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APPLICATION OF SCALE MODELLING IN THE DESIGN OF A VENTILATION SYSTEM FOR AN UNDERGROUND CARPARK

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SUMMARY

Modelling techniques were used to verify the success of a "minimal duct" ventilation system proposed for an underground carpark in a prestigious project in Singapore. The procedure to determine the various model scaling parameters, to select the appropriate fans to model the supply and exhaust fans, as well as the "dilution fans", and to simulate the movement of air through the vehicle access ramps are presented. Measurements from the model were found to be reasonable when compared with results from numerical simulation. However, certain modifications were needed to improve the system.

1. INTRODUCTION

The building code in Singapore stipulates the use of mechanical ventilation in an enclosed space such as a carpark when ventilation by natural means is not possible. To reduce the energy required to move large quantity of air inside ducts in a mechanical ventilation system, a "Minimal Duct Ventilation System" was proposed for a two-storey basement carpark in a prestigeous building project in Singapore. In such systems, supply and exhaust ventilation ducts are kept to a minimum and ss much as where possible, ventilation air is taken from openings to the exterior such as vehicular access ramps. Only main supply ducts and exhaust air ducts were used in the lower storey of the basement. Air movement is supplemented by "dilution" fans mounted at strategic locations within the carpark to move air by entrainment towards the exhaust air vents.

The designers of the system have recommended that the ventilation system operate in the following manner:

- Supply and exhaust fans operate on low speed setting to provide 7.5 air changes per hour (ACH).

- When carbon monoxide (CO) concentration exceed the preset levels and flow sensors in sprinkler mains relay sprinkler operation, then supply and exhaust fans shall operate to provide 15 ACH. Dilution fans in affected areas shall respond to dilute and move air to exhaust vent locations.

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- Under extreme conditions, the supply and exhaust fans shall operate at 15 ACH while all dilution fans operate to help air movement towards the exhaust vents.

The ventilation system in each of the two basement levels operates independently of each other.

The objectives of the study was to verify that the desired air flow within the carpark can be achieved in the various modes of operation, to identify the flow stagnation regions, and to observe the efficacy of the dilution fans in promoting air movement.

A scale model of the structure was used in the study. There were several constraints to consider: these included determination of a suitable scale so as to keep down the cost of the model, determination of other model scaling parameters, selection of the appropriate fans to model the dilution fans, and simulation of air movement through the vehicle access ramps.

2. CONTROL PARAMETERS

The important variables of a general ventilation problem are as follows:

V	=	mean velocity flow
Vd	=	velocity of flow through dilution fans
L	=	linear dimension of the body
Q	=	total flowrate
Qd	=	flowrate through dilution fan
Qr	=	flowrate through vehicular ramp
t	=	time scale
$\Delta\rho g/\rho$	=	change of weight per unit mass
ν	=	kinematic viscosity of the fluid (air)

There are nine variables with two basic dimensions from which seven dimensionless parameters can be formed. The following grouping of variables were found to be satisfactory.

(1) $$\frac{Q}{VL^2}=f\left(\frac{V_d}{V},\frac{Q_d}{Q},\frac{Q_r}{Q},\frac{tV}{L},\frac{\Delta\rho gL}{\rho V^2},\frac{VL}{\nu}\right)$$

The group on the left hand side of equation (1) can be considered the flowrate coefficient, or

(2) $$C_Q=\frac{Q}{VL^2}$$

Combining the first and second term on the right hand side of equation (1) gives the momentum ratio. Physically, it represents the ratio of momentum produced by a dilution fan and total momentum of the mean stream.

(3) $$\frac{V_dQ_d}{VQ}=\frac{\rho V_dQ_d}{\rho VQ}$$

The third group on the right hand side of equation (1) is the flowrate ratio. It represents the ratio of flowrate induced from ramps to that of the mean stream.

(4) $$\frac{Q_r}{Q}$$

Equation (4) can also be expressed as,

(5) $$\frac{Q_e - Q_s}{Q}$$

where Q_e is the total flowrate through the exhaust fans and Q_s is the total flowrate through the supply fans.

The fourth term on the right hand side of equation (1) is related to the number of air changes per hour (ACH). Multiplying the term on the left hand side and the fourth term on the right hand side of equation (1) will give equation (6) which represents the number of air change per unit time. t can be considered as a time scale.

