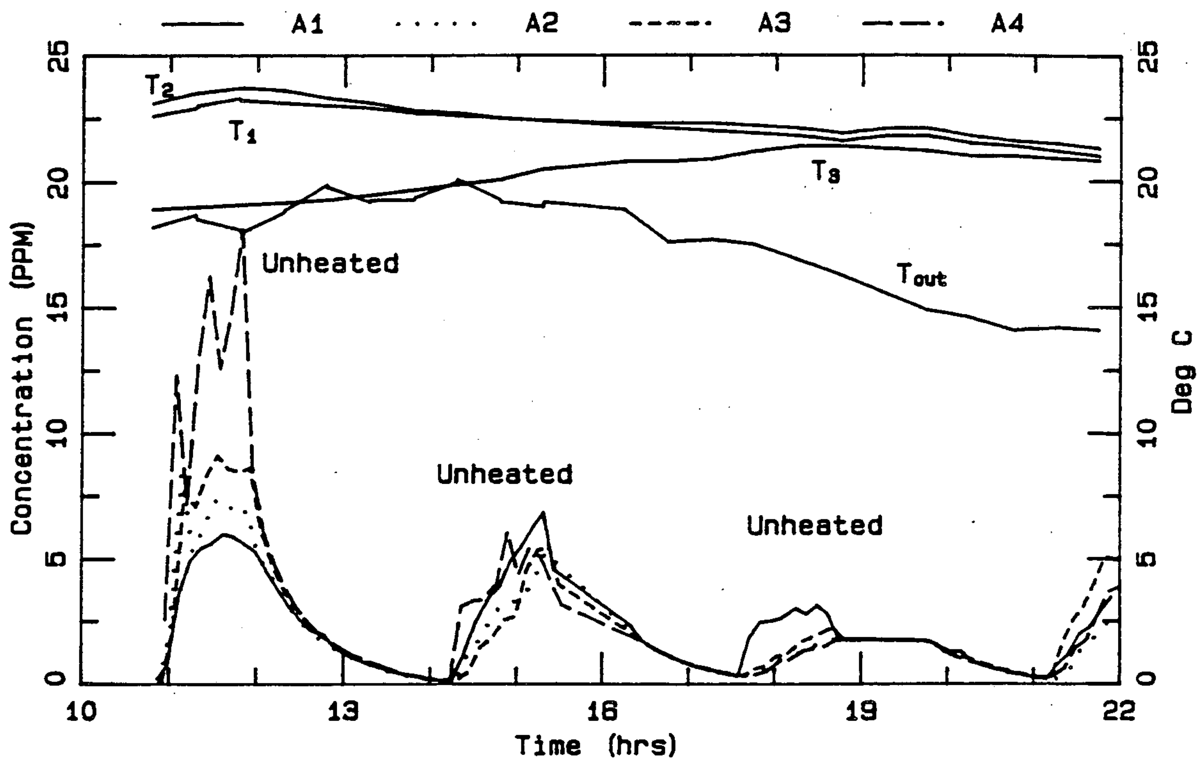




Figure A8. Range hood tests at 80.0 l/sec.



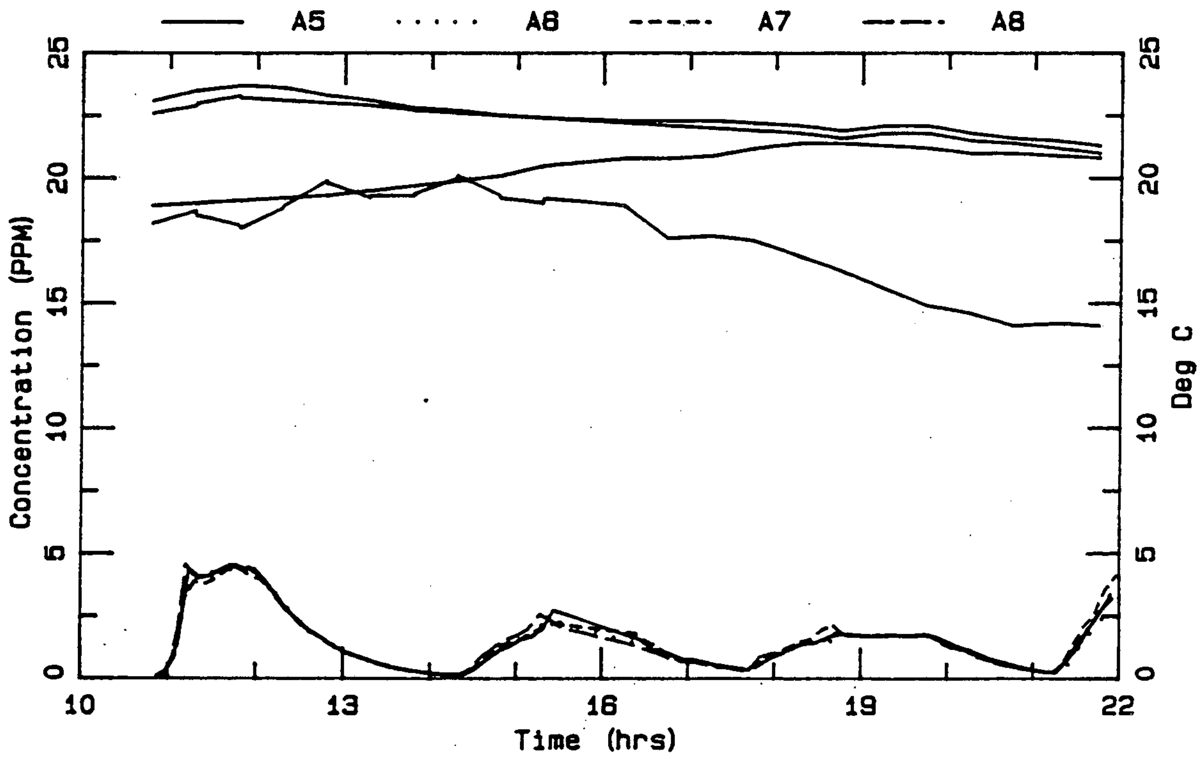




Figure A7. Range hood tests at 24.2 l/sec.

## APPENDIX B:

## THE TWO-ROOM MODEL WITH UNEQUAL ROOM VOLUMES

We assume that the volumes of the two rooms are unequal, and that $V_2$ is so small relative to $V_1$ that the infiltration rate of $V_2$ may be taken as 0. Under these conditions, Figure 4 must be changed: the flow rate into room 1 is now F and that into room 2 is 0; the flow rate from 2 to 1 is $\beta$F and that from 1 to 2 is $(1+\beta)$F. The concentrations in rooms 1 and 2 are again given by eqns. 5-8, except that the volume V of eqn. 5 must be replaced by $V_1$ and that of eqn. 8 by $V_2$. The eigenvalues, $\lambda_1$ and $\lambda_2$, which appear in eqns. 9 and 10, are now given by

$$\lambda_1 = \frac{F}{2}(A + B); \qquad \lambda_2 = \frac{F}{2}(A - B), \tag{B1}$$

where

$$A = -(1+\beta)\left(\frac{1}{V_1} + \frac{1}{V_2}\right) \tag{B2}$$

$$B = \sqrt{(1+\beta)^2\left[\frac{1}{V_1} + \frac{1}{V_2}\right]^2 - \frac{4(1+\beta)}{V_1 V_2}} \tag{B3}$$

For $\beta<1$, the eigenvalues are approximately

$$\lambda_1 = -\frac{F}{V_1}; \qquad \lambda_2 = -\frac{F(1+\beta)}{V_2}. \tag{B4}$$

Because $V_2$ appears only in combination with b (eqn. 10), which depends inversely on $\lambda_2$ and hence directly on $V_2$, the model remains valid even when $V_2$ becomes vanishingly small.

## APPENDIX C:

## FURTHER DISCUSSION OF WALL FAN EXPERIMENTS

The results of the tests with the source in room 1 are shown in Figs. C1-C4. There is, as with the range hood tests with unheated source, a certain variability between runs at a given flow rate, the variability appearing both in the intra- and inter-room mixing. Run 1 at 10.3 l/sec (Fig. C1) shows a high degree of mixing within room 1 and a delay of about an hour before the appearance of high concentrations in room 2. Run 2 shows much higher concentrations at the point nearest the source (B2) than at all other points, for which the dependence of concentration on time is almost identical. No attempt was made to rigorously fit the data to the mass balance model, but examination of the predictions of the model for several values of the mixing factor showed that the closest approximations to the data were obtained for $\beta \sim 3.5$, 17.5, and 5.0 for runs 1, 2, and 3, respectively. The theoretical curves are, in Fig. C2, shown superimposed upon the spatial average data of run 1. The fit to the room 1 data is seen to be reasonably good; the relatively poor fit to the room 2 data may be accounted for by the changes in mixing factor over time and by the less than complete mixing within room 2.

A similar variablility was seen at 19.4 l/sec; the three runs made are displayed in Fig. C3. Mixing factors of 1-2 were found to produce the best fit, although in none of the runs was the fit to the room 2 concentrations very good. Figure C4 shows the theoretical curves for $\beta=1.0$ superimposed upon the spatial average data of run 3. It is interesting that the mixing factor appears to diminish with increasing fan flow rate. For constant mixing factor, the room 1 concentration is inversely proportional to flow rate; our data indicate that the room 1 concentration at 20.3 l/sec is ~80 % of that at 10.3 l/sec, so that the effectiveness of the fan in removing tracer gas from room 1 has actually diminshed. The causes of this result are not entirely clear, although it is possible that the air movement produced by the fan overcomes the free-convective transport from room 2 to room 1. At higher fan flow rates than those used in these experiments, it is probable that turbulence would produce a high degree of mixing between rooms and, therefore, a high ventilation efficiency. Further tests should be conducted to test this hypothesis.

We turn now to the tests conducted with both the source of tracer gas and the fan in room 2. No indication of a decline in the mixing factor is apparent until a flow rate of 39.2 l/sec is reached. The first run at 10.3 l/sec is consistent with a mixing factor of about 5, similar to the result with the source in room 1 (Figure C5); the transit time between rooms 2 and 1 is seen to be on the order of a few minutes. As would be expected from the mass balance model, the concentration appearing in the outlet duct is greater than that in room 1 throughout the injection period. The second run shows almost perfect mixing, probably as a result of the increased indoor-outdoor temperature difference.

The initial run at 19.4 l/sec (Fig. C6) shows a sharply increased transit time between rooms 2 and 1; ~1.5 hrs elapse before the concentration in room 1 reaches 5 ppm. However, the overall pattern for run 1 is inconsistent with the mass balance model in that the concentrations in rooms 1 and 2 are greater than that in the outlet duct. The second run presents a pattern similar to the second run at 10.3 l/sec, i.e., there is almost perfect mixing; the larger indoor-outdoor temperature difference prevailing during this run may account for this result.

At 31.7 l/sec (Fig. C7), the first run shows relatively poor mixing up to the point at which the outdoor temperature begins to decline. The second run shows a very high degree of mixing, the pattern being quite similar to that found in the second runs at 10.3 and 19.4 l/sec.

When we reach 39.2 l/sec (Fig. C8), we find that the concentration in the exhaust is, for both runs, greater than the concentration in either room, which is indicative of increased local ventilation efficiency. The first run, in particular, shows a very high ventilation efficiency, the tracer gas appearing in room 1 in significant amounts only ~2 hrs after the start of injection. This run also shows a very high local concentration at the sampling point nearest the source, with considerable fluctuation; this result is probably the consequence of a particular combination of air flow patterns and is not representative of the mean concentration in the center of the room. The results offer an opportunity to test the modified mass-balance model of Appendix B, in which one "room" consists of a small volume containing the source and the outlet. Using an arbitrary "room 2" volume of 5000 l, we find mixing factors of 0.35 and 0.75 for the first and second runs, respectively. The theoretical and experimental data for the second run are shown in Figure C9.

At 45.2 l/sec (Fig. C10), we encounter a pattern which is inconsistent with the theory. There is an initial increase of concentration in both rooms, the highest level being greater than that at 39.2 l/sec, but, after about an hour, the levels diminish steadily, so that the integrated concentration over the injection period is considerably below that of the lower rate. The level in the outlet duct is slightly greater than that predicted by theory, even at a mixing factor of 0. The temperature data suggest very strongly that the ambient conditions play an important role in transport, even at flow rates of this magnitude. At the point at which the room concentrations begin to diminish, the indoor-outdoor temperature difference vanishes, thereby removing the driving force of free convection. It is inferred that, at this point, forced convection produced by the fan becomes dominant and the test space is cleared of the tracer gas.