(6) $$\frac{Q}{VL^2} \cdot \frac{tV}{L} = \frac{Q}{L^3/t}$$

The fifth term represents the ratio of buoyancy to inertia force of the mean flow. It is significant only when there exist a temperature gradient. For the present ventilation study, this term can be neglected.

The last term on the right hand side of equation (1) is Reynold's Number, or

(7) $$Re = \frac{VL}{\nu}$$

Thus to model the general ventilation problem without temperature gradient, the following equations must be satisfied,

(8) $$\left(\frac{Q}{VL^2}\right)_m = \left(\frac{Q}{VL^2}\right)_p$$

(9) $$\left(\frac{V_d Q_d}{VQ}\right)_m = \left(\frac{V_d Q_d}{VQ}\right)_p$$

(10) $$\left(\frac{Q_e - Q_s}{Q}\right)_m = \left(\frac{Q_e - Q_s}{Q}\right)_p$$

(11) $$\left(\frac{Qt}{L^3}\right)_m = \left(\frac{Qt}{L^3}\right)_p$$

(12)
$$\left(\frac{VL}{\nu}\right)_m = \left(\frac{VL}{\nu}\right)_p$$

where the subscript m refers to model parameters and subscript p refers to prototype parameters, respectively.

The time scale t represents the time required for air particles to travel from one place to another. It is important in studies involving smoke movement. For complete similarity between model and prototype, equations (8) to (12) must be satisfied. In practice, it was not possible to satisfy all of them in the same problem.

3. MODEL DESIGN AND FABRICATION

3.1 Reynold's Number of the model

Equations (8) to (12) form the basis to simulate the parameters of the model. The first three equations can be modelled. However, it was not possible to satisfy both equations (11) and (12) [1]. If equation (11) is satisfied, the velocity of airflow within the model would be too small to measure. Thus the model's Reynold's Number would be much smaller than that of the prototype. For meaningful scaling of the model, the Reynold's number of the flow in the model and that of the flow in the prototype must both be either all turbulent or all laminar.

Under conditions of 15 ACH, the mean stream velocity of the prototype can be found by the relationship

(13)
$$V_p = \frac{15LWH}{3600WH}$$

where L, B and H are the length, width and height of the prototype, respectively. Substituting the values, L = 180 m, W = 108 m, and H = 3 m, gives V_p = 0.76 m/s. Similarly for the model, taking $V_m = V_p$, and $L_m = 0.05L_p$, the Reynolds number for the prototype and model are respectively,

$$Re_p = \left(\frac{HV}{\nu}\right)_p = 1.5 \times 10^5 \qquad Re_m = \left(\frac{HV}{\nu}\right)_m = 7.5 \times 10^3$$

Schlichting [2] reported that the critical Reynold's number of two-dimensional channel flow is about 5 x 10^3 which is smaller than the Reynold's number for the model. Since the Reynold's number of both model and prototype are in the turbulent range the results of experiment would be acceptable.

When modelling 7.5 ACH, the mean velocity is reduced to half of its value for 15 ACH which gives a Reynold's number less than the critical value. However, since this critical value was calculated under smooth surface condition, whereas within the model there are various internal structures such as columns, beams, ramps, and so on which could be considered as roughness or sources of disturbance to the flow the critical

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Reynold's number would be much lower than 5 x 10^3 [2]. It is reasonable to believe that the flow pattern at 7.5 ACH is also turbulent. This could be verified from the experimental results.

3.2 Model fabrication

The model was constructed from plywood which formed the base, and had a top made from perspex sheets. Due to limitation of space and cost, the model was made to 1:20 scale, thus the model dimensions were approximately 10 m long by 5 m across and 0.15 m high. Figure 1 shows the plan of the carpark nodel and the layout of the ventilation system.

The model was constructed upside down so that the ceiling slab formed the base of the model and the floor slab formed the perspex top of the model. The reason for this was that the main beams in the ceiling slab were included in the model to give the relative roughness of the surface and these would have prevented a clear view of smoke movement if it were constructed the right way round. As buoyancy effect of the air was not simulated, this technique will not affect the accuracy of the simulation.

Only one level of the structure was tested at any one time starting with tests on level B2. The model was reconfigured for level B1 upon completion of tests on B2.