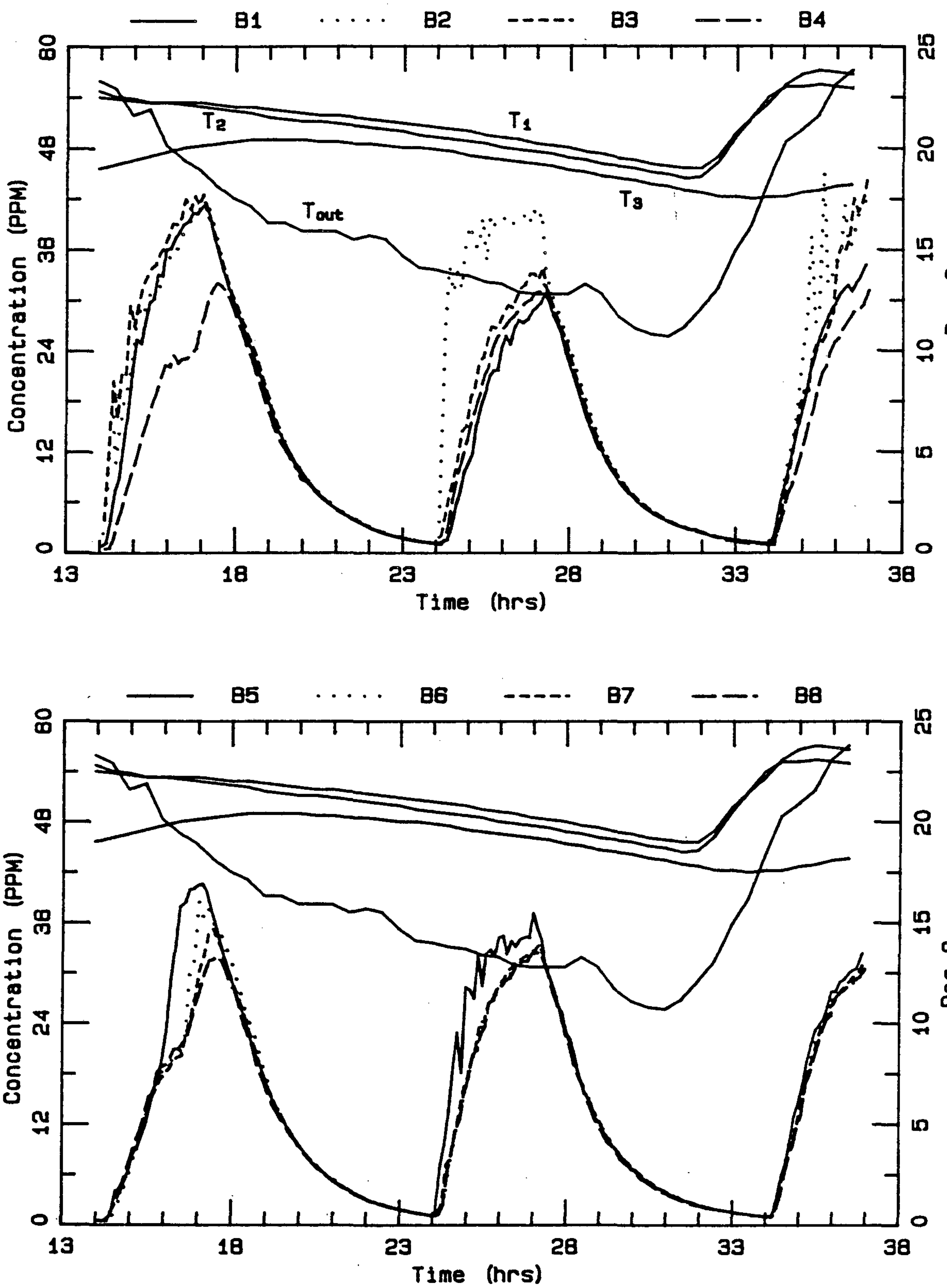



Figure C1. Fan with source in room 1 at 10.3 l/sec.

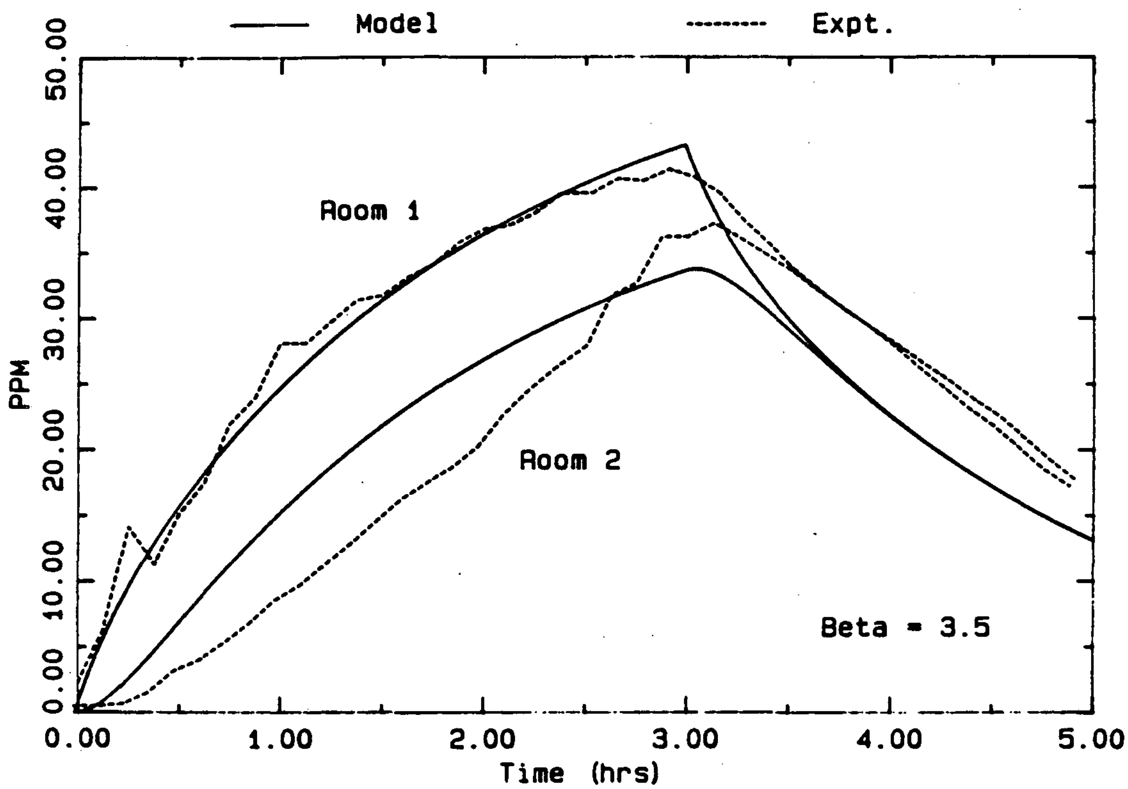
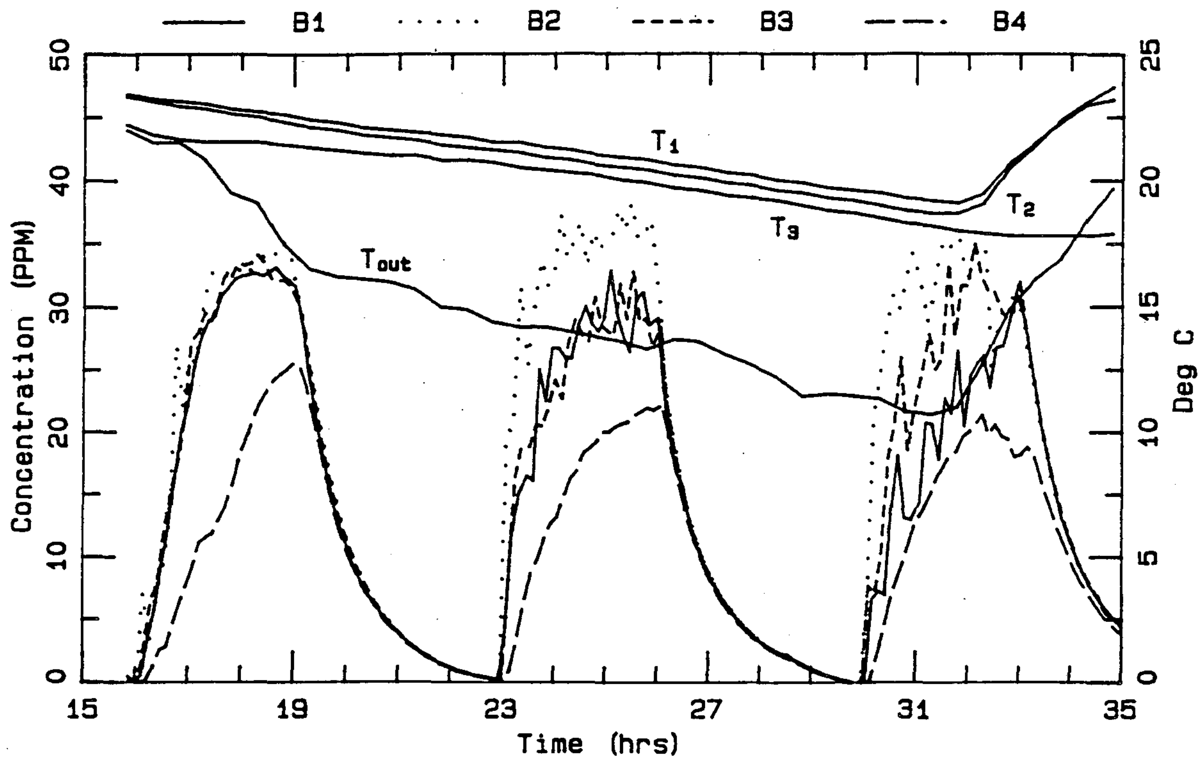