Instrument cooling fans were used to simulate dilution fan action. All fans were calibrated according to the manufacturer's characteristic curves to obtain the correct air flowrate before they were installed on the model. In level B1, those fans used to simulate air movement through the ramps to level B2 were calibrated in a similar manner.

3.3 Modelling of supply and exhaust fans

Equations (8) and (10) were used to simulate the flow rate of all the supply and exhaust fans. The flow rates of the supply and exhaust fans of the model is represented by,

$$Q_m = \frac{L_m^2}{L_p^2} Q_p$$

In order to check that equation (10) is valid, the ratio of air supplied from ramps in the prototype compared to that in the model was shown to be equal to 0.64 in both cases. The calculated flowrates for the model in levels B2 and B1 are shown in Tables 1 and 2.

A.	Supply Fans	Location Grid		Prototype Flowrate m^3/h	Model Flowrate m^3/h
(1)	FA-1-5/8	A-K/14	Q_{s1}	161000.0	402.0
(2)	FA-1-4	I/4	Q_{s2}	35800.0	89.5
(3)	FA-1-3	K/6	Q_{s3}	35800.0	89.5
	Total		Q_s	232600.0	581.0
B.	Exhaust Fans	Location Grid		Prototype Flowrate m^3/h	Model Flowrate m^3/h
(1)	EA-B2-1/4	B/4	Q_{e1}	4x16200.0	648.0
(2)	EA-B2-5/6	B/23	Q_{e2}	4x16200.0	648.0
(3)	EA-B2-9/12	M/23	Q_{e3}	4x64800.0	162.0
(4)	EA-B2-13/16	M/1	Q_{e4}	4x64800.0	162.0
	Total			648000.0	1620.0

Table 1
Prototype and model fan flowrates in level B2

3.4 Modelling of dilution fans

Equation (9) was used to assess the characteristics of dilution fans in the model. V_d and Q_d are exhaust velocity and flowrate of dilution fans, respectively. V is the mean stream velocity, and Q can be considered the total exhaust flowrate. It was assumed that the flow rate and exit velocity of the dilution fan in the prototype, shown as the "1X" fan in Figure 1, are V_{dp} = 9 m/s, and Q_{dp} = 8500 m^3/h, respectively and in level B2, V_p = 0.76 m/s, and Q_{ep} = 648000 m^3/h, respectively.

Substituting the above values to equation (9), gives the following relationship,

(14)
$$(V_d Q_d)_m = 191.25 \frac{m^3}{h} \cdot \frac{m}{s}$$

$$= 0.053 \frac{m^4}{s^2}$$

"San-Ace 25", 0.12 m x 0.12 m, instrument cooling fans were selected to model the dilution fans. The flow rate Q_{dm} was 8.3 x 10^{-3} m^3/s which was calibrated in a small wind tunnel.

The exit velocity of the dilution fan in the model was found from equation (14) to be V_{dm} = 6.38 m/s. The exit velocity of the San-Ace 25 cooling fan at 2750 rpm was calculated to be V = 6.79 m/s.

The two values are very close. As the flowrates of 2X and 3X fans in B1 and B2 are twice and thrice the flowrate of the 1X fans, multiples of the San-Ace 25 cooling fans operating in parallel were used to simulate these in the model.

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(2)	EA-B2-5/6	B/23	Q_{e2}	4x16200.0	648.0
(3)	EA-B2-9/12	M/23	Q_{e3}	4x64800.0	162.0
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A.	Supply Fans	Location Grid		Model Flowrate m^3/h
(1)	FA-B1-17/22	L/15	Q_{s1}	322.3
(2)	FA-B1-1/4	B/8	Q_{s2}	170.0
(3)	FA-B1-5/8	B/9	Q_{s3}	170.0
(4)	FA-B1-9/12	B/14	Q_{s4}	170.0
(5)	FA-B1-13/16	B/12	Q_{s5}	170.0
	Total		Q_s	1002.3
B.	**Exhaust Fans**	**Location Grid**		**Model Flowrate m^3/h**
(1)	EA-B1-1/4	B/4	Q_{e1}	393.4
(2)	EA-B1-5/8	B/23	Q_{e2}	359.6
(3)	EA-B1-9/12	M/23	Q_{e3}	366.5
(4)	EA-B1-13/16	M/2	Q_{e4}	350.0
	Total			1469.5

Table 2
Model fan flowrates in level B1

3.5 Simulating flow through vehicular ramps

The ramps from which air flows into the space were modelled by openings of the appropriate size and shape. These included the ramps connecting level B1 to B2 when modelling level B2, and the ramps connecting first storey to level B1 when modelling level B1. The air flow through the ramps would be equal to the difference between the exhaust plus any other flow out of the model, and supply air flow. However, in situations when air flows out of the model from the ramps would require more care to model. These situations occur in modelling level B1 where air flows through the ramps to level B2. Equation (10) shows that the relative flow rate of air supplied from the ramps must be identical for both prototype and model. It was calculated that 64 % of the air going to level B2 must come from the ramps. This is computed for the model as follows:

Air flow from storey 1 to B1 through the ramps must be equal to: $\Sigma Q_e - \Sigma Q_s = 2508.5 - 1002.3 = 1506.2\ m^3/h$

The air velocities through the four ramps connecting B1 to B2 were measured from the model of level B2. The relative flowrates were calculated from the air velocity and size of ramp opening. The when level B2 was supplied with an equivalent of 7.5 ACH, total flow through the four ramps must equal to: 1039/2 = 519.5 m^3/h. The flowrates are shown in Table 3. The flows from these ramps were simulated using fans to exhaust air from the level B1 model space.

Ramp (n)	(1)	(2)	(3)	(4)	Total
Grid Location	M/4	G/21	L/6	B/9	
Flowrate Q_{rn} m^3/h	144.2	166.3	152.6	56.4	519.5

Table 3
Flowrates through ramps from level B1 to level B2

4. NUMERICAL SIMULATION

A numerical simulation of the flow under 15 ACH in level B2 was conducted. All air supplied via supply vents and ramps, and air exhausted via exhaust vents were modelled on constant air flowrate. Dilution fan action was not simulated. The result of the simulation are shown in Figure 2. The general flow pattern agrees well with the experimental result shown in Figure
3(a).

5. EXPERIMENTAL RESULTS AND DISCUSSION

5.1 Test Procedure

Consultation with the designers were made upon completion of model construction. Preliminary smoke visualisation tests were conducted on the model to view the general flow pattern. Recommendation for adjustments to the model were discussed and agreed with the designers before commencement of flow measurements according to a grid of probe points on the model. Observations of the smoke movement within the model were recorded manually and plotted onto the floor plans of the respective levels.

For each level, measurements and flow pattern visualisation were made according to the following schedule:

- General air flowrate to simulate 15 ACH with and without dilution fans in action.
- General air flow rate to simulate 15 ACH with selected dilution fans in action.
- General air flowrate to simulate 7.5 ACH without dilution fan.

5.2 Level B2

5.2.1 Modifications

From preliminary test observations, modifications were made to the model to reduce regions of dead-air motion. These modifications include relocation or addition of dilution fans, and redistribution of exhaust fan capacities. The flowrates of Exhaust Fan EA-B2-1/4 was increased to 4x32400 m^3/h while that for Exhuast Fan EA-B2-13/16 was reduced to 4x48600 m^3/h. The flowrates in the model was changed accordingly. Figure 1(a) shows the system layout after modifications.

5.2.2 Discussion

The air flow pattern for 15 ACH flow without operation of dilution fan is similar to the results from the numerical study. There were no distinct dead-air region as shown in Figure 3(a). Air from ramp opening at Grid L/6-7 causes flow towards exhaust vent at Grid N-M/23.

With dilution fan in operation at 15 ACH, movement of air towards the exhaust vents were increased as shown in Figure 3(b). A significant difference in the flow caused by dilution fan operation was observed along Grid L/11-13 where the direction of flow was reversed.

For 15 ACH with selected dilution fans in operation, velocity measurements showed that there was little change in the air flow pattern in the regions where all dilution fans were operating. Thus it would be reasonable to expect the flow patterns to be similar to those shown in Figure 3(b) where all dilution fans in the model are in action.

The flow pattern at 7.5 ACH without dilution fan action is similar to the pattern for 15 ACH. The velocities measured were generally less than that for 15 ACH.