Figure C2. Comparison of model with first run of Figure C1.



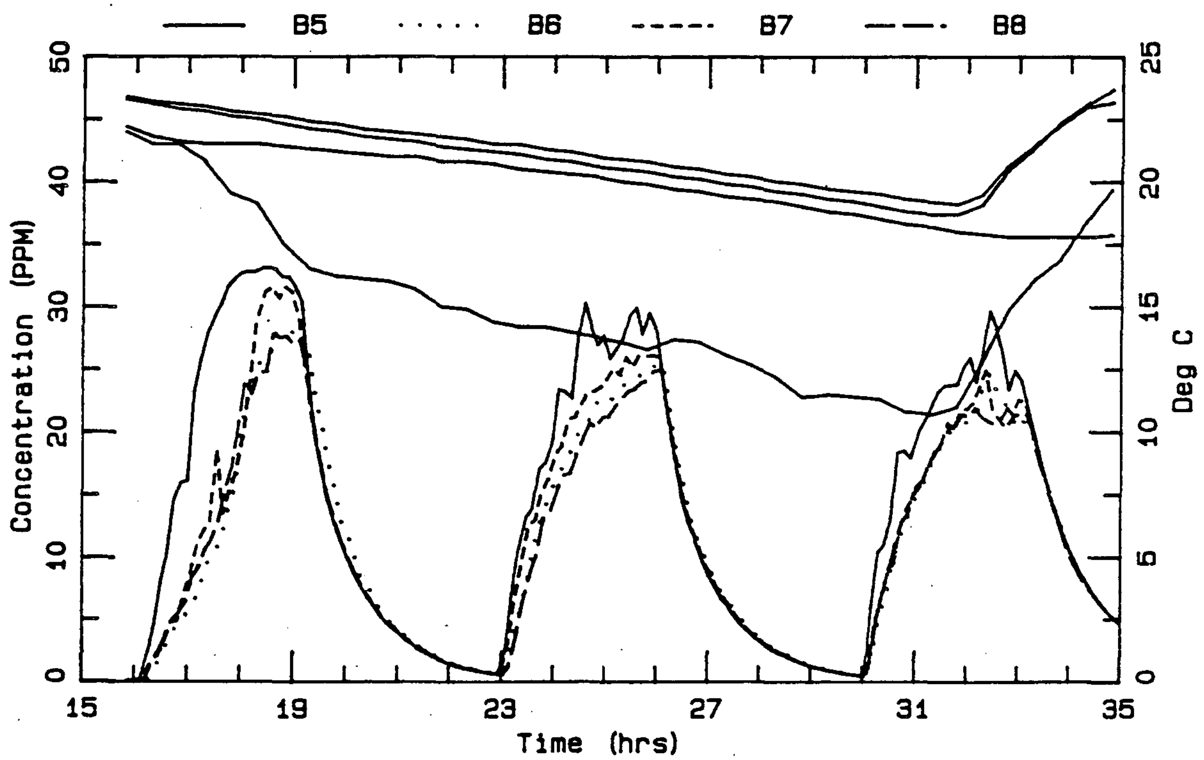




Figure C3. Fan with source in room 1 at 20.3 l/sec.

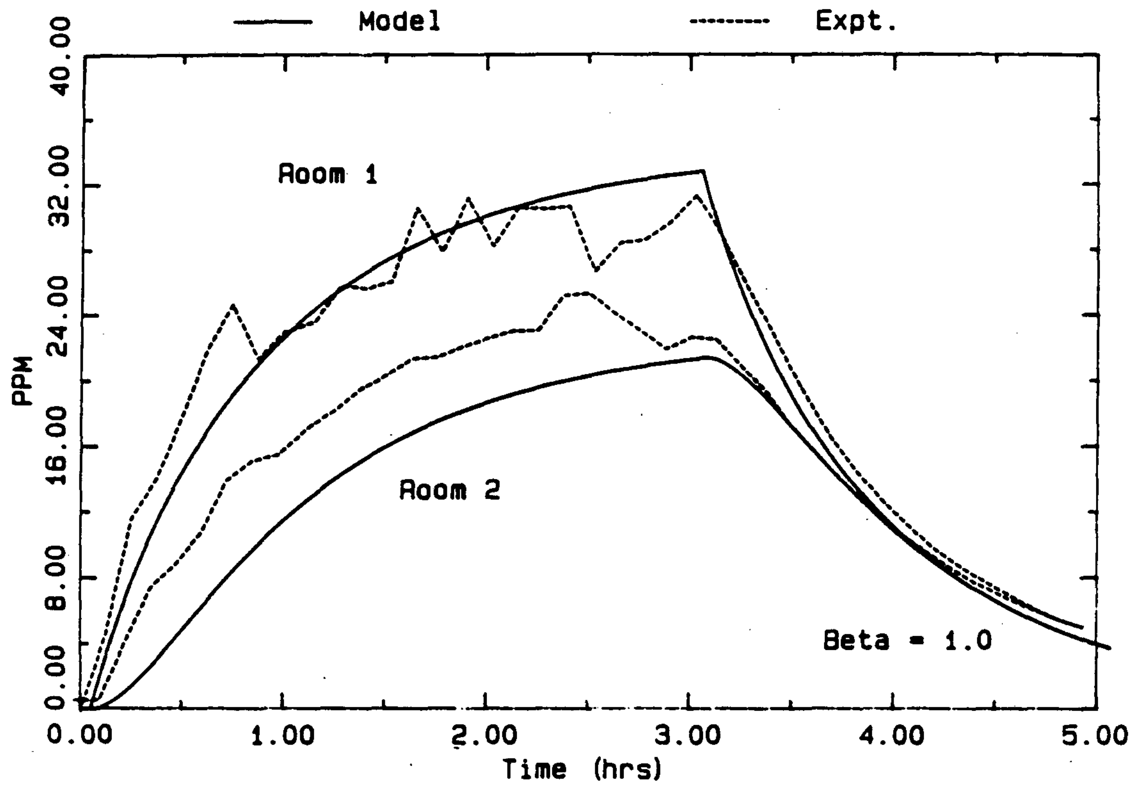



Figure C4. Comparison of model with third run of Figure C3.