5.3 Level B1

5.3.1 Modifications

Some dilution fans in level B1 were relocated or added to improve air flow in the model. The system layout during the test is shown in Figure 1(b). Preliminary tests also revealed that substantial amount of air could be sucked into level B2 through the ramps. Walls were added to close the openings on the side of the ramps leading to B2 to reduce this entrainment effect, as it would be possible that smoke originating in B1 could be moved to B2.

5.3.2 Discussion

Figure 4(a) shows the air flow in level B1 at 15 ACH with dilution fans. The flow paths are well defined in the directions towards the exhaust air outlets. Improvement in the air flow was achieved by enclosing three sides of the ramps leading to B2. There was no significant dead-air region.

There was no significant change in the air flow pattern without dilution fans. When selected dilution fans were operating, the flow pattern did not differ significantly from the situation when all dilution fans were operating.

At 7.5 ACH air flowrate the pattern are generally similar the case of 15 ACH flow without dilution fans except for the difference along Grid A-E/8-14 where localised dead-air regions exist. This is shown in Figure 4(b).

6. CONCLUSION

The study showed that the modelling technique used gave reasonable results. This could be verified with the result of the numerical study. The study enabled the design to be checked and corrections were made in the sizing of the exhaust air fans to correct the imbalance of air flow. Verification of the alignment and location of the dilution fans which constituted an important component of the ventilation design were also made possible from the test observations. Finally, it was shown that air flow through the ramps connecting the two levels could be reduced by enclosing the sides of the ramps.

7. NOMENCLATURE

B	width dimension of body, m
H	height dimension of body, m
L	linear dimension of body, m
g	acceleration due to gravity, 9.8 m/s^2
Q	flowrate, m^3/h
Q_d	flowrate through dilution fan, m^3/h
Q_e	flowrate through exhaust fans, m^3/h
Q_r	flowrate through vehicular ramp, m^3/h
Q_s	flowrate through supply fans, m^3/h
t	time scale, s
V	mean stream velocity, m/s
V_d	velocity of flow through dilution fans, m/s
$\Delta\rho g/\rho$	change of weight per unit mass, dimensionless

Dimensionless groups

Re	Reynolds Number, $Re = VL/\nu$

Greek letters

Δ	difference
Σ	sum
ν	kinematic viscosity of the fluid (air), m^2/s
ρ	density of fluid, kg/m^3

Subscripts

m	model parameters
p	prototype parameters

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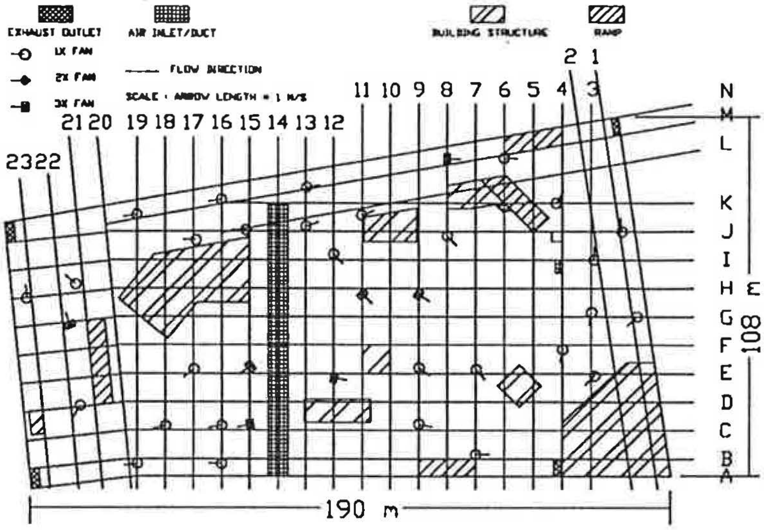
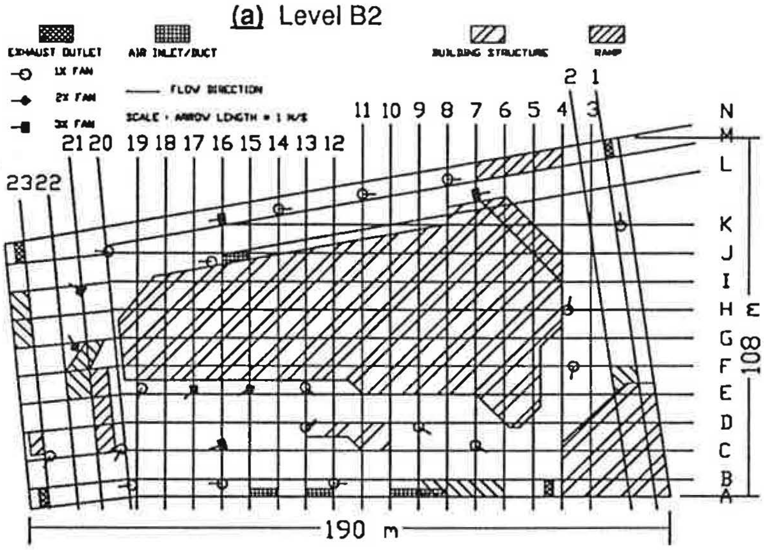


(a) Level B2



(b) Level B1

Figure 1 Floor plans and system layout

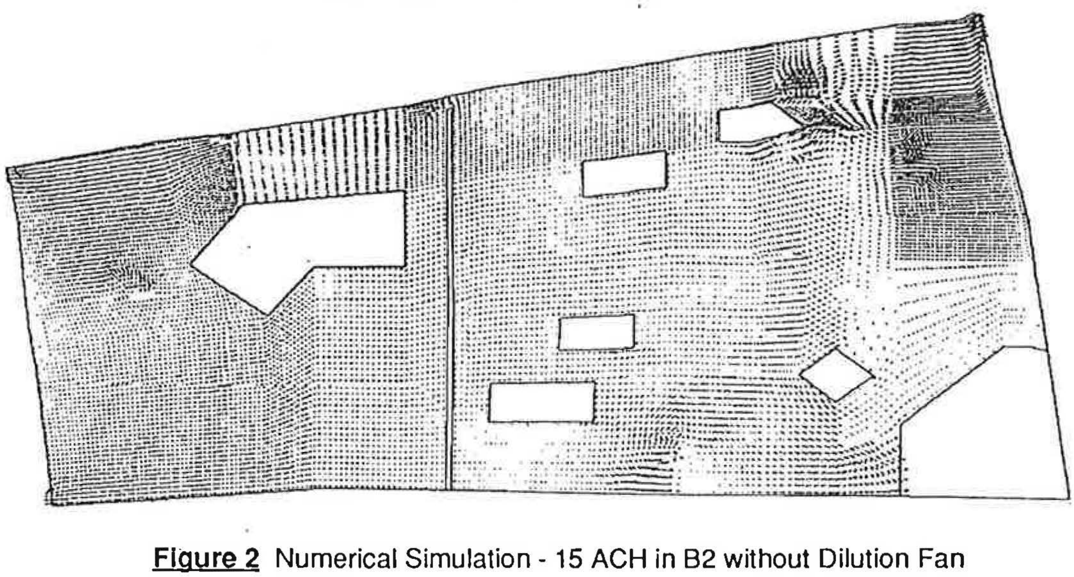