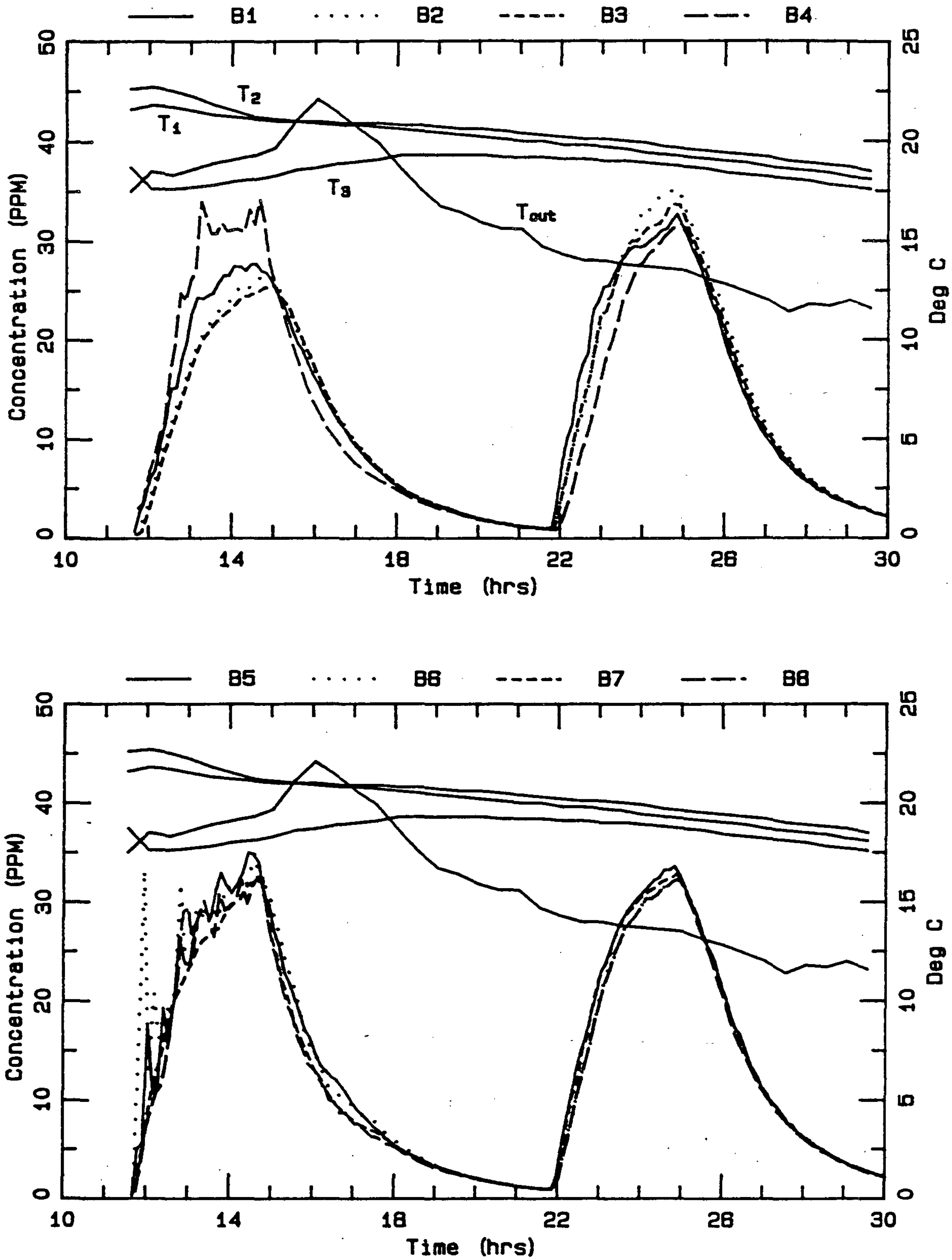
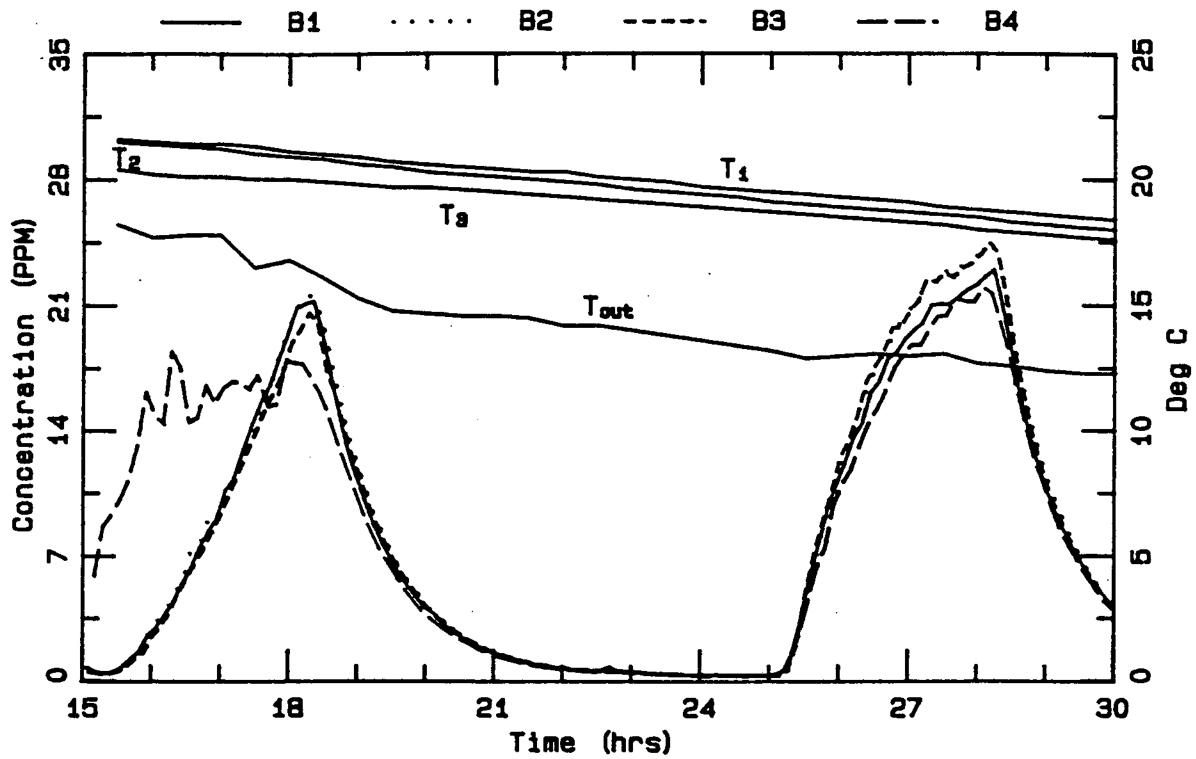



Figure C5. Fan with source in room 2 at 10.3 l/sec.