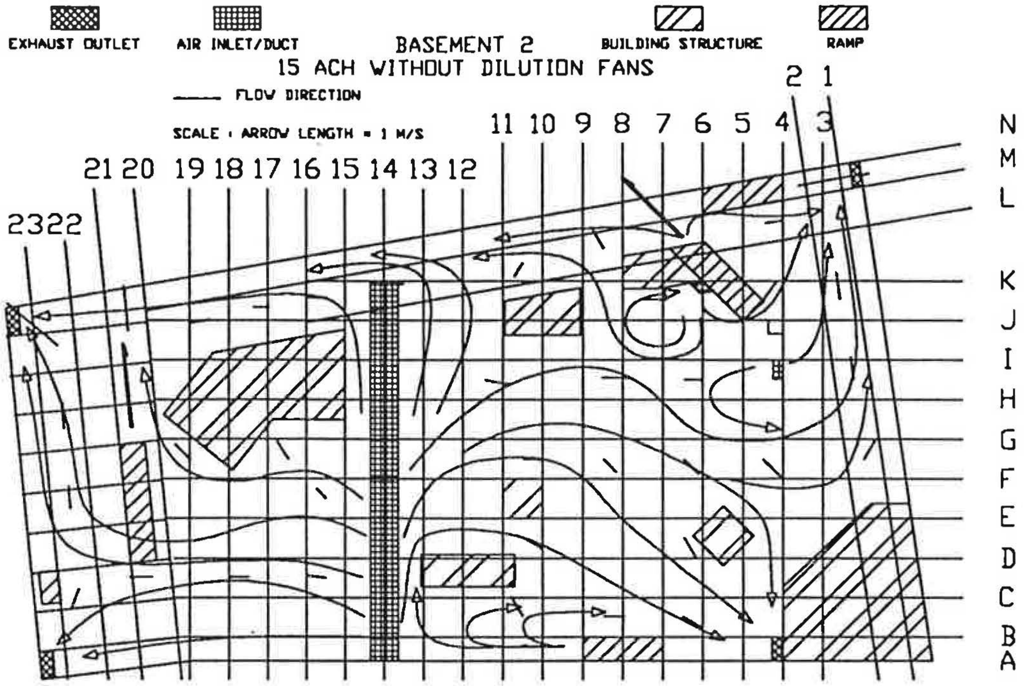
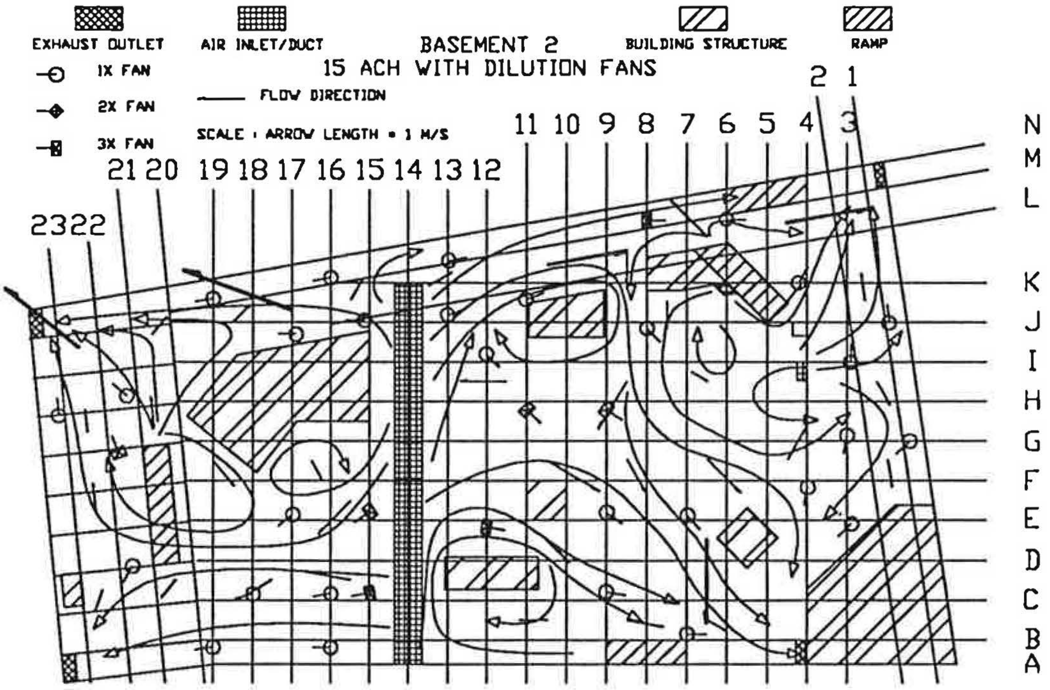


Figure 2 Numerical Simulation - 15 ACH in B2 without Dilution Fan

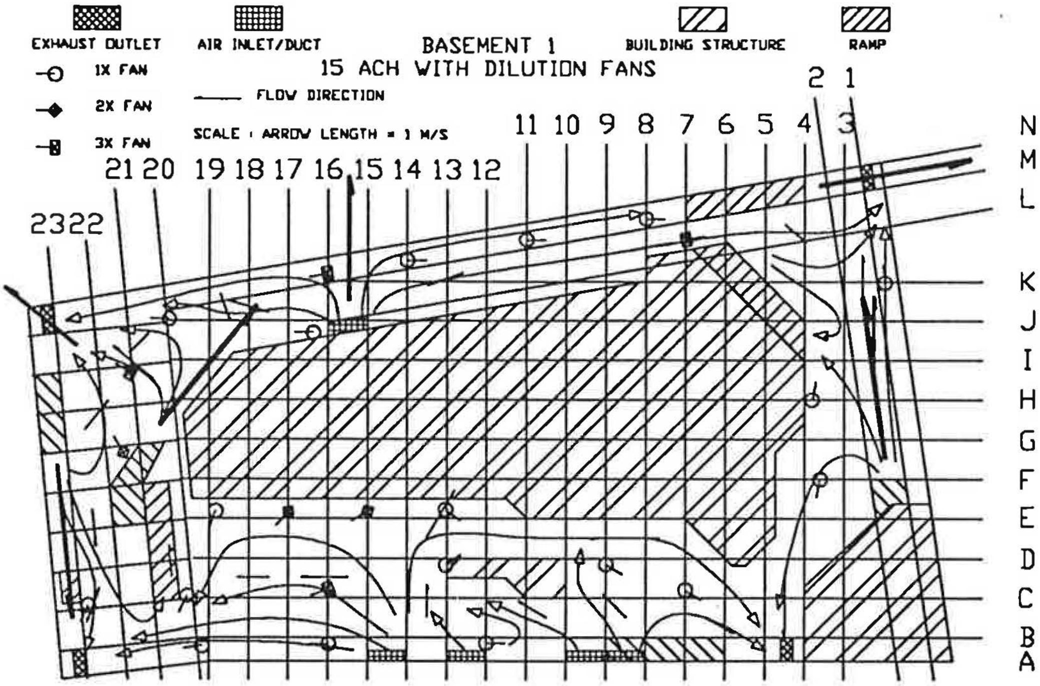


(a) 15 ACH without dilution fans

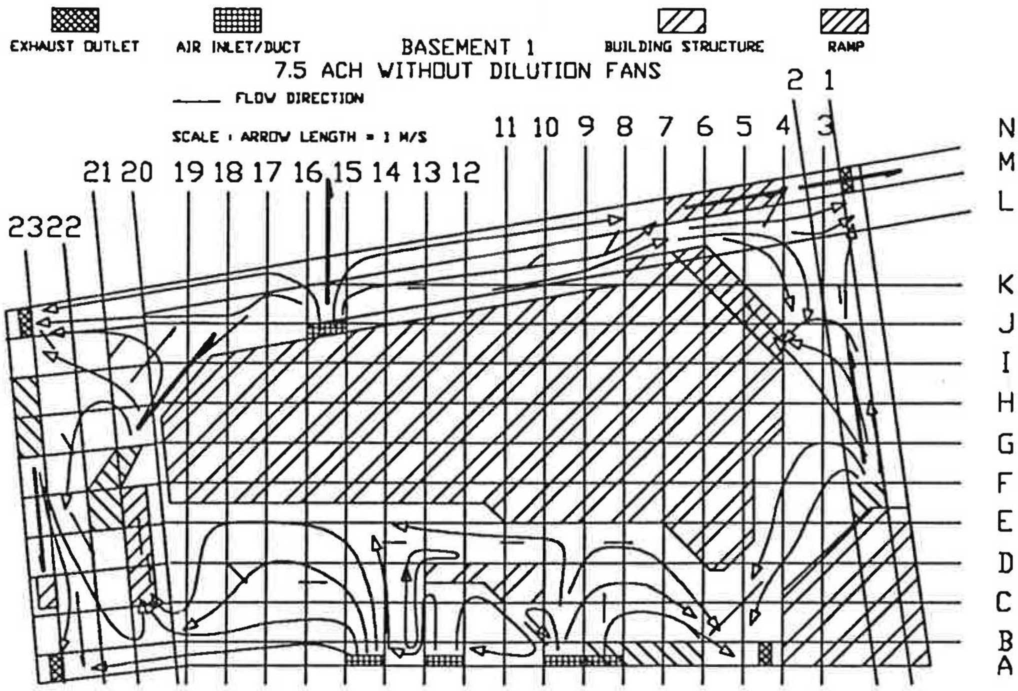


(b) 15 ACH with dilution fans

Figure 3 Flow observation - Level B2



(a) 15 ACH with dilution fans



(b) 7.5 ACH without dilution fans

Figure 4 Flow observation - Level B1