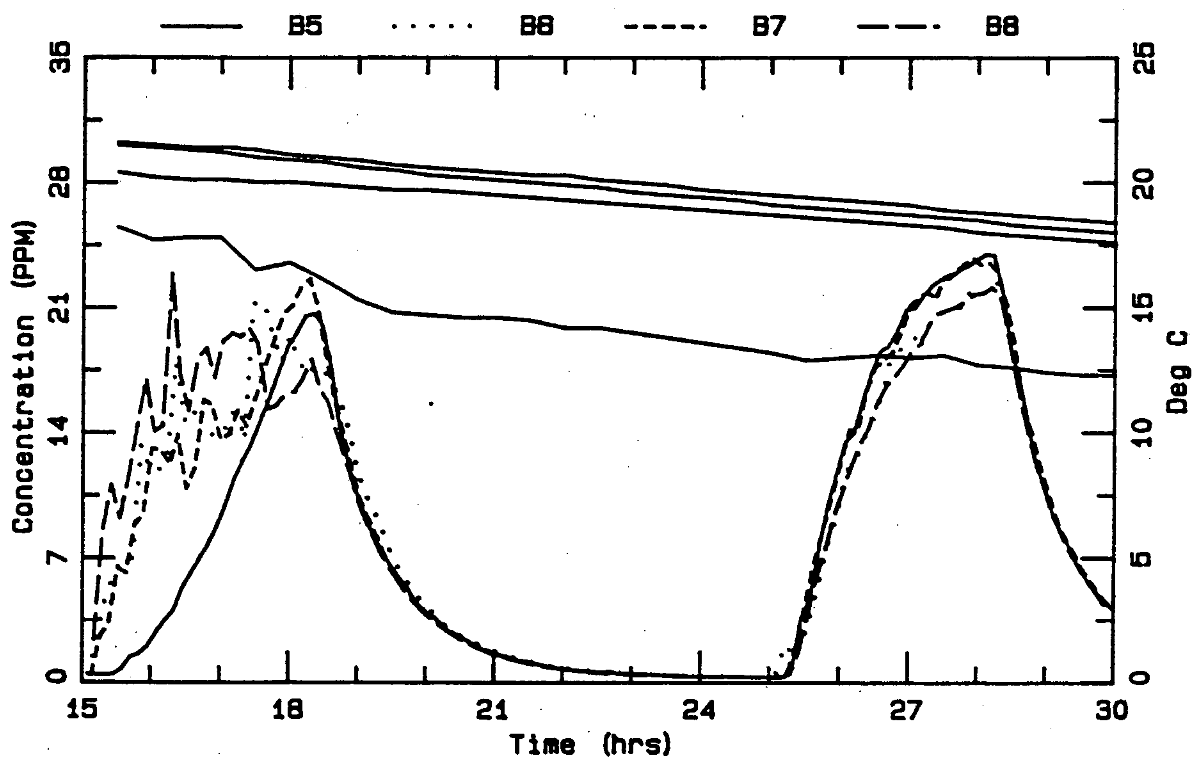




Figure C8. Fan with source in room 2 at 19.4 l/sec.

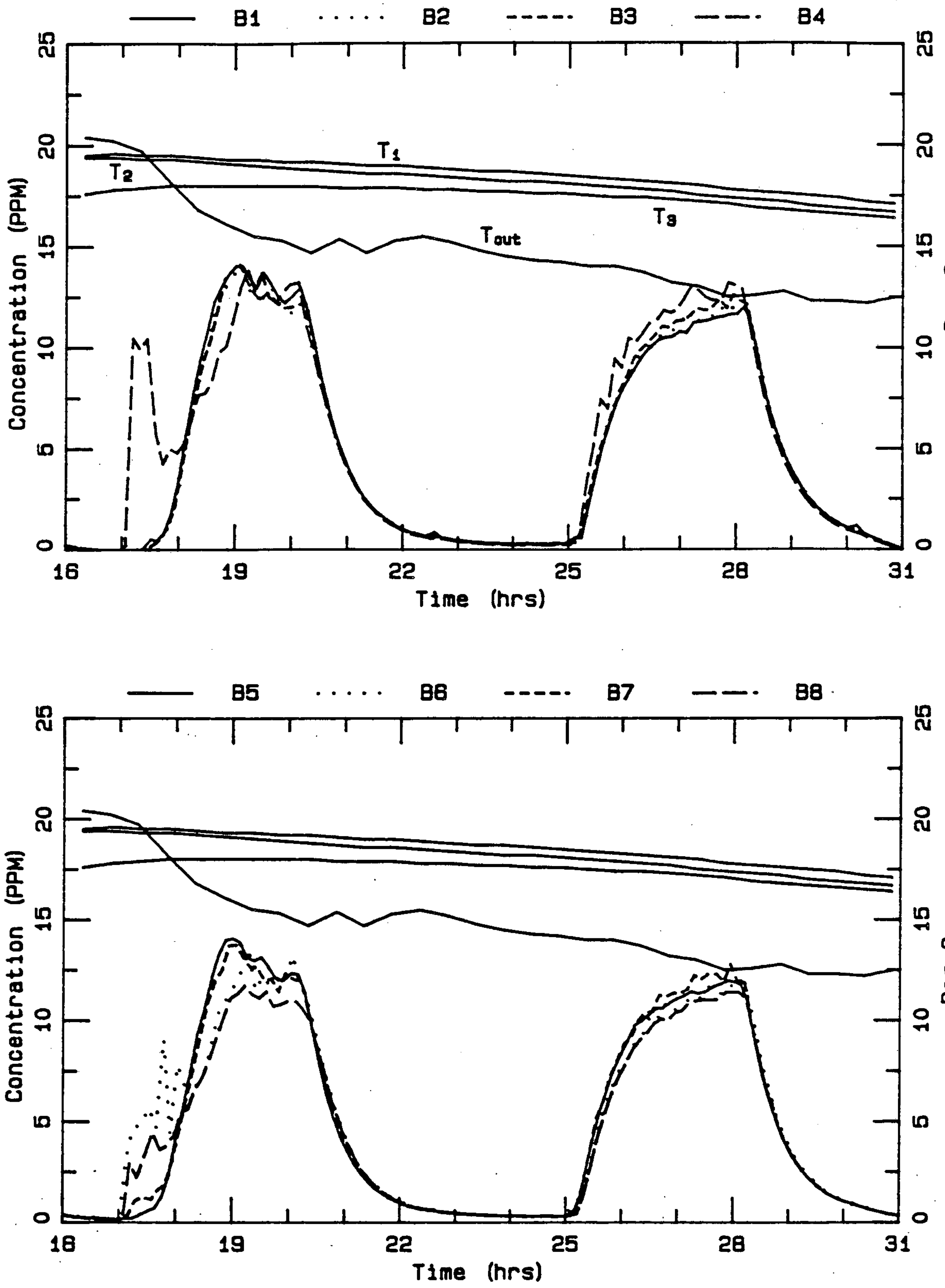



Figure C7. Fan with source in room 2 at 31.7 l/sec.

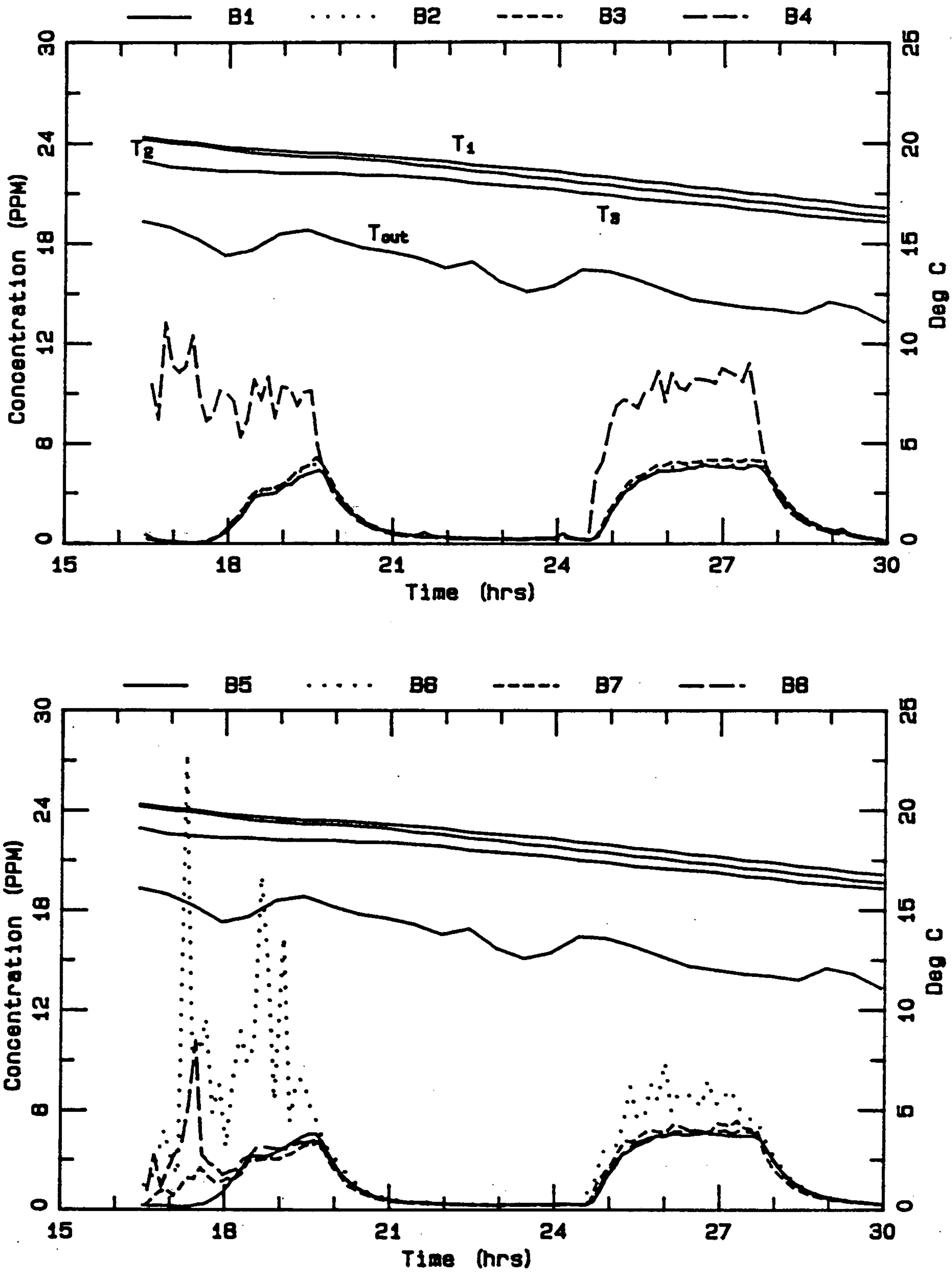



Figure C8. Fan with source in room 2 at 38.2 l/sec.

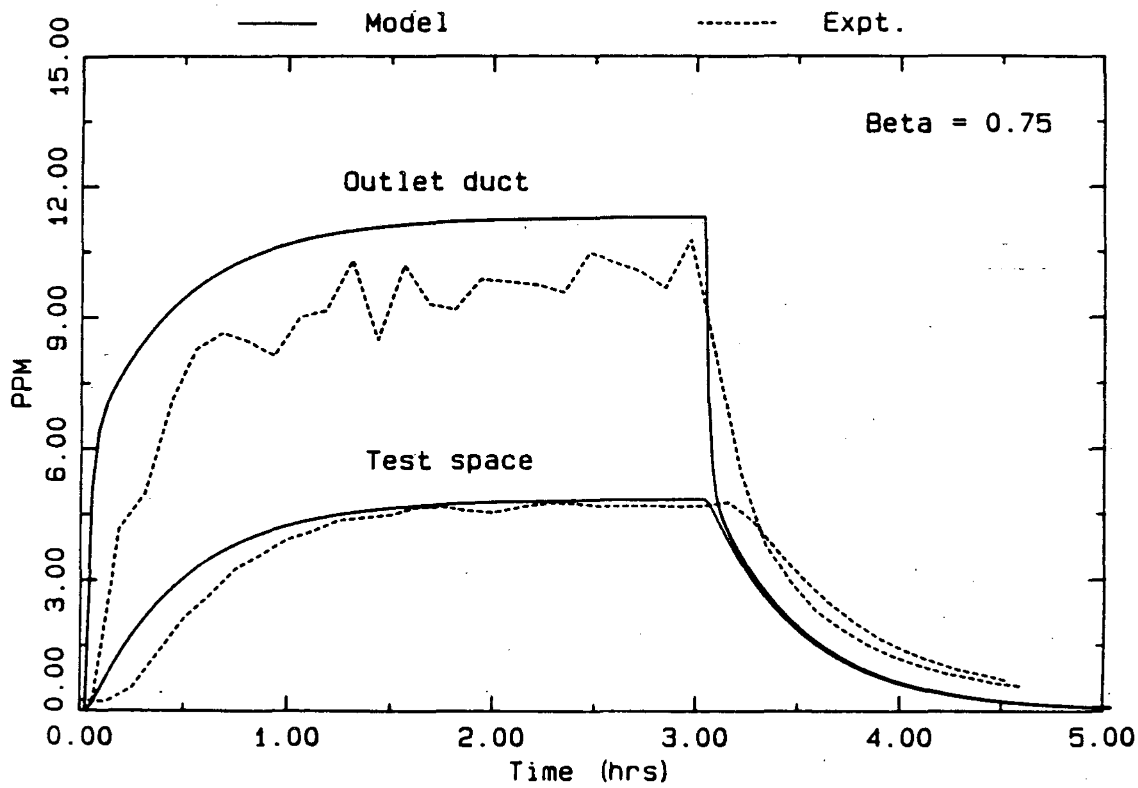
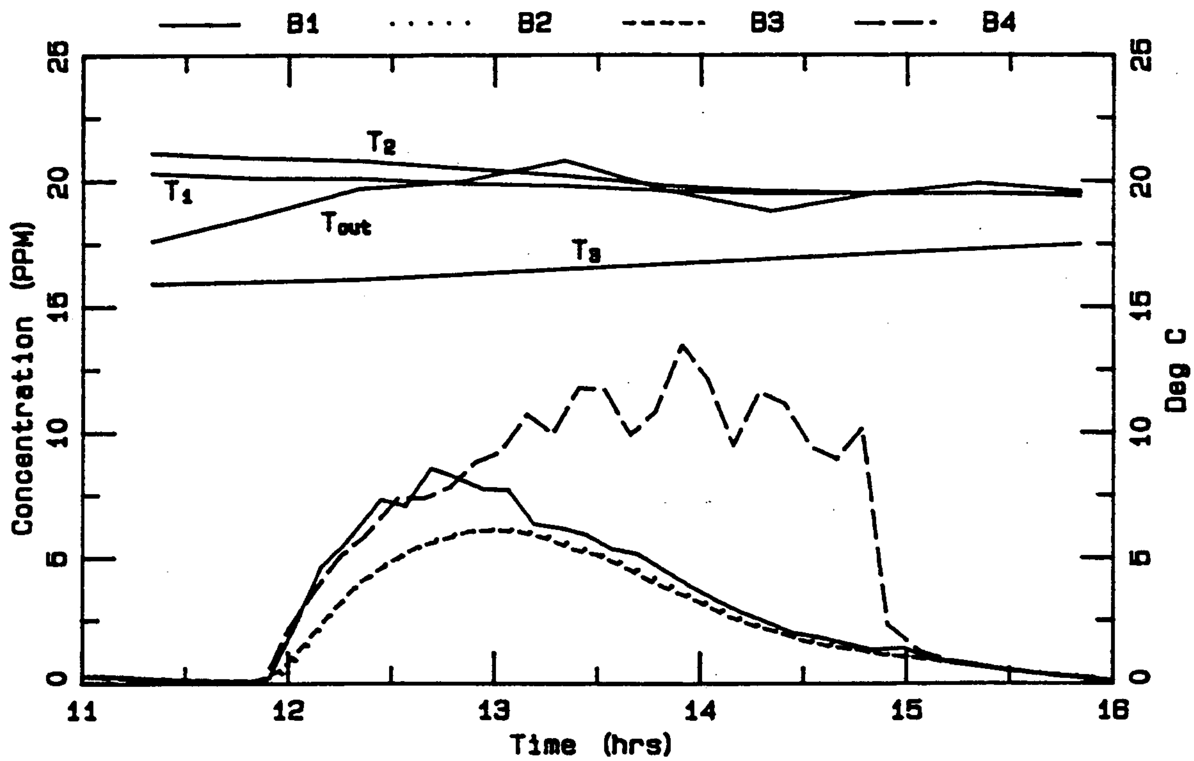



Figure C9. Comparison of model with second run of Figure C8.



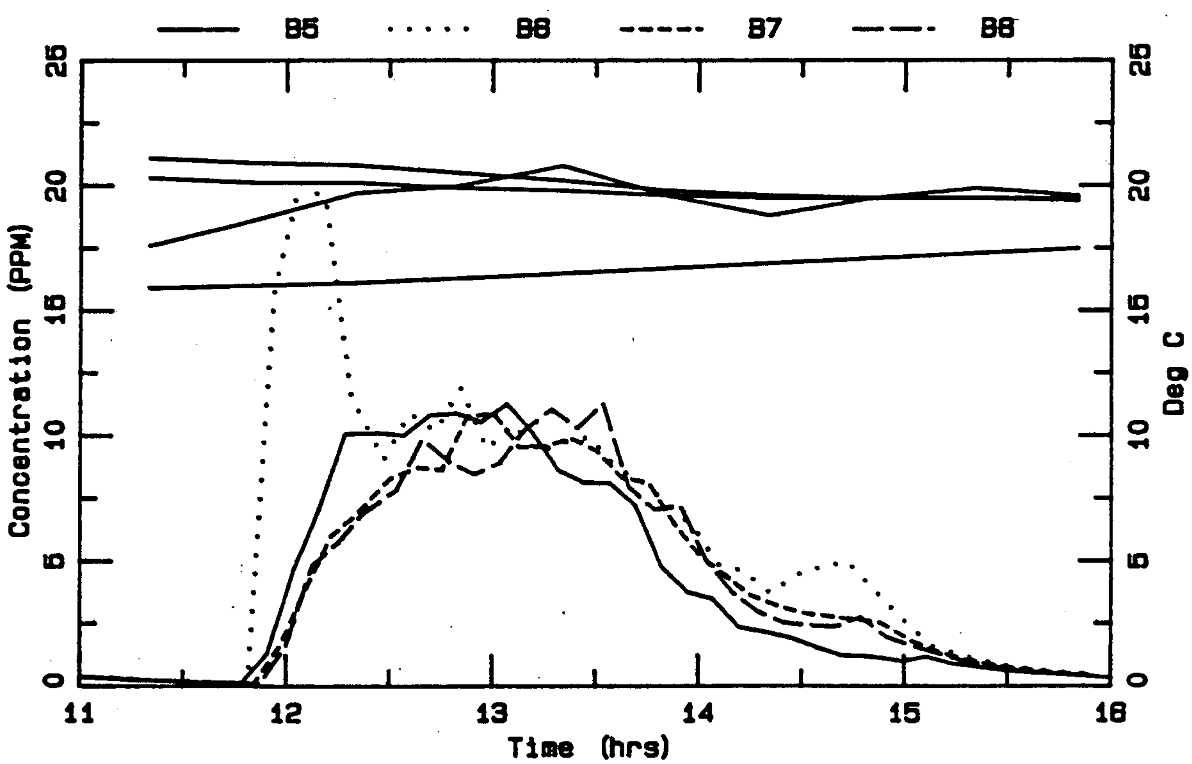




Figure C10. Fan with source in room 2 at 45.2 l/sec.

TECHNICAL INFORMATION DEPARTMENT
LAWRENCE BERKELEY LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY, CALIFORNIA 94